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2000

ISSN 0280–5316
ISRN LUTFD2/TFRT--7589--SE

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February 2000

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Document name
INTERNAL REPORT

Date of issue
February 2000

Document Number
ISRN LUTFD2/TFRT--7589--SE

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Sponsoring organisation

Title and subtitle
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In this paper, an active control system is designed to handle processes with undamped modes. A modular approach has been taken, where an active control system has been designed, which consists of an allpass filter and a bandpass filter. To determine the parameters of these two filters the only information needed is a few characteristics of the process frequency response.

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Classification system and/ or index terms (if any)

Supplementary bibliographical information

ISSN and key title	*ISBN*
0280–5316	

Language	*Number of pages*	*Recipient's notes*
English	10	
Security classification		

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Abstract In this paper, an active control system is designed to handle processes with undamped modes. A modular approach has been taken, where an active control system has been designed, which consists of an allpass filter and a bandpass filter. To determine the parameters of these two filters the only information needed is a few characteristics of the process frequency response.

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Introduction

Undamped systems occur in many areas, for example in mechatronics, in industrial robots, in drive systems as steel mills, in HVDC, in combustion control, in cranes, in automotive systems, and in ship movements. They are classified into two cases: those who are intentionally made to contain undamped modes and those who are not. For example, cranes are intentionally made to have undamped modes on the contrary to motors which are affected by vibrations. Furthermore, todays' trends show that future machines will be faster, lighter, have softer materials and be more flexible. This implies that future machines will be unintentionally made to contain undamped modes. Examples are industrial robots and vehicles.

The standard PID controller have many advantages, but there are cases when it does not suffice to meet the required control performance. For example, when the process contains long time delays, undamped modes, and nonlinearities. There are two ways to improve the performance of the closed loop system, by either using a modular/synthetic approach or a unimodular/analytic one. For example, in the case of processes with long time delays a modular approach is taken when a Smith predictor is inserted into the loop to improve the performance of the closed loop system. In this paper the former approach is used to improve the performance of the closed loop system for processes with undamped modes.

The control structure proposed in this paper has two degrees of freedom. A controller $C(s)$ stabilizes the plant, and an active control system F rejects harmonic disturbances. In the conventional method the controller $C(s)$ must simultaneously play both roles, leading to performance limitations.

Background

Due to physical, economical and safety constraints the oscillation and vibration control have become a prime consideration in the general manufacturing industry. For example, a ships roll motion due to the waves is a highly resonant system. It is then, necessary for ships carrying passengers or weapon platforms to control this undesirable motion. Another example is the use of cranes to move containers. In this case it is required that the payload does not undergo excessive swing, if not damage to personnel and cargo may occur. Moreover, in the automotive industry recent trends are towards smaller and lighter vehicles for better fuel economy and market expansion, see Karkosch and Svaricek (1999). At the same time, there has been a major increase in consumer awareness of long term health impacts of exposure to high noise and vibration levels.

Due to these physical, economical and safety constraints, conventional techniques such as passive sound absorptive materials, mass tuned dampers, modification of the system design, etc., by themselves would not satisfy the requirements of producing a minimum weight system with an optimal fuel economy and in those cases needed a high comfort level. Consider for example, an engine mount; in order to limit engine movement, it is desired to have a very stiff mount. However, to minimize transmission of engine vibrations into the passenger compartment, a very soft engine mount is required. These contradictory requirements have made engine mount manufactures to design passive vibration absorbers that provide an optimal compromise. But, passive noise and vibration treatments at low frequencies are physically not realizable. Consequently, active control technologies have been investigated for low frequency noise and vibration control. Consequently, the complete solution to the noise and vibration problems can be obtained by integration of passive and active systems.

The problem of active control of noise and vibrations has been a subject of much research in recent years, see Karkosch and Svaricek (1999) and references there in. The main part of the published literature makes use of adaptive signal processing and filtering techniques. According to Ohmori *et al.* (1999), the most popular techniques for harmonic disturbance rejection are (i) the feedback controller design method based on internal model principle; and (ii) the feed forward controller design method based on external model principle.

In the feedback control structure of (i) approach the requirement is

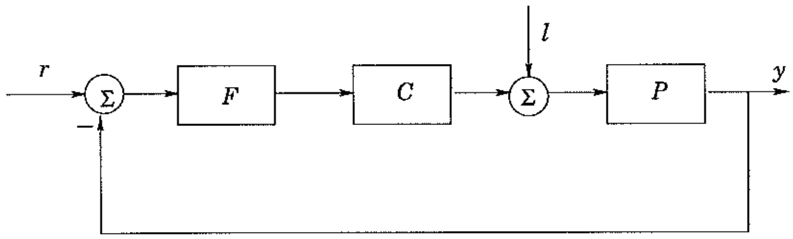

Figure 1 The use of a notch filter F to handle undesired excitation of undamped modes of the process P.

an asymptotic disturbance rejection. This can be realized by the insertion of notch filters. The advantages of this type of algorithm are that it is linear making analysis easier, and that convergence is very rapid. The disadvantages are that there are some performance limitations because the algorithm must satisfy both closed loop stability and disturbance rejection.

On the other hand, in external model controllers of (ii) approach, the disturbance model is placed outside the basic feedback loop. The disturbance model is adjusted adaptively to match the actual disturbance. The advantage of this approach is that compensation is like feed forward, then the effect on the nominal open loop gain can be made small. The disadvantages are that the analysis and implementation are somewhat more complex than for the internal model based algorithms.

Notch Filtering Limitations

A classical method to avoid unnecessary excitation of undamped modes is illustrated in Figure 1. The process P which consists of undamped modes is controlled by a controller C. To avoid unnecessary excitation of the undamped modes in P a notch filter F is introduced. Unfortunately, the notch filter F will not provide any additional damping of these modes, when the disturbance l in Figure 1 excites them.The paper will present a new method to overcome this problem.

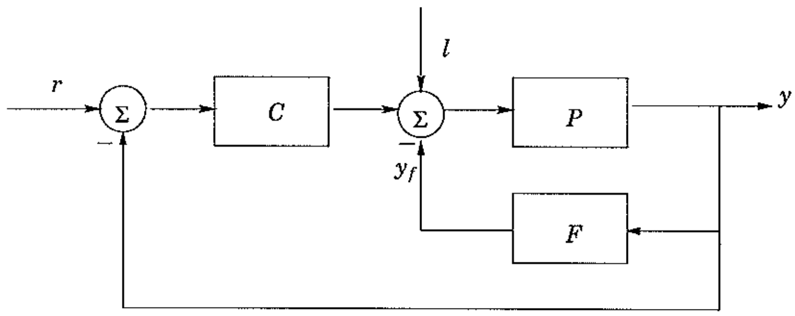

Figure 2 Control with active damping. In the inner feedback loop the active control system F rejects undamped modes of the process P. In the outer control loop the controller C gives good disturbance rejection, robustness to model uncertainties, and set point following.

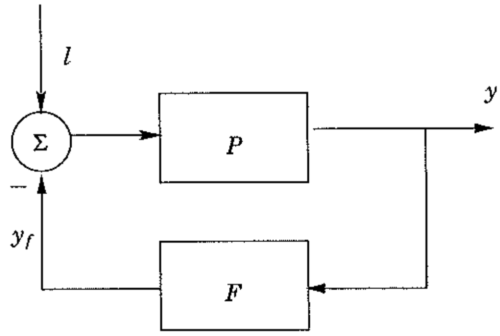

Figure 3 The reduced block diagram, describing the relation from load disturbance l to measurement output y.

The New Approach

Consider the two degree of freedom controller in Figure 2. The controller consists of an inner feedback loop where the purpose of the active control system F is to increase the damping of the plant. In the outer control loop the purpose of controller C is to achieve good disturbance rejection, robustness to model uncertainties and set point following. There are no requirements on the controller structure and design. The objective of the active control system F is to damp the oscillations in the output y, when the disturbance l excites the undamped modes of the process P. Thus, the design problem of F given in Figure 2 is reduced to the block diagram in Figure 3.

Assume that the disturbance l is a sine wave whose frequency corresponds to the one of the undamped modes of the process P. The purpose of F is then to synthesize a waveform y_f which is identical in magnitude, phase and frequency to the original signal l. This is realized if the filter F is the product of an allpass filter F_a with transfer function,

$$F_a(s) = \frac{s^2 - 2\zeta_a\omega_a s + \omega_a^2}{s^2 + 2\zeta_a\omega_a s + \omega_a^2}, \tag{1}$$

and a bandpass filter F_b with the transfer function,

$$F_b(s) = K_b \frac{s}{s^2 + 2\zeta_b\omega_b s + \omega_b^2}. \tag{2}$$

The allpass filter F_a makes it possible to obtain the right amount of phase lag of the output y at a specific frequency without affecting the gain, because, F_a gives only a phase and no gain contribution for all frequencies, compare with the left figure of Figure 4.

The bandpass filter F_b makes it possible to obtain the right amplitude of the output y at a specific frequency without affecting the phase, because, F_b gives only a gain but no phase contribution at a specific frequency, compare with the right figure of Figure 4. A good overview of different filters is found in Wie and Byun (1989).

In the next section it is shown how to determine the parameters of the allpass filter F_a: ζ_a, ω_a, and of the bandpass filter F_b: K_b, ζ_b, ω_b based on a few characteristics of the process frequency response.

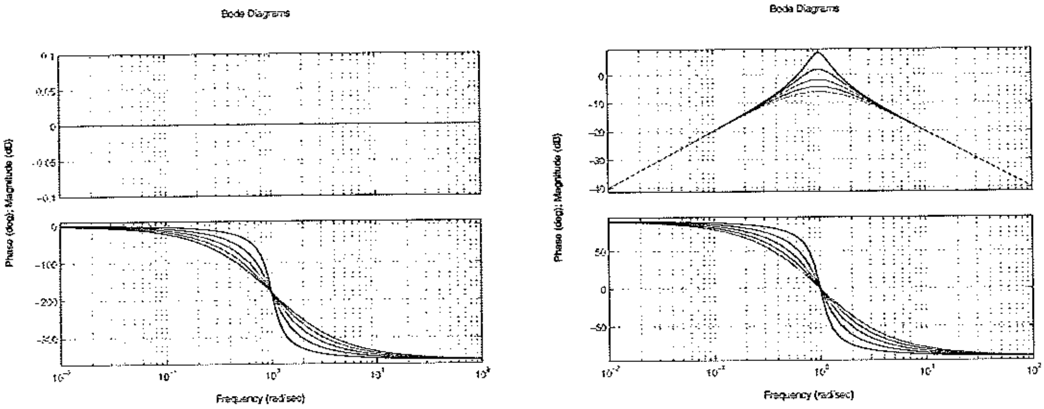

Figure 4 The Bode diagram of the allpass filter F_a (left) with $\omega_a = 1$,$\zeta_a = 0.2, 0.4, 0.6, 0.8$, and the bandpass filter F_b (right) with $\omega_b = 1$, $K_b = 1$, $\zeta_b = 0.2, 0.4, 0.6, 0.8$.

Design of the Active Control System F

For the design of the active control system F it is assumed that the undamped process P has the same structure as the flexible shaft system. This is no severe restriction, since a lot of undamped systems occurring in different areas may be modeled as a flexible shaft system. The system consists of two flywheels which are coupled with a spring, where the spring coefficient is assumed to be constant. It is driven by a torque input on the driving fly wheel, and the speed of the load flywheel is measured. The transfer function of the system is given by,

$$P(s) = \frac{\omega_p^2}{(s+\omega_f)(s^2 + 2\zeta_p\omega_p s + \omega_p^2)}, \tag{3}$$

where ζ_p and ω_p are the damping and the resonance frequency of the undamped modes of P.

The active control system F is the product of the allpass filter F_a in (1), and the bandpass filter F_b in (2). To determine the parameters of the two filters the following characteristics of the process frequency response $P(i\omega)$ is needed: the resonance frequency ω_p, and the amplitudes P_{max} and P_{min} defined in Figure 5.

The Allpass Filter Design: The purpose of the allpass filter F_a is to shift the phase of the output y at the resonance frequency ω_p such that,

$$\arg\left\{P(i\omega_p)\right\} + \arg\left\{F_a(i\omega_p)\right\} = -360°,$$

which is equal to

$$\arg\left\{\frac{s^2 - 2\zeta_a\omega_a s + \omega_a^2}{s^2 + 2\zeta_a\omega_a s + \omega_a^2}\right\}_{|s=i\omega_p} = -\arg\left\{P(i\omega_p)\right\} - 360°, \tag{4}$$

where the damping coefficient ζ_a determines the rate of decrease of $\arg F_a(i\omega)$, compare with the left figure of Figure 4. It was noted from simulations that the

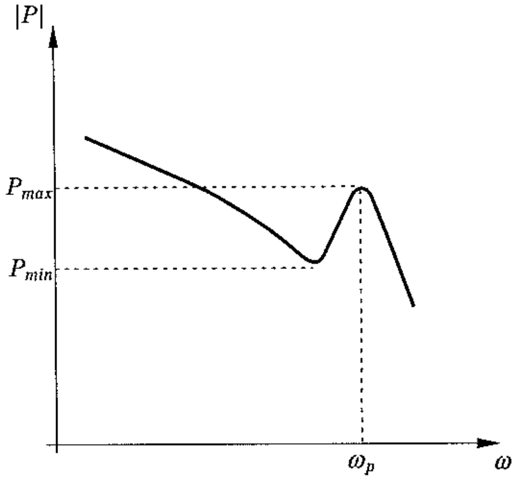

Figure 5 Characteristics of the process frequency response $P(i\omega)$ needed to design the active control system F.

choice of ζ_a is quite insensitive to performance. The choice of $\zeta_a = 0.5$ was made based on Wie and Byun (1989).

If Equation (4) is solved for ω_a, with $\zeta_a = 0.5$ the filter F_a will give the desired phase lag at ω_p.

The Bandpass Filter Design: The purpose of the bandpass filter is to change the amplitude of the output y at the frequency ω_p, such that $|P(i\omega_p)| \cdot |F_b(i\omega_p)| = 1$ at $\omega_b = \omega_p$, that is,

$$|P(i\omega_p)| \cdot \left| K_b \frac{s}{s^2 + 2\zeta_b \omega_p s + \omega_p^2} \right|_{s=i\omega_p} = 1, \tag{5}$$

where Equation (2) have been used. If Equation (5) is solved for K_b, then

$$K_b = \frac{2\zeta_b \omega_p}{|P_{max}|}, \tag{6}$$

gives the right amount of gain reduction at $\omega = \omega_p$.

In the right figure of Figure 4 it is illustrated how the damping coefficient ζ_b affects the amplitude and phase curve of $F_b(i\omega)$. In this case, the effect of ζ_b on the amplitude curve is the interesting one. A small value will give a narrow width of the notch gain at ω_p compared to larger values of ζ_b. It was noted from the simulations that best performance was obtained by using relative large values of ζ_b. The choice $\zeta_b = 0.9$ gave a desirable performance.

The Characteristics of the Resulting Design To analyze the effects of the determined active control system F, consider the transfer function

$$\frac{Y(s)}{L(s)} = \frac{P(s)}{1 + P(s)F(s)},$$

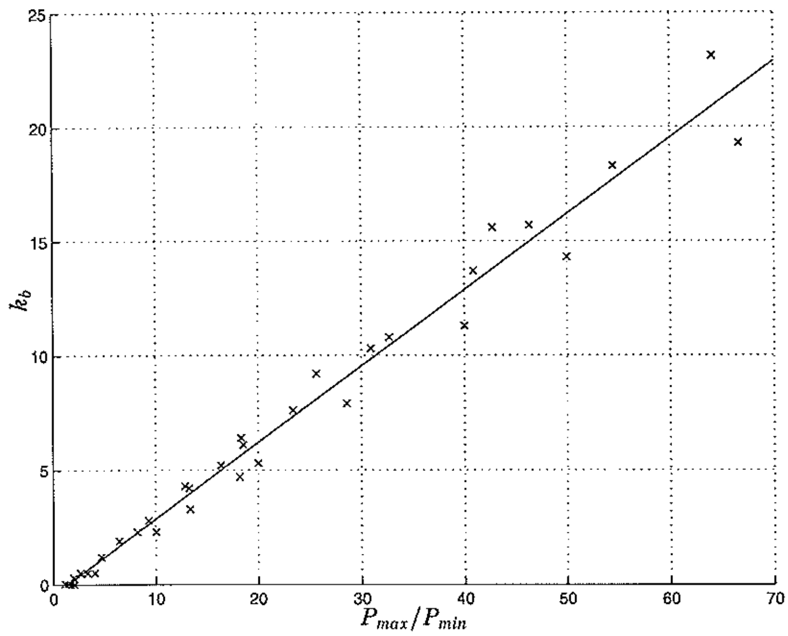

Figure 6 The gain k_b of the bandpass filter F_b plotted as a function of the characteristics P_{max}/P_{min} of the process frequency function $P(i\omega)$.

given by Figure 3. A performance measure of the design of the active filter F is its damping at the resonance frequency ω_p, that is,

$$\left|\frac{P(i\omega_p)}{1+P(i\omega_p)F(i\omega_p)}\right|. \tag{7}$$

To estimate the magnitude of Equation (7), the following results are needed: $|P(i\omega_p)| = P_{max}$ and $|F_a(i\omega_p)| = 1$. Furthermore, from Equation (1) and (6) it follows that $|F_b(i\omega_p)| = K_b/2\zeta_b = 1/P_{max}$. Then, Equation (7) gives,

$$\left|\frac{P(i\omega_p)}{1+P(i\omega_p)F(i\omega_p)}\right| \geq \frac{|P(i\omega_p)|}{1+|P(i\omega_p)|\,|F(i\omega_p)|} = \frac{P_{max}}{2}, \tag{8}$$

where the triangle inequality has been used. Consequently, the filter F will reduce the gain at the most with half the magnitude of the process P at the resonance frequency ω_p.

The performance of the designed active filter F was evaluated on a test batch for the process (3) with $\omega_f = 1$, and different values of ω_p and ζ_p. As a performance measure the integrated absolute error IAE, defined in Åström and Hägglund (1995), was computed for $e = r - y$ with $r = 0$. The calculated IAE for the different processes in the test batch showed that in some cases smaller values could be obtained. In other words, the use of K_b in Equation (6) will not give the optimal gain reduction at ω_p for all processes in the test batch. For some processes K_b should be larger for other ones it should be smaller. Consequently, the performance of the active filter F would improve if its gain K_b depends on the process characteristic

$$\frac{P_{max}}{P_{min}},$$

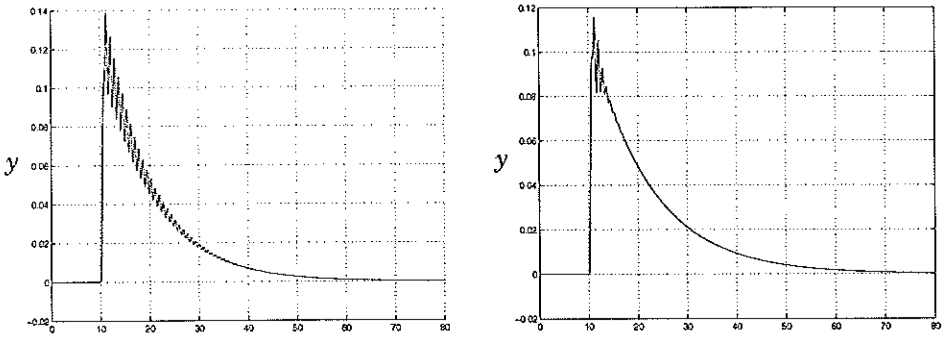

Figure 7 The time response for a short pulse on the input of the flexible shaft system system for the cases with no active control (left) and with active control (right).

given in Figure 5. Equation (6) should then be refined to,

$$K_b = k_b \frac{2\zeta_b w_b}{|P(i\omega_b)|}, \tag{9}$$

where k_b is a function of P_{max}/P_{min}. The function $k_b(P_{max}/P_{min})$ is obtained in the following way: for each process in the test batch calculate the IAE values of the closed loop system in Figure 3 for a set of k_b and choose the k_b which gives the minimum IAE. These values of k_b have been marked with a cross (x) in Figure 6 together with corresponding values of P_{max}/P_{min} . A rough approximation of $k_b(P_{max}/P_{min})$ is given by,

$$k_b = -0.46 + 0.33\frac{P_{max}}{P_{min}}, \tag{10}$$

which corresponds to the full line drawn in Figure 6.

Concluding Remarks: In this section an active control system F, has been designed. It rejects the excitation of undamped process modes in the output y, which are excited by the disturbance l in Figure 3. The active filter F is the product of the allpass filter F_a in (1), and the bandpass filter F_a in (2). The filter parameters of F_a, and F_b are determined in the following way:

ω_a**:** solve Equation (4).

ζ_a**:** ζ_a=0.5

ω_b**:** $\omega_b = \omega_p$

ζ_b**:** ζ_b=0.9

k_b**:** $k_b = -0.46 + 0.33P_{max}/P_{min}$

Example

The proposed active control system has been applied to an example found in the literature, the flexible shaft system in Farrugio (1998).

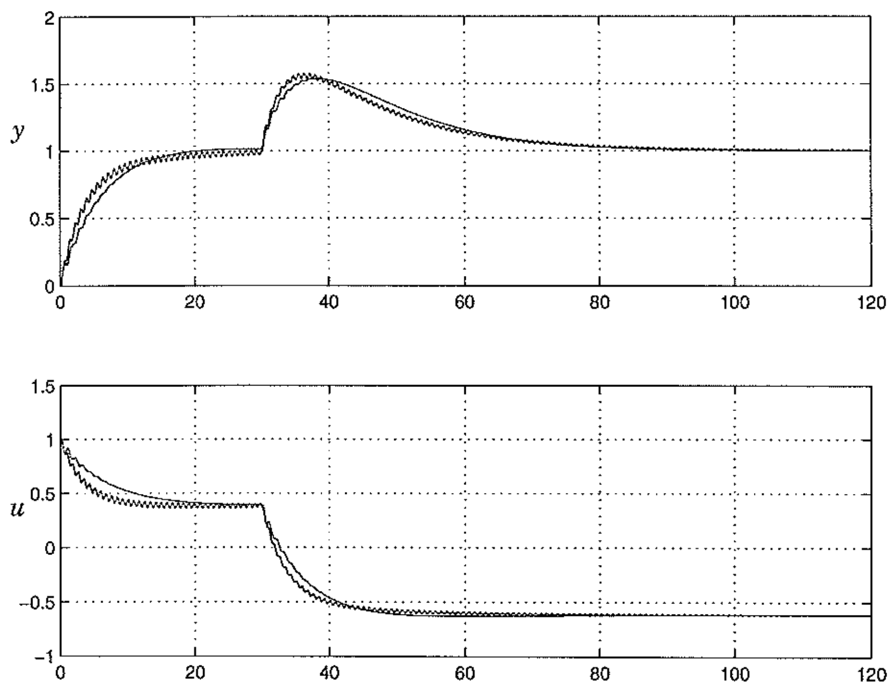

Figure 8 The time responses of the closed loop system when controlling the flexible shaft system system with a PI controller for the cases with no active control (undamped response) and with active control (well damped response).

The Flexible Shaft System

The flexible shaft system from the paper Farrugio (1998) consists of two flywheels which are coupled by a spring and can be approximated by the transfer function,

$$P(s) = \frac{\kappa}{(s+\omega_f)(s^2 + 2\zeta_p\omega_p s + \omega_p^2)}.$$

The proposed active control system has been tested on the nominal model in Farrugio (1998), with $\kappa = 15.21$, $\zeta_p = 0.02$, $\omega_p = 7.66$, $\omega_f = 0.10$. To determine the active filter F the parameters needed are: $\omega_b = 7.66, P_{max} = 0.85$, $P_{min} = 0.088$, $\omega_a = 7.64$, and $k_b = 2.75$. In Figure 7 the time response for a short pulse on the input of the flexible shaft system system is shown for the cases with no active control in the left figure and with active control in the right figure. Consequently, the active control system gives an improvement in performance.

Furthermore, the active control system has been tested in closed loop where a PI controller with controller parameters $K = 1.0$ and $T_i = 12.5$ is used. In Figure 8 the time response of the closed loop system is shown for the cases with no active control and with active control. The well damped response is obtained with the active controller which gives an improved performance of the closed loop system.

Conclusions

In this paper, an active control system is designed to handle processes with undamped modes. The objective of the active controller is to damp the oscillations in the process output, when a disturbance excites the undamped modes of the process. A modular approach has been taken, where the active filter consists of an allpass filter and a bandpass filter. The parameters of the allpass and bandpass filter are determined in a systematic way, where the only information needed is a few characteristics of the process frequency response. There are no requirements on the controller structure and design. The active filter has been applied to a variety of typical undamped systems, and it works well.

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