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A feedforward approach to mid-ranging control

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ABSTRACT

Mid-ranging control is a control strategy that is used when there are more than one manipulated variable available to control a process variable. Mid-ranging control handles the redundancy by coordinating the roles of the different manipulated variables. The most common approach is to introduce valve position controllers (VPC) that control the relations between the manipulated variables. In this paper, some drawbacks of the VPC approach are pointed out, and a new control strategy that overcomes these drawbacks is presented. Instead of using valve position controllers, the new strategy uses feedforward control to coordinate the manipulated variables. Design methods for both the new strategy and the VPC controller are presented in the paper.

1. Introduction

The regulatory control layer is an important part of process control instrumentations. Development and improvements of basic control structures in this layer can give large improvements and increase the efficiency of process control plants. See Hägglund (2013) and Hägglund and Guzmán (2018). This paper presents a new mid-ranging control strategy that provides significant improvements compared to previous strategies.

Mid-ranging control is a control strategy that is used when there are more than one manipulated variable available to control a process variable. This paper is restricted to the most common case when there are two manipulated variables and one process variable. The mid-ranging control problem is first illustrated by two very common applications.

Application 1: Precise flow control using parallel flows

In the first application, the problem is to control a flow that may vary significantly with a high precision. Since the flow may be large, a large valve or pump is required. It is, however, often difficult to obtain a high precision using these large final control elements. A large valve will, e.g., get friction and backlash after some time in operation that makes precise control impossible.

Instead of having just one large pump or valve, the solution presented in Fig. 1 can be used. The flow is divided into two parallel flows. A large valve, v_2, placed on a large tube, ensures that the large flow variations can be handled. A small, fast, and precise valve, v_1, placed on a smaller tube, takes care of the small and fast variations to provide the precise control.

Suppose that the small valve v_1 is in the middle of its operating range and that only small disturbances are acting on the system. In this case, one controller that manipulates valve v_1 is able to take care of the control problem. However, when larger disturbances occur, valve v_1 may saturate. In this case, the larger valve v_2 must also be manipulated to ensure that the desired flow is reached, and that the small valve returns to its operating region again to be able to act on future disturbances. This problem is handled by mid-ranging control strategies. □

Application 2: Flow control using both a pump and a valve

The second application of mid-ranging control is presented in Fig. 2. Also in this application, there is a flow to be controlled, and the flow may vary significantly. The flow is to be controlled by both a pump and a valve. The electrical power needed to drive a pump is proportional to the cube of the pump speed. To save energy, it is therefore desirable to keep the speed of the pump as low as possible. This is obtained by keeping the valve as close to fully open as possible. It is often recommended to have the valve operating at around 90% open in steady state. This problem can be handled using mid-ranging control strategies. □

Applications 1 and 2 give simple and very common examples of mid-ranging control problems. Mid-ranging control problems do also appear in more complicated applications. Allison and Ogawa (2003) and Karlsson, Slätteke, Wittenmark, and Stenström (2005) describe applications in the pulp and paper industry, and Haugwitz, Henningsson, Velut, and Hagander (2005), Johnsson, Sahlin, Linde, Lidén, and Hägglund (2015), and Velut, de Maré, and Hagander (2007) have applied mid-ranging strategies for control of bio reactors. Soltesz, Dumont, van Heusden, Hägglund, and Ansermino (2012) describes use of mid-ranging control in a medical drug delivery system and Santillo, Magner, Uhrish, and Jankovic (2016) presents an automobile application where mid-ranging is used to control a three-way catalyst. Several additional applications are given in Shinskey (1981).

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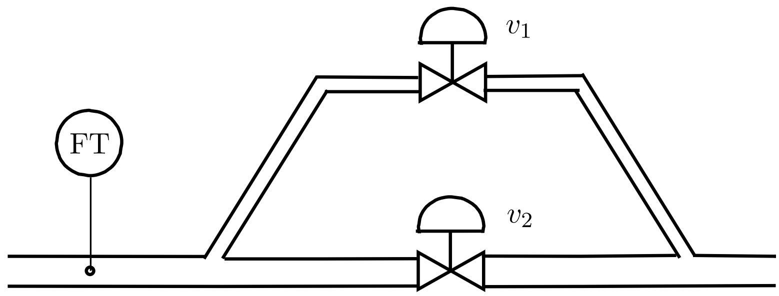

Fig. 1. Mid-ranging control Application 1.

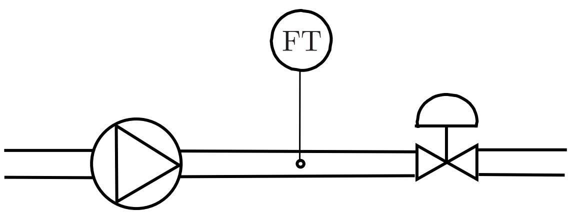

Fig. 2. Mid-ranging control Application 2.

Mid-ranging control shows up under several names. The oldest and most common approach is valve position control (VPC). In Balchen and Mumme (1988), mid-ranging control is described under the heading parallel control, Popiel, Matsko, and Brosilow (1986) name the approach coordinated control, and Henson, Ogunnaike, and Schwaber (1995) use the notation habituating control. Yet another name is input resetting.

Mid-ranging control can be accomplished using model predictive control, and this is sometimes a good solution if such a controller is already present. However, as stated in Alsop (2016), "...due to the cost and complexity of installing MPC, it is unlikely that this technology can be justified for mid-ranging control unless it forms a part of a wider control strategy". The Ref. Alsop (2016), addresses several practical issues for implementation of mid-ranging control.

The most common way to solve the mid-ranging control problem is to use the valve position control approach. This approach is presented in the next section. The VPC works well in many applications, but there are some severe drawbacks associated with the method. These drawbacks are also outlined in the next section.

In Section 3 a new mid-ranging control strategy that overcomes the drawbacks of the VPC is presented. It is based on feedforward control and is named Feedforward Mid-Ranging Control (FFMRC). A design procedure and simple tuning rules for the FFMRC are derived in Section 4. Simulation examples are given in Section 5.

2. Valve position control

In the valve position control approach, a valve position controller is added to the control configuration. The VPC controls one of the control signals to a desired setpoint. The approach is illustrated for the two applications given in the previous section.

Application 1: Precise flow control using parallel flows

The valve position control strategy applied to Application 1 is illustrated in Fig. 3. Flow controller FC takes the flow setpoint y_{sp} and the flow signal as inputs and manipulates the small valve v_1. A second controller, the valve position controller VPC, takes the control signal from FC as input and tries to control it to a setpoint u_{sp} by manipulating the large valve v_2. If both controllers have integral action, the flow will be at the setpoint y_{sp} and the valve v_1 will be at the setpoint u_{sp} in steady state. □

Application 2: Flow control using both a pump and a valve

Fig. 4 illustrates valve position control applied to Application 2. Flow controller FC takes the flow signal and the flow setpoint y_{sp} as inputs and controls the valve. The VPC controller takes the control signal from FC and the corresponding setpoint u_{sp} as inputs and tries to control the valve position to the setpoint u_{sp} by manipulating the pump. If both controllers have integral action, both the flow and the valve position will be at their setpoints in steady state. □

A block diagram of the valve position control strategy is given in Fig. 5. Process P_1 and controller C_1 together form a fast and precise feedback loop. The VPC controller C_2 controls the output of controller C_1 via the process output y. This means that the output of controller C_1 is controlled by driving the process output y away from the setpoint. The control performance can be improved by adding a feedforward path from control signal u_2 to controller C_1, see e.g. Åström and Hägglund (2006). However, this feedforward can be quite difficult to design and is seldom used in practice.

Controller C_1 is normally a PI controller and it can be tuned in standard ways. Controller C_2 must, however, be tuned conservatively so that it does not disturb the other loop too much. The idea is that the VPC controller should adjust the output from C_1 slowly in the background. It is often recommended that VPC controller C_2 is a pure integrating controller without proportional action. See e.g. Allison and Isaksson (1998) and Shinskey (1996). It means that there is only one parameter to tune in this controller, the integral gain.

2.1. Drawbacks with the VPC approach

The VPC approach is a simple and the most common way to treat the mid-ranging control problem, and it works well in many applications. There are, however, some drawbacks associated with the approach.

As mentioned above, it is recommended that controller C_2 has integral action. If the output from C_2 manipulates a large valve, as in Application 1, it is then likely that stick–slip motion will occur because of stiction and backlash in the valve. This will disturb the other loop and the control precision will be lost.

It is possible to avoid stick–slip motion by introducing a deadzone in controller C_2 like the one presented for the new strategy in Section 3. Another way to avoid the stick–slip motion is to remove the integral action by letting C_2 be a P controller. It is interesting to see how large the control error then becomes, i.e. the difference between u_1 and its desired value u_{sp}.

From Fig. 5, the transfer function between flow setpoint y_{sp} and control signal u_1 is given by

$$U_1 = \frac{C_1}{1 + C_1 P_1 - C_1 P_2 C_2} Y_{sp}$$

Since C_1 is a PI controller, it will be approximately $K_1/(sT_{i1})$ at low frequencies, where K_1 and T_{i1} are the gain and the integral time of the controller. Controller C_2 is a P controller in this calculation with gain $-K_2$. (The gain is negative, since a positive control error $u_{sp} - u_1$ should result in a decreased control signal u_2.) Finally, processes P_1 and P_2 are assumed to be stable which means that they can be approximated by

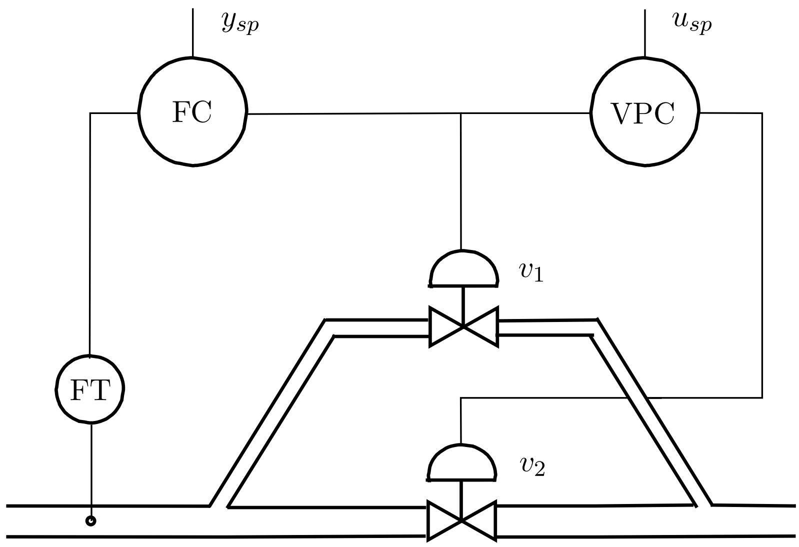

Fig. 3. Valve position control applied to Application 1.

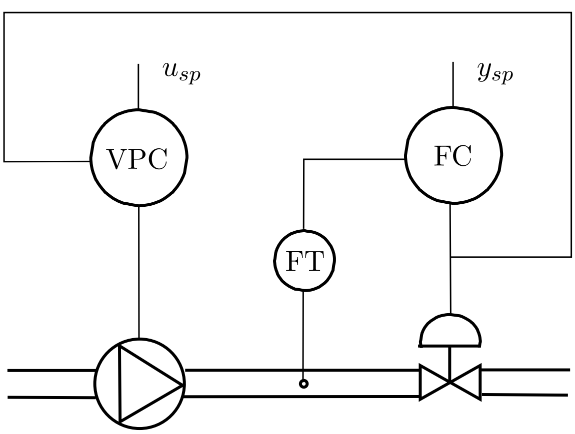

Fig. 4. Valve position control applied to Application 2.

their static gains K_1 and K_2, respectively, at low frequencies. This gives the following relation between Y_{sp} and U_1 in steady-state, i.e. when $s \to 0$.

$$U_1 = \frac{1}{K_{p1} + K_{p2}K_2} Y_{sp}$$

Suppose that a step change in the setpoint with magnitude Δy_{sp} is performed. The steady-state change Δu_1 then becomes

$$\Delta u_1 = \frac{1}{K_{p1} + K_{p2}K_2} \Delta y_{sp} \tag{1}$$

This is normally not acceptable. It means that a change in the flow setpoint will result in a significant change of the steady-state value of control signal u_1. Using e.g. the numerical values of the parameters used in the simulations in Section 5, the factor in Eq. (1) becomes 0.36. It is normally not important the obtain $u_1 = u_{sp}$ in steady state, but it is a severe problem that the deviation in practice is unknown. The deviation may also be quite large, and this may cause problems, especially in cases like Application 2 where the goal is to be close to the operating limit of the control signal. For this reason, VPC is almost never used without integral action in the VPC controller.

A second drawback with the VPC approach is illustrated in Fig. 6, where the VPC approach is applied to the two process models

$$P_1 = \frac{K_{p1}}{(1 + sT_{p1})^2} \qquad P_2 = \frac{K_{p2}}{(1 + sT_{p2})^2}$$

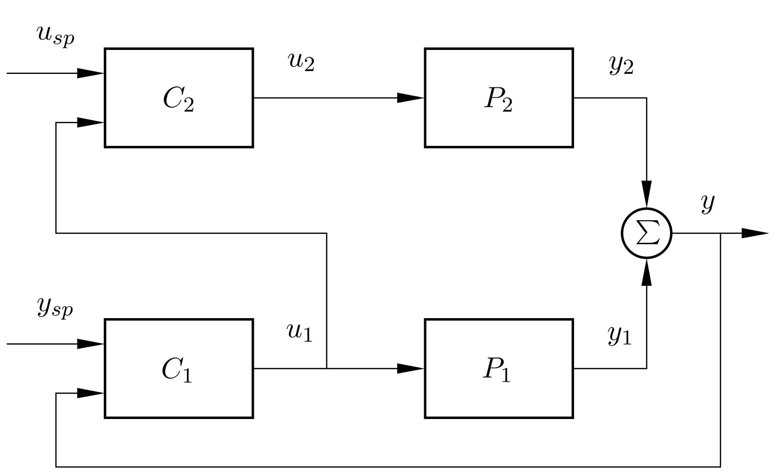

Fig. 5. Block diagram describing the valve position control approach.

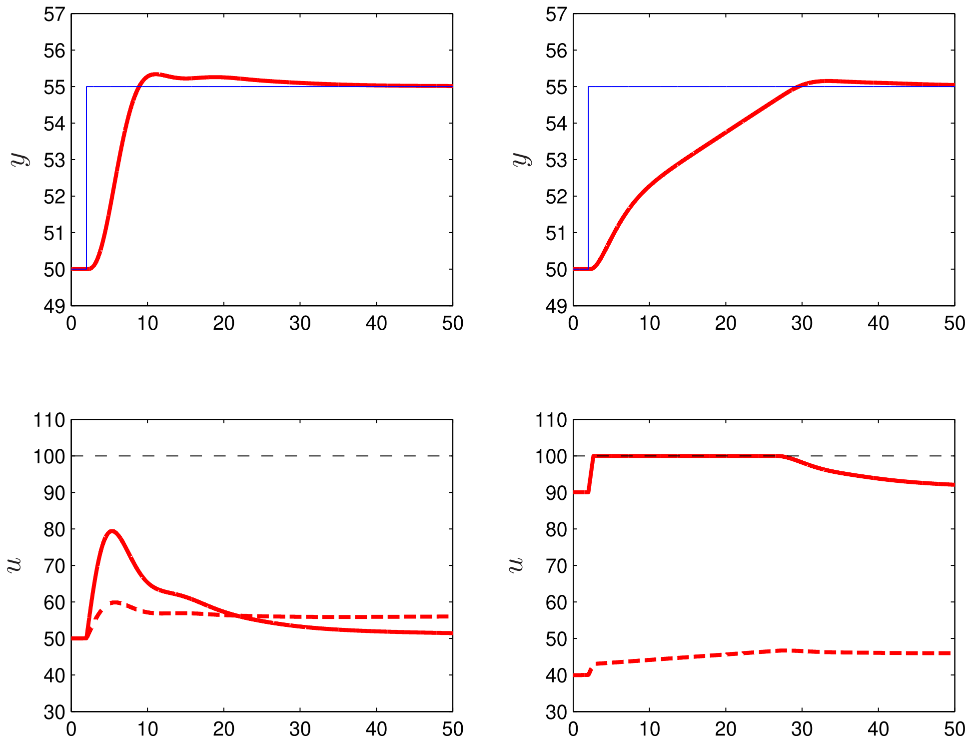

Fig. 6. Responses to a step change in setpoint y_{sp} (blue lines) using the VPC approach when control signal u_1 is not saturated (left) and when it is saturated (right). Control signal u_1 is solid and control signal u_2 is dashed. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

where $K_{p1} = 0.2$, $T_{p1} = 2$, $K_{p2} = 0.8$, and $T_{p2} = 10$. Process P_1 is thus faster with a lower gain than process P_2. These process models are motivated further in Section 5. Both controllers are PI controllers with parameters $K_1 = 8.35$, $T_{i1} = 2.68$, $K_2 = 0.30$, and $T_{i1} = 20$. The figure shows the responses to a step change in setpoint y_{sp}. The plots to the left show control when $u_{sp} = 50\%$, and the control works as expected. Note that control signal u_1 returns to its setpoint $u_{sp} = 50\%$ after the transient. The plots to the right show the case when $u_{sp} = 90\%$, illustrating e.g. Application 2. In this case, control signal u_1 becomes saturated, which leads to a very slow response in the VPC controller. The reason why the response is so slow is that the control error in the VPC controller is small, only 10%. So, the second drawback of the VPC controller is that the responses may become very slow at large setpoint changes or load disturbances when control signal u_1 becomes saturated.

A third drawback is that the flow controller C_1 cannot be switched to manual mode without interrupting the control. Even though the VPC controller C_2 manipulates the flow, it cannot control it since it does not take the flow signal as input. So, there is a configuration with redundancy since there are two manipulators that can be used to control the flow, but this redundancy cannot be utilized for e.g. valve maintenance.

2.2. Summary

To summarize, this section has demonstrated some severe drawbacks with the VPC approach. Since the VPC requires integral action, it is likely that stick–slip motion will occur when the controller is connected to a large valve, as in Application 1. This stick–slip motion will cause flow disturbances, which is undesirable when precise flow is desired. Another drawback is that the control may become sluggish at large setpoint changes or load disturbances, when controller C_1 becomes saturated. A final problem is that the flow controllers cannot be switched to manual mode with the flow still under control, since the VPC controllers are not able to control the flow.

Because of all these drawbacks, it is common that people in industry have developed their own solutions to the mid-ranging control problems. These are normally based on logic schemes that handles the adjustment of the large valve or pump in cases when the fast valve gets close to its saturation limits.

3. The feedforward mid-ranging control strategy

A block diagram of the new mid-ranging control strategy is presented in Fig. 7. As for the VPC approach, given in Fig. 5, process P_1 and controller C_1 form the fast and precise feedback loop. Controller C_2 is no longer a VPC, but also this controller takes y as input and y_{sp} as setpoint. Thus, both controllers act on the same signals. Controller C_1 is normally a PI controller, and it should be tuned in standard ways, in the same way as for the VPC approach. The controller can be extended to a PID controller or even more advanced controllers if desired, but in this paper the PI structure is kept.

Controller C_2 is a P controller. To avoid stick–slip motion, integral action is not introduced in this controller. Controller C_2 should be tuned conservatively, so that the fast loop is not disturbed too much. Therefore, there is normally no reason to add derivative action to this controller.

The mid-ranging of u_1 is obtained by adding a feedforward signal to C_2. For this reason, the approach in Fig. 7 is called Feedforward Mid-Ranging Control (FFMRC).

Feedforward signal u_3 is obtained in the following way. Control signal u_1 is first passed through a deadzone, where the user has to specify parameters u_{low} and u_{high}, that define an acceptable region for the stationary value of u_1, $u_{\text{low}} \le u_1 \le u_{\text{high}}$. The output of the deadzone, y_{dz} is given by

$$y_{dz} = \begin{cases} u_1 - u_{\text{high}} & u_1 > u_{\text{high}} \\ 0 & u_{\text{low}} \le u_1 \le u_{\text{high}} \\ u_1 - u_{\text{low}} & u_1 < u_{\text{low}} \end{cases}$$

This output is fed to a third controller, C_3, with setpoint equal to zero. The output from C_3, u_3, is the feedforward signal that is added to controller C_2. Controller C_3 is a PI controller, and as for controller C_2,

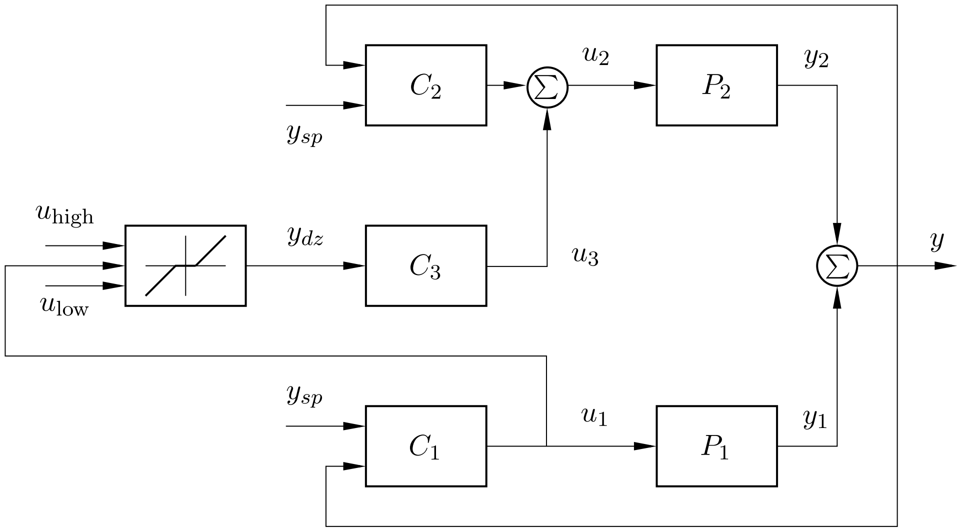

Fig. 7. The feedforward mid-ranging control strategy.

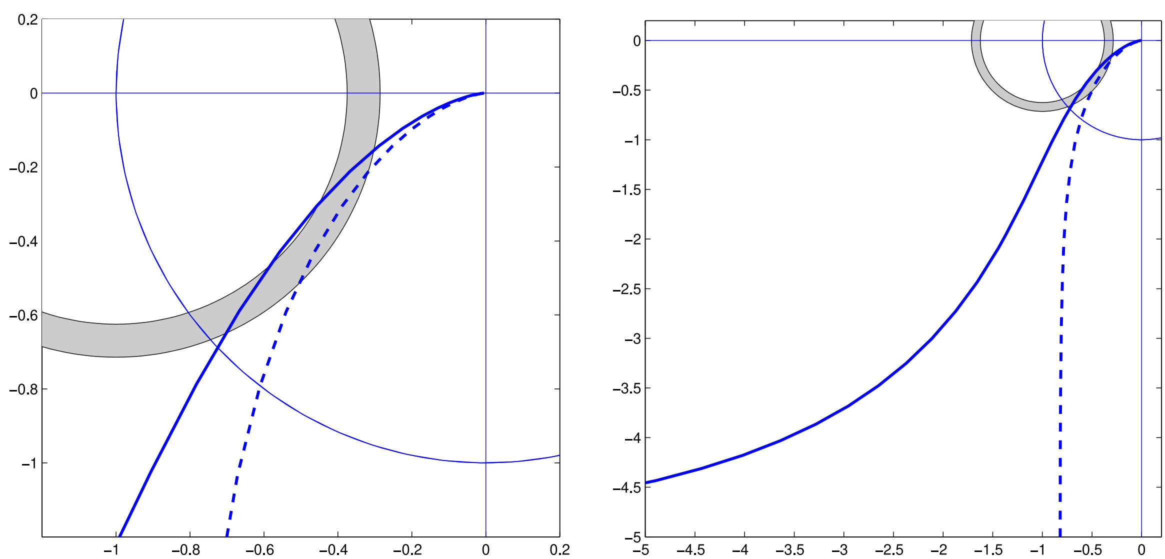

Fig. 8. Nyquist plots of loop transfer function L_1 (dashed line), and the loop transfer function of the FFMRC structure (solid line).

controller C_3 should be tuned conservatively so that the fast loop is not disturbed too much. Therefore, there is no reason for not keeping the PI structure in this controller.

The deadzone is introduced to avoid stick–slip motion caused by friction in the final control element of P_2. Without the deadzone, stick–slip motion would have occurred because of the integral action in C_3. For the same reason as for the VPC controller in the VPC approach, integral action is needed in C_3 to ensure that the stationary values of u_1 is kept under control. However, if there is no risk for stick–slip motion, the deadzone can be removed by setting $u_{low} = u_{high} = u_{sp}$.

The fact that u_1 is not guaranteed to reach a certain setpoint u_{sp} in steady state, but just to stay within a certain region specified by u_{low} and u_{high}, is normally not a problem. In most applications, it is sufficient to keep the stationary value of u_1 within such a region. For Application 1, these specifications can e.g. be $u_{low} = 40\%$ and $u_{high} = 60\%$. For Application 2, the specifications may e.g. be $u_{low} = 85\%$ and $u_{high} = 95\%$.

Already from the structure of the FFMRC it is seen that two of the drawbacks with the VPC approach have been removed. Since there is no integral action in C_2, there will not be any stick–slip motion as long as u_1 stays within the bounds $u_{low} \leq u_1 \leq u_{high}$.

Since both C_1 and C_2 have y_{sp} and y as inputs, one of them can be switched to manual mode and let the other one take over the control. If desired, controller C_2 can be switched to a PI controller using gain scheduling when controller C_1 is in manual mode.

Since C_2 acts on y and not u_1, the drawback that the control becomes sluggish at large setpoint changes or load disturbances because of the saturation of u_1, is also removed. This will be shown in Section 5.

3.1. Practical issues

Controllers have normally an output range between 0% and 100%, which means that the outputs only take positive values. The output of controller C_3 may, however, take both positive and negative values. It is recommended to choose the range of this controller to −50% to 50%, giving it the same size of its range as the other controllers. If this is not the case, the ranges of the different controllers must be taken into account when determining their gains.

There are also some operational aspects to take care of. If controller C_1 or controller C_2 is switched to manual mode, controller C_3 must also be switched to manual mode to avoid integrator windup in this controller. Controller C_3 can, however, be switched to manual mode with the other controllers running in automatic mode.

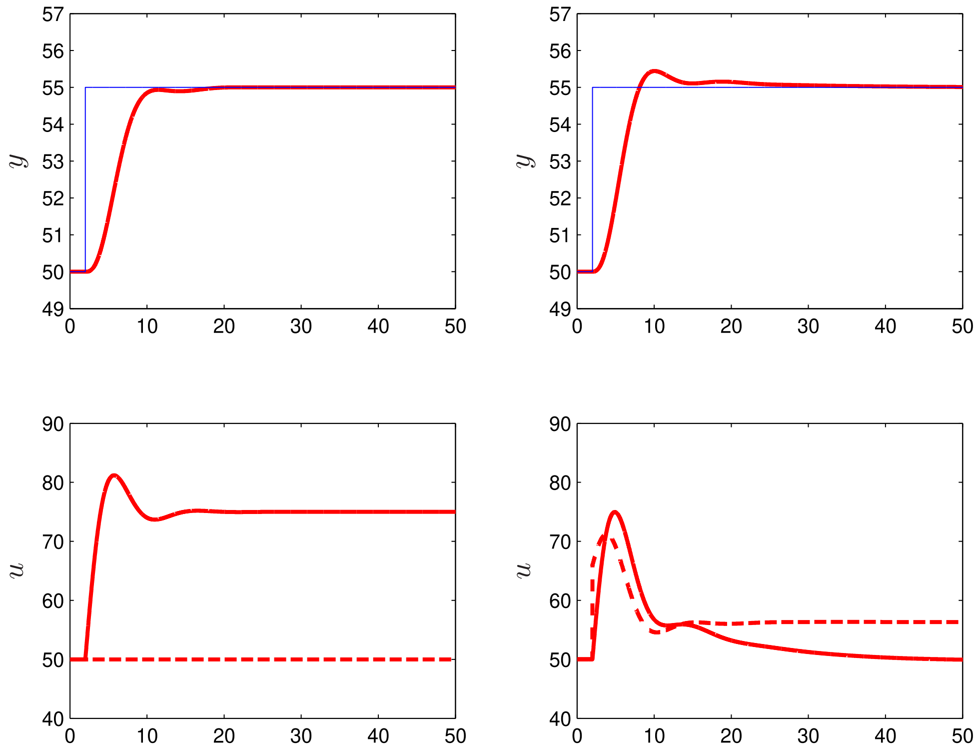

Fig. 9. Responses to a step change in setpoint y_{sp} (blue lines) when only controller C_1 is active (left plots), and for the FFMRC approach (right plots). Control signal u_1 is solid and control signal u_2 is dashed. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

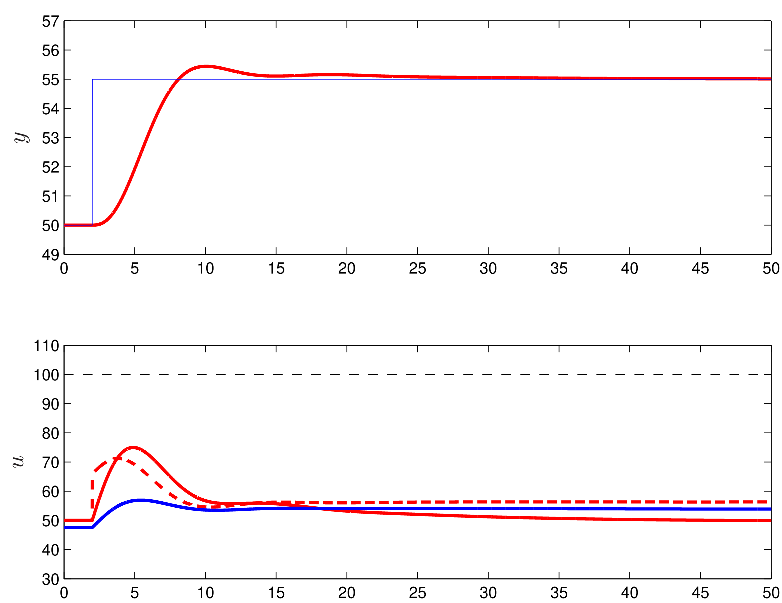

Fig. 10. Responses to a step change in the setpoint using the FFMRC approach. The desired bounds of u_1 are $u_{\text{high}} = u_{\text{low}} = 50\%$. The upper plot shows y_{sp} (blue line) and process output y (red line). In the lower plot, the solid red line is u_1, the dashed line is u_2, and the solid blue line is u_3. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

4. Design of the FFMRC

The FFMRC consists of three controllers, and design procedures for these controllers are given in this section. The three controllers are

$$C_1 = K_1\left(1 + \frac{1}{sT_{i1}}\right) \qquad C_2 = K_2 \qquad C_3 = K_3\left(1 + \frac{1}{sT_{i3}}\right)$$

The basic idea is to tune C_1 using standard procedures like the Lambda method, see Dahlin (1968) and Higham (1968), the SIMC method, see Skogestad (2003), or the AMIGO method, see Åström and Hägglund (2006). The other two controllers are then tuned conservatively in such a way that the fast loop with controller C_1 is not disturbed too much. The closed-loop dynamics of the fast loop should be about the same when C_2 and C_3 are in manual mode and automatic mode, respectively.

From Fig. 7, the following relation between process output y and its corresponding setpoint y_{sp} is

$$Y = P_1C_1\left(Y_{sp} - Y\right) + P_2C_2\left(Y_{sp} - Y\right) + P_2C_3C_1\left(Y_{sp} - Y\right)$$

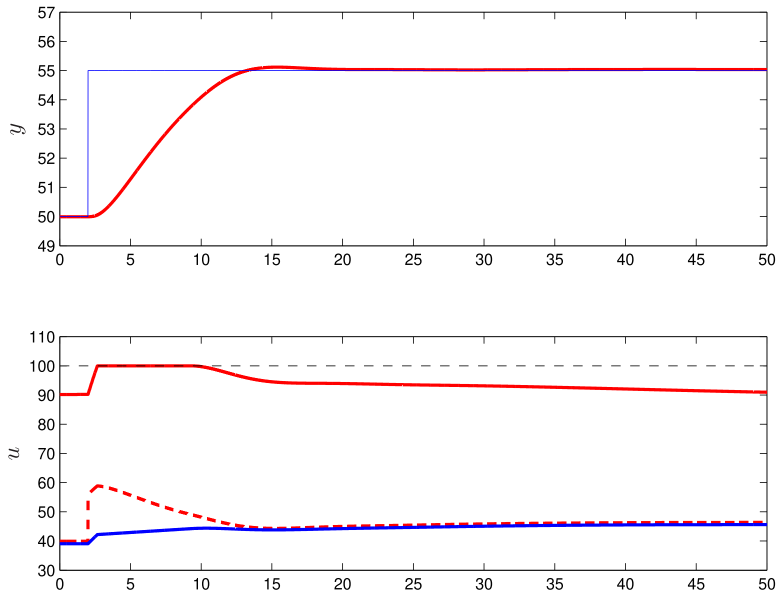

Fig. 11. Responses to a step change in the setpoint using the FFMRC approach. The desired bounds of u_1 are $u_{\text{high}} = u_{\text{low}} = 90\%$. The upper plot shows y_{sp} (blue line) and process output y (red line). In the lower plot, the solid red line is u_1, the dashed line is u_2, and the solid blue line is u_3. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

The deadzone can be neglected, since the interesting case is when all three controllers are active, i.e. when u_1 is outside the deadzone. The closed-loop transfer function is therefore given by

$$Y = \frac{P_1C_1 + P_2C_2 + P_2C_3C_1}{1 + P_1C_1 + P_2C_2 + P_2C_3C_1} Y_{sp}$$

This means that the system can be seen as a simple feedback loop with loop transfer function

$$L = P_1C_1 + P_2C_2 + P_2C_3C_1$$

which means that loop-shaping design methods can be used. The goal is that this loop transfer function should be approximately equal to the loop transfer function of the fast loop, i.e.

$$L = P_1C_1 + P_2C_2 + P_2C_3C_1 \approx L_1 = P_1C_1$$

This means that $|P_2C_2|$ and $|P_2C_3C_1|$ should be small compared to $|P_1C_1|$ in the important frequency region where the robustness properties are determined.

The following design procedure is proposed. It must be used with care, since it is based on considerations at one frequency only. This is commented on later in the section.

1. Tune C_1 using some standard tuning procedure.
2. Determine the crossover frequency of L_1, i.e. frequency ω_1 where $|L_1(i\omega_1)| = 1$.
3. Determine gain K_2 in C_2 so that $|P_2(i\omega_1)C_2(i\omega_1)| \leq \gamma$.
4. Determine integral time T_{i3} in C_3 as N/ω_1.
5. Determine gain K_3 in C_3 so that $|P_2(i\omega_1)C_3(i\omega_1)C_1(i\omega_1)| \leq \gamma$.

Parameter γ should be chosen so small that the robustness properties of the fast loop are not changed too much when it is complemented by the mid-ranging strategy. Choosing $\gamma = 0.1$ guarantees that each of the two controllers C_2 and C_3 will not change the phase margin more than around 5°, which means that the two controllers together may change it by at most 10°.

Parameter N should be chosen so that the corner frequency of the PI controller is not too close to the crossover frequency ω_1, and $N = 5$ is found to be a reasonable choice in many cases.

4.1. Example

The following example illustrates the design procedure. Let the two process transfer functions be the same as in Section 2, i.e.

$$P_1 = \frac{K_{p1}}{(1 + sT_{p1})^2} \qquad P_2 = \frac{K_{p2}}{(1 + sT_{p2})^2}$$

where $K_{p1} = 0.2$, $T_{p1} = 2$, $K_{p2} = 0.8$, and $T_{p2} = 10$. Process P_1 is thus faster with a lower gain than process P_2.

Controller C_1 is first tuned using the AMIGO tuning rule, see Åström and Hägglund (2006). The parameters become $K_1 = 8.35$ and $T_{i1} = 2.68$. The crossover frequency of loop transfer function L_1 becomes $\omega_1 = 0.5$.

The next step is to determine gain K_2. It is determined from

$$|P_2(i\omega_1)C_2(i\omega_1)| = \frac{K_{p2}K_2}{1 + \omega_1^2 T_{p2}^2} = \frac{0.8K_2}{1 + 0.25 \cdot 100} \approx 0.031K_2 = \gamma$$

With $\gamma = 0.1$ this gives $K_2 = 3.2$.

Using N = 5, the integral time of controller C_3 is given by

$$T_{i3} = \frac{N}{\omega_1} = \frac{5}{0.5} = 10$$

Finally, gain K_3 of controller C_3 is obtained from

$$|P_2(i\omega_1)C_3(i\omega_1)C_1(i\omega_1)| = \frac{K_{p2}}{1 + \omega_1^2 T_{p2}^2} \cdot \frac{K_3\sqrt{1 + \omega_1^2 T_{i3}^2}}{\omega_1 T_{i3}} \cdot \frac{K_1\sqrt{1 + \omega_1^2 T_{i1}^2}}{\omega_1 T_{i1}} = \gamma$$

With $\gamma = 0.1$ and using the numerical values for the process parameters and controller parameters obtained so far, this gives $K_3 = 0.31$.

Fig. 8 shows the Nyquist plots of loop transfer function L_1 of the fast loop and loop transfer function L of the FFMRC structure, respectively, in two different scales. The mid-ranging decreases the phase margin from 52° to 44°. The shaded areas in the figures mark the regions where $1.4 \leq M_s \leq 1.6$. The design objective of the AMIGO design method is to obtain $M_s = 1.4$. From the figure it is noted that the mid-ranging increases the M_s value from about 1.5 to 1.6.

The role of integral time T_{i3} is not visible in the left plot of Fig. 8. The figure would have looked the same if controller C_3 were a pure P controller, $C_3 = K_3$. This means that with the proposed choice of integral time T_{i3}, the robustness degradation is determined by controller gains K_2 and K_3 only.

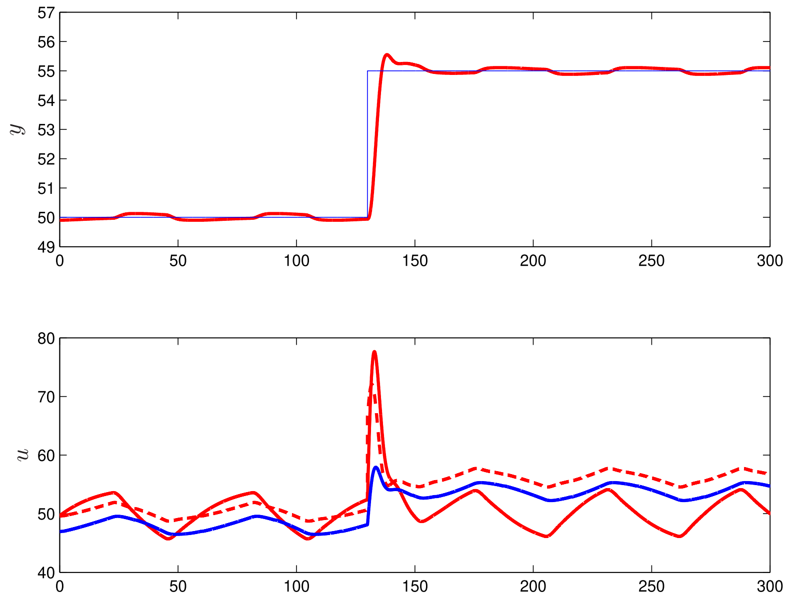

Fig. 12. Responses to a step change in the setpoint using the FFMRC approach when there is stiction in the valve in process P_2. The desired bounds of u_1 are $u_{high} = u_{low} = 50\%$. The upper plot shows y_{sp} (blue line) and process output y (red line). In the lower plot, the solid red line is u_1, the dashed line is u_2, and the solid blue line is u_3. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

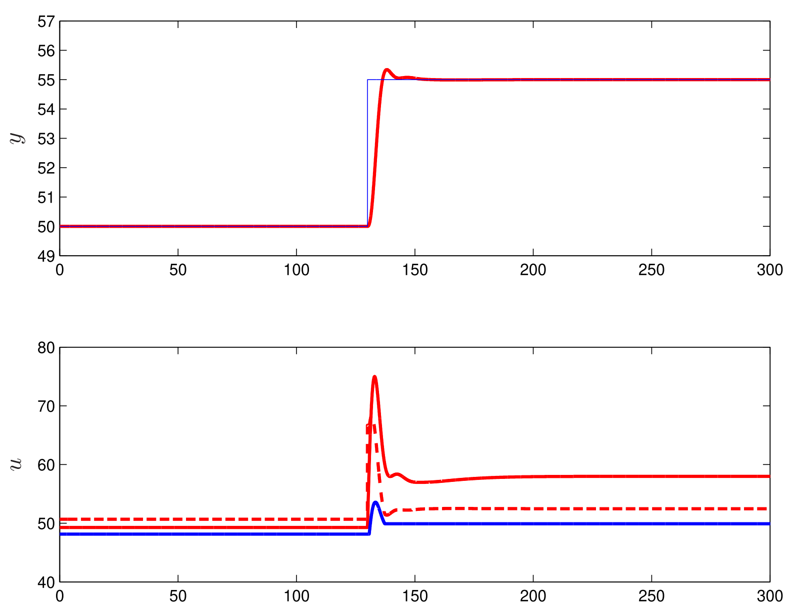

Fig. 13. Responses to a step change in the setpoint using the FFMRC approach when there is stiction in the valve in process P_2. The desired bounds of u_1 are $u_{low} = 40\%$ and $u_{high} = 60\%$. The upper plot shows y_{sp} (blue line) and process output y (red line). In the lower plot, the solid red line is u_1, the dashed line is u_2, and the solid blue line is u_3. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

However, since $P_2C_3C_1$ has two integrators, this term in the loop transfer function will dominate at frequencies lower than those presented in the left plot of Fig. 8. This effect is seen in the right plot of the figure. Therefore, it is crucial to choose T_i large enough to avoid robustness problems caused by the integral term of controller C_3.

Fig. 9 shows responses to a step change in setpoint y_{sp} for the simple feedback loop obtained when controllers C_2 and C_3 are in manual mode (left), and when the full FFMRC is active (right), respectively. The figure shows that the responses in y are similar in the two cases. Note also that control signal u_2 is constant when C_2 is in manual mode, whereas it drives control signal u_1 back to 50% when the FFMRC is active. This will be discussed further in the next section.

The example shows that the design procedure works as desired, i.e. that the introduction of the FFMRC structure does not influence the robustness properties of the fast feedback loop too much compared to the properties given when the FFMRC is disconnected. However, it should be stressed that the design procedure must be used with care, since the loop shaping is based on properties at frequency ω_1 only. One must ensure that integral time T_{i3} is large enough so that the low-frequency part of the term $P_2C_3C_1$ does not cause robustness problems.

4.2. A simple design procedure

The design procedure presented above may be too time consuming and complicated for industrial use in process industry. Therefore, it is desired to have simpler tuning procedures to find the three parameters of controllers C_2 and C_3. Inspired by the design procedure presented above, the following simple design procedure is suggested.

1. Design controller C_1 in traditional ways, with controllers C_2 and C_3 in manual mode.
2. With controller C_1 in automatic mode and C_3 still in manual mode, adjust gain K_2 of controller C_2 so that the fast loop is only slightly influenced.
3. Choose integral time T_{i3} of controller C_3 as $M \cdot T_{i1}$ where M is around 5 to 10.
4. With controllers C_1 and C_2 in automatic mode, adjust gain K_3 of controller C_3 so that the fast loop is only slightly influenced.

4.2.1. Initial tuning parameters

In the simple design procedure, it is suggested to tune gains K_2 and K_3 manually. Reasonable and conservative starting points for this tuning are presented in this section.

Gain K_2 should be determined so that $|P_2(i\omega_1)C_2(i\omega_1)| \leq \gamma$. Assuming that $|P_2(i\omega_1)| \leq |P_2(0)| = K_{p2}$, a conservative choice is given by $K_{p2}K_2 = \gamma$, i.e. $K_2 = \gamma/K_{p2} = 0.1/K_{p2}$. The choice is often too conservative. Another possibility is to use the recommendation in the One-Third Rule for PI controller tuning, namely $K_2 = 1/(3K_{p2})$, see Hägglund (2019), or the more aggressive choice $K_2 = 1/K_{p2}$.

Gain K_3 should be determined so that $|P_2(i\omega_1)C_3(i\omega_1)C_1(i\omega_1)| \leq \gamma$. Using the conservative approximation above, $|P_2(i\omega_1)| \leq K_{p2} = \gamma/K_2$. For the frequency ω_1, we can also make the approximations $|C_1(i\omega_1)| \approx K_1$ and $|C_3(i\omega_1)| \approx K_3$. This gives

$$|P_2(i\omega_1)C_3(i\omega_1)C_1(i\omega_1)| \leq \frac{\gamma K_3 K_1}{K_2} = \gamma$$

which gives the initial value $K_3 = K_2/K_1$.

4.3. Design of the VPC controller

As a by product of the analysis presented in this section, a new design method for the VPC controller can easily be obtained. From Fig. 5, the following relation between process output y and its corresponding setpoint y_{sp} in the VPC structure is

$$Y = P_1C_1\left(Y_{sp} - Y\right) + P_2C_2C_1\left(Y_{sp} - Y\right)$$

The closed-loop transfer function is therefore given by

$$Y = \frac{P_1C_1 + P_2C_2C_1}{1 + P_1C_1 + P_2C_2C_1} Y_{sp}$$

This means that the system can be seen as a simple feedback loop with loop transfer function

$$L = P_1C_1 + P_2C_2C_1$$

The same design approach as used for the FFMRC can now be applied, i.e.,

1. Tune C_1 using some standard tuning procedure.
2. Determine the crossover frequency ω_1 of $L_1 = P_1C_1$.
3. Determine integral time T_{i2} in C_2 as N/ω_1.
4. Determine gain K_2 in C_2 so that $|P_2(i\omega_1)C_2(i\omega_1)C_1(i\omega_1)| \leq \gamma$.

where N and γ are chosen in the same way as for the FFMRC structure.

The simple design procedure for the FFMRC can of course also be applied to the VPC controller.

5. Simulation examples

This section presents some simulation results from the use of the FFMRC approach. The same process models and controllers as presented previously will be used, i.e.

$$P_1 = \frac{K_{p1}}{(1 + sT_{p1})^2} \qquad P_2 = \frac{K_{p2}}{(1 + sT_{p2})^2}$$

where $K_{p1} = 0.2$, $T_{p1} = 2$, $K_{p2} = 0.8$, and $T_{p2} = 10$. These two process models are realistic models of flow control processes. The dynamics of a flow control system is composed of the dynamics of the flow sensor, the dynamics of the valve or the bump, and the liquid dynamics. The liquid dynamics is normally very fast and can be neglected. The main dynamics comes from the actuator and the sensor, including filters. According to Shinskey (1996), these models are normally multi-capacity models as P_1 and P_2.

The controllers are

$$C_1 = K_1\left(1 + \frac{1}{sT_{i1}}\right), \qquad C_2 = K_2, \qquad C_3 = K_3\left(1 + \frac{1}{sT_{i3}}\right)$$

where $K_1 = 8.35$, $T_{i1} = 2.68$, $K_2 = 3.2$, $K_3 = 0.31$, and $T_{i3} = 10$.

Fig. 10 shows the result of a setpoint change when the desired bounds of u_1 are $u_{\text{high}} = u_{\text{low}} = 50\%$. The experiment is the same as the one shown for the VPC approach in the left part of Fig. 6. A comparison between these two figures shows that the VPC and FFMRC give similar results in this case.

Fig. 11 shows the same experiment as Fig. 10 except for the desired bounds of u_1 which are changed to $u_{\text{high}} = u_{\text{low}} = 90\%$, corresponding to e.g. Application 2. The result can be compared with the similar experiment for the VPC presented on the right part of Fig. 6. The difference between the two approaches is great. In the VPC approach, controller C_2 takes u_{sp} and u_1 as inputs, and since u_1 is saturated at 100%, the control becomes very sluggish. In the FFMRC approach, controller C_2 takes y_{sp} and y as inputs, which results in a much faster response to the setpoint change. It is interesting to compare control signal u_2 in the two cases.

The following simulations are performed with simulated stiction in a valve corresponding to process P_2. Control signal u_2 is the input to the stiction function, and the output, u_s, is the signal that enters in P_2. The stiction function is taken from Hägglund (2011) and is given by

$$u_s(k) = \begin{cases} u_s(k-1) & |u_2(k) - u_s(k-1)| \leq d \\ u_2(k) & \text{otherwise} \end{cases} \tag{2}$$

where d is the valve-stiction band. The model assumes that the valve is stuck as long as the magnitude of the difference between controller output u_2 and valve output u_s is within the band d. When the difference becomes larger than d, the valve output slips to the desired output, i.e. the controller output u_2. There are many other, more advanced, stiction models presented in the literature. See e.g. the survey given in Jelali and Huang (2010). However, the model (2) is good enough to illustrate the stiction phenomena in this paper.

Fig. 12 shows the result of a setpoint change when the desired bounds of u_1 are $u_{\text{high}} = u_{\text{low}} = 50\%$, and where the stiction band is $d = 3\%$. Because of the stiction and the fact that controller C_3 has integral action, stick–slip motion occurs and the whole system is oscillating. A similar behavior would have been obtained for the VPC approach.

In Fig. 13, the deadzone in the FFMRC structure is introduced by selecting the bounds for u_1 to $u_{\text{low}} = 40\%$ and $u_{\text{high}} = 60\%$. Because of the deadzone, the integral action of controller C_3 is not active when u_1 is inside the deadzone, and the oscillations disappear.

Fig. 14 illustrates how the FFMRC approach works when load disturbances and measurement noise are present. The figure shows a step change in setpoint followed by a step load disturbance, corresponding to e.g. a sudden pressure drop in the tube. White noise measurement noise is also added to the process output. The figure shows that the FFMRC approach works as desired also in the presence of disturbances.

6. Conclusions

This paper has pointed out some drawbacks with the valve position control (VPC) approach to solve mid-ranging control problems. These drawbacks are that stiction in valves may result in stick–slip motion that makes the VPC approach unsuitable, that sluggish control may occur when the small manipulator becomes saturated, and that it is impossible to keep the control in automatic mode when one of the controllers is switched to manual mode.

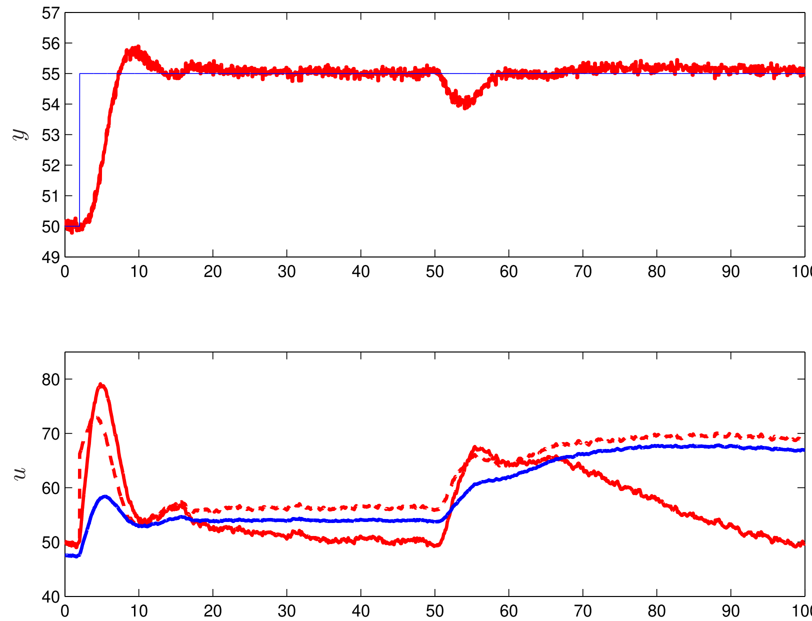

Fig. 14. Responses to a step change in the setpoint followed by a load disturbance response at time $t = 50$ using the FFMRC approach. White noise is added to the process output. The desired bounds of u_1 are $u_{\text{low}} = u_{\text{high}} = 50\%$. The upper plot shows y_{sp} (blue line) and process output y (red line). In the lower plot, the solid red line is u_1, the dashed line is u_2, and the solid blue line is u_3. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

A new mid-ranging control approach, the feedforward mid-ranging control approach (FFMRC) is proposed in this paper. It is shown that the FFMRC overcomes the problems associated with the VPC approach. Design methods and simple tuning rules for the FFMRC are provided. The design methods are extended to the VPC as well.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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