DESIGN SUPPORT FOR MOTION CONTROL SYSTEMS

Application to the Philips Fast Component Mounter

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Abstract: In (Coelingh, 2000) tools have been developed that support mechatronic design of controllers for electromechanical motion systems. In order to evaluate and illustrate application of some of these tools, an industrial motion system, the placement module of the Philips Fast Component Mounter, is considered. The presented design support helps the designer to more easily gain insight in the design problem, without requiring advanced control engineering skills, while indicating whether performance and robustness demands of the final design are being satisfied. Important consequences are that better-founded design decisions can be made and that the required overall development time decreases significantly.

1. INTRODUCTION

Global market developments show an increasing need for electromechanical motion systems with higher performance and good reliability that are developed within shorter time. This can be illustrated by considering current developments in electronics. Technological advances lead to larger variations in the size of electronic components and to components with more pins. As a consequence, the requirements for assembly machines for printed circuit boards (PCB's) are changing. Assembly operations must be performed more flexibly, faster and with higher accuracy. These demands can be partially met by the application of a mechatronic design approach (Van Brussel, 1996; Van Amerongen, 2000). The design of the control system should be considered as a part of the design of the system as a *whole*. To fully exploit the advantages of mechatronic design a deep understanding of the design problem has to be obtained. This formed the motivation for the development of design support for motion control systems, as described in (Coelingh, 2000). In here, the aim of research was defined as: "Enhance (mechatronic) design of control systems for electromechanical motion systems, such that insight in the design problem is obtained more easily, the control engineering skills required of the designer decrease and performance and robustness properties of the final design improve. As a result, the required development time should decrease".

In order to evaluate and illustrate application of the tools described in (Coelingh, 2000), we consider an industrial motion system: the placement module (PM module) of the Philips Fast Component Mounter (Philips, 2000). This machine is capable of placing 60,000 components on PCB's per hour, under nominal conditions. The FCM comprises a series of up to 16 servo-controlled pick-and-place robots, the so-called PM modules. In Fig. 1 a schematic diagram of the motion in *y*-direction, *i.e.* the "long stroke" of the PM module is shown.

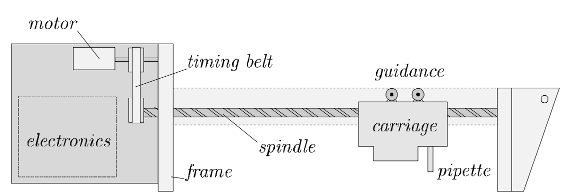

Fig. 1 The placement module of the FCM.

The design support has been developed for application in a mechatronic design process, where a realisation of the complete electromechanical subsystem is not existing during the design of the control system. However, the actual design of other subsystems than the control system is out of the scope of this paper. Therefore, an existing PM module is used. It is not an objective to maximise the performance of this specific PM module, using detailed knowledge of the plant dynamics obtained from system identification. Rather, our aim is to satisfy the requirements in a short design cycle. We will only use that plant knowledge that generally would have been available at a particular design stage. The designed control systems will be applied without additional tuning, in order to get a fair impression of their practical relevance.

In section 2, a simple model of the PM module is built and the assessment method for conceptual design is applied. In section 3, the plant model is extended with additional dynamic effects and a structured design method is applied. Section 4 discusses the practical realisation of the different control systems. Section 5 presents the conclusions.

2. CONCEPTUAL DESIGN

The conceptual design stage is crucial in a mechatronic design process. Here, the functional interaction between domain specific subsystems is determined (De Vries, 1994). Therefore, an assessment method is formulated that supports the design of a feasible reference path generator, control system and electromechanical plant with appropriate sensor locations, in an *integrated* way. This method is based on the work of (Groenhuis, 1991).

2.1 Characterisation of the task

A 1-dimensional motion along the spindle of the PM module is considered. The corresponding task is to place a component with an accuracy of 100 [μm] at 30 [ms] after the motion time. The maximum velocity and acceleration of the system are specified as 1 [ms^{-1}], respectively 10 [ms^{-2}] (Philips, 1998). A second-degree reference path is used, that is characterised by a motion distance h_m of 0.1 [m] and a motion time t_m of 0.2 [s]. This is the characteristic task of the controlled system, *i.e.* the performance obtained for this task is characteristic for the controlled system, as it only consists of acceleration and deceleration (Koster *et al.*, 1999).

Table 1 Specifications for the PM module

quantity		value	
maximum error (after t_s)	e_0	100	[μm]
motion time	t_m	250	[ms]
motion distance	h_m	0.15	[m]
settling time	t_s	30	[ms]
maximum acceleration	a_{max}	10	[m/s^2]
maximum velocity	v_{max}	1	[m/s]

The task that is actually used in simulations and experiments also includes a period of 0.05 [s] with constant maximum velocity (Table 1).

2.2 Characterisation of the plant dynamics

For conceptual design a plant model is used that only represents the dominant dynamic behaviour of the electromechanical system, consisting of the rigid body mode and the first mode of vibration. In Fig. 2 a so-called standard model is indicated that represents the PM module. In (Coelingh, 2000) four standard models (*classes*) are described that represent 4th-order electromechanical plant models, each with the dominant compliance located in another mechanical subsystem, *e.g.* mechanism, frame, guidance or actuator suspension (as in Fig. 2).

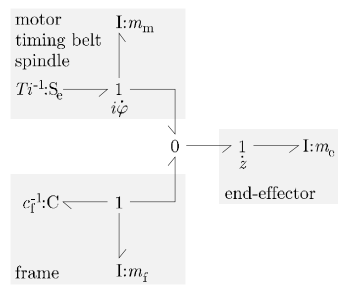

Fig. 2 Standard model of the PM module.

This model is obtained by simplifying and reducing a plant model that was built from several component models. The corresponding parameter values are indicated in Table 2, where the motor mass m_m is a transformed quantity that contains the inertia of the motor, the pulleys and the spindle.

Table 2 Parameter values for standard model

quantity		value	
motor mass	m_m	6.53	[kg]
end-effector mass	m_e	2.3	[kg]
frame mass	m_f	16.5	[kg]
frame stiffness	c_f	$4.3 \cdot 10^6$	[$N \cdot m^{-1}$]

The model indicates that the resulting velocity of the end-effector is a summation (0-junction) of the velocity of the flexible frame and the velocity of the motor mass, *i.e.* the class of flexible actuator suspension.

2.3 Assessment

The plant model is in the standard form, such that the assessment method can be applied in steps. For details about this method is referred to (Coelingh, 2000). Here, merely the straightforward application is shown and the results are interpreted.

1. *Determine the class of electromechanical motion system that is at hand.*

We consider the plant transfer function from the input force Ti^{-1} to the measured actuator position $i\varphi$. The total mass to be moved equals:

$$m = m_{\mathrm{m}} + m_{\mathrm{e}} = 8.83 \quad [\mathrm{kg}] \tag{1}$$

When the actuator position is considered the transfer function is of type *AR*, *i.e.* in the frequency response one first finds a complex zero-pair (anti-resonance) and consecutively a complex pole-pair (resonance). The expression for the plant transfer function is:

$$P_{\mathrm{AR}}(s) = \frac{1}{ms^2} \cdot \frac{s^2 + \omega_{\mathrm{ar}}^2}{s^2 + \omega_{\mathrm{r}}^2} \cdot \frac{\omega_{\mathrm{r}}^2}{\omega_{\mathrm{ar}}^2}, \quad \omega_{\mathrm{ar}} < \omega_{\mathrm{r}} \tag{2}$$

The anti-resonance frequency respectively resonance frequency of the lowest mode of vibration is:

$$\omega_{\mathrm{ar}} = \sqrt{\frac{c}{m_{\mathrm{f}} + m_{\mathrm{e}}}} = 478 \quad [\mathrm{rad/s}] \tag{3}$$

$$\omega_{\mathrm{r}} = \sqrt{\frac{c\,(m_{\mathrm{m}} + m_{\mathrm{e}})}{m_{\mathrm{m}}\,(m_{\mathrm{f}} + m_{\mathrm{e}}) + m_{\mathrm{f}}\, m_{\mathrm{e}}}} = 486\,[\mathrm{rad/s}] \tag{4}$$

2. *Determine the concept that is at hand, by looking at the location of the position and velocity sensor and verify whether optimal dimensionless controller settings can validly be applied.*
   For the actual PM module we use position and velocity feedback from the motor axis, *i.e.* concept *AR*. Whether the optimal dimensionless controller settings, as indicated in the assessment method, can validly be applied depends on the value of the frequency ratio ρ. For concept *AR*, the frequency ratio must have a value between 0.1 and 0.8. The actual value of the frequency ratio is:

$$\rho = \left(\frac{\omega_{\mathrm{ar}}}{\omega_{\mathrm{r}}}\right)^2 = 0.97 \tag{5}$$

   which is larger than the permitted 0.8. Despite this fact, we will continue application of the assessment method, but we have to reconsider the violation of the bounds on ρ afterwards.

3. *Depending on the situation, perform one of three alternatives:*
   a. When we assume the reference path and the desired performance, in terms of the maximal positional error e_0, to be fixed, we calculate the periodic ratio τ according:

$$\tau = \sqrt{\frac{e_0}{\varepsilon \cdot h_{\mathrm{m}}}} = \sqrt{\frac{1.0 \cdot 10^{-4}}{0.09 \cdot 0.1}} = 0.11 \tag{6}$$

   The value for the constant ε for concept *AR* is 0.09. The minimal required anti-resonance frequency of the plant is:

$$\omega_{\mathrm{ar,req}} = \frac{2\pi}{\tau t_{\mathrm{m}}} = \frac{2\pi}{0.11 \cdot 0.2} = 298\,[\mathrm{rad/s}] \tag{7}$$

   which is smaller than the actual anti-resonance frequency, thus the lowest mode of vibration is not a limitation for the desired performance.

   b. When we assume the reference path and the anti-resonance frequency to be fixed, we calculate the periodic ratio τ according:

$$\tau = \frac{2\pi}{\omega_{\mathrm{ar}} t_{\mathrm{m}}} = 0.066 \tag{8}$$

   The attainable performance in this situation is predicted as:

$$e_0 = \varepsilon \cdot \tau^2 \cdot h_{\mathrm{m}} = 39 \quad [\mu\mathrm{m}] \tag{9}$$

   which is smaller than the desired accuracy of 100 [μm].

   c. When we assume the desired performance and the anti-resonance frequency to be fixed, we can propose a characteristic reference path. This requires a trade-off between the motion time and the motion distance:

$$\begin{aligned} \frac{2\pi}{\omega_{\mathrm{ar}} t_{\mathrm{m}}} &= \sqrt{\frac{e_0}{\varepsilon \cdot h_{\mathrm{m}}}} \\ t_{\mathrm{m}}^2 &= 0.16 \cdot h_{\mathrm{m}} \end{aligned} \tag{10}$$

4. *Determine the control system for a particular problem setting.*
   The dimensionless optimal controller settings for concept *AR* are $\Omega_{\mathrm{p}} = 0.8$ and $\Omega_{\mathrm{d}} = 0.9$. For the particular problem setting of the PM module, with $\omega_{\mathrm{ar}} = 478$[rad/s], we obtain:

$$\begin{aligned} k_{\mathrm{p}} &= m \cdot \left(\Omega_{\mathrm{p}} \cdot \omega_{\mathrm{ar}}\right)^2 = 1.29 \cdot 10^6 \\ k_{\mathrm{d}} &= m \cdot \Omega_{\mathrm{d}} \cdot \omega_{\mathrm{ar}} = 3.80 \cdot 10^3 \end{aligned} \tag{11}$$

From application of the assessment method we learn that the specifications can be met with the given electromechanical subsystem. However, the frequency ratio (5) of the PM module is not within the bounds specified by the assessment method, thus the stability margin may be smaller than guaranteed. The assessment method can also be used to evaluate different sensor locations, *e.g.* on the end-effector, but this is not discussed here.

2.4 Evaluation

By means of simulations we will evaluate the conceptual design. The end-effector appeared to follow the reference trajectory well. The error during the point-to-point motion rises up to 3 [mm] and the required torque of the actuator is about 0.3 [Nm]. The positional error after the motion time t_{m}, *i.e.* the error of interest, is shown in Fig. 3.

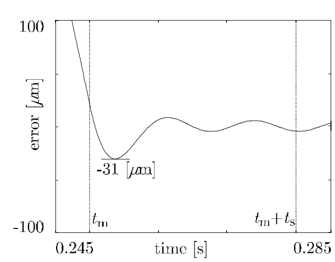

Fig. 3 The positional error of the conceptual design.

The simulations show that the positional error after the motion time t_m is 31 [μm], which is indeed smaller than the predicted maximum positional error of 39 [μm]. After the settling time t_s of 30 [ms], we see that the error is decreased even further. It is concluded that a feasible design for the controlled system is obtained. The design process is continued, keeping in mind the relatively small stability margin.

3. DETAILED DESIGN

After conceptual design, the design process will proceed towards a design proposal that is to be the guideline for prototyping. The character of the design problem at hand will gradually change. During detailed design frequency-domain design methods are being used, as aspects such as performance, stability, disturbance attenuation and robustness are conveniently expressed in these terms, in case of electromechanical motion systems. For detailed design a structured design method has been described in (Coelingh, 2000) for a control configuration consisting of a reference path generator, a feedback component, a feedforward component and a disturbance observer. In this paper the design of only the feedback component is discussed.

3.1 Modelling the plant

The plant model of Fig. 2 is extended with additional dynamic effects. Fig. 4 shows the different component models and their interaction. The transmission, spindle and guidance of the end-effector are considered to be flexible, thus introducing higher-order modes. Also the limitation of the input current of the plant is incorporated (± 5 [A]).

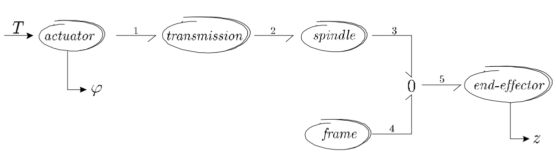

Fig. 4 Extended plant model.

3.2 Design of a feedback component

For the feedback component two different configurations are used, a PID compensator and a PID controller. The essential difference is that in a PID controller the derivative action does not operate on the reference path.

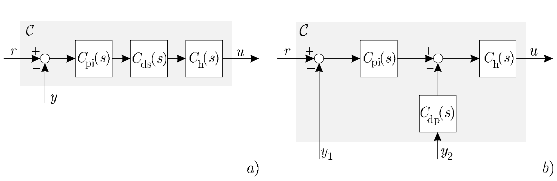

Fig. 5 a) PID compensator and b) PID controller configuration.

The expressions for the subcomponents are

$$\begin{aligned} C_{pi}(s) &= K \cdot \frac{\tau_i s + 1}{\tau_i s} \\ C_{ds}(s) &= \frac{\tau_d s + 1}{\tau_d \beta s + 1} \\ C_{dp}(s) &= K \cdot \frac{\tau_d s}{\tau_d \beta s + 1} \\ C_h(s) &= \frac{1}{\tau_h s + 1} \end{aligned} \tag{12}$$

where K denotes the proportional gain, τ_i the integral time constant, τ_d the derivative time constant and τ_h the high-frequency roll-off time constant. The tameness factor β is chosen such that the derivative action is effective over a limited frequency range only. In order to have a derivative effect, β is chosen smaller than 1.

Before the design of a feedback component the design specifications have to be formulated. The design procedure requires the specifications to be expressed in terms of the bandwidth and the low-frequency behaviour of the input sensitivity function. This method is based on the work in (De Roover, 1997). An indication for performance has already been obtained during conceptual design; therefore, it is desirable to reuse this result. We will determine the bandwidth ω_b of the conceptual design with the standard plant model:

$$\omega_b = 266 \quad \text{[rad/s]} \tag{13}$$

This is the frequency where $|1 - T(j\omega)|$ crosses the 0 dB-line, where $T(j\omega)$ is the complementary sensitivity function. Note that in case of a PID controller $|1 - T(j\omega)|$ does not equal the sensitivity function $S(j\omega)$.

We will therefore state the specification as:

1. sufficiently high bandwidth: ω_b = 266 [rad/s].
2. sufficient suppression of low-frequency vibrations: $|S_{wz}(j\omega)| < s_l$ [dB] for $\omega < \omega_l$ [rad/s], with s_l = -120 [dB] and $\omega_l = 10\pi$ [rad/s].
3. maximum peak-value of the sensitivity magnitude: M_S = 6 [dB]
4. limited control energy.

where $S_{wz}(j\omega)$ is the input sensitivity function. The specification for the suppression of the low-frequency vibrations is chosen rather arbitrarily: disturbance forces up to 10π [rad/s], acting on the input of the plant, have to be suppressed with at least –120 [dB]. This means that forces with a frequency smaller than 10π [rad/s] and an amplitude of 10 [N], may result in a displacement of 10 [μm]. The design procedure for PID controllers (Coelingh, 2000) is applied in two steps. Firstly, a PD controller (Fig. 5b) is designed in order to illustrate the reuse of the conceptual design and secondly we add the integral action.

1. *Determine m, i.e. the total mass to be displaced.*
   In the previous section we determined:

$$m = 8.83 \quad \text{[kg]} \tag{14}$$

2. *Determine whether to use a PID compensator or a PID controller.*
   We start with the design of a PID controller, as this configuration was used during conceptual design.

3. *Select the parameter β, i.e. the tameness factor.*
   It is chosen equal to the default value 0.1.

4. *Determine the value of the derivative time constant τ_d to obtain extra phase lead.*
   For the controller we use.

$$\tau_d = \frac{1}{4\omega_b\sqrt{\beta}} = 2.97\cdot10^{-3} \tag{15}$$

5. *Determine the proportional gain K to obtain the desired bandwidth ω_b.*
   For the controller we use:

$$K = 2\omega_b^2 m\left(\frac{\omega_b^2\tau_d^2\beta^2+1}{\omega_b^2\tau_d^2\beta^2+2\omega_b^2\tau_d^2\beta+1}\right) = 1.12\cdot10^6 \tag{16}$$

The newly obtained PD controller can be compared with the controller settings of the assessment method (11). The proportional gain of the conceptual design is a little larger and the derivative action can be rewritten as:

$$\frac{k_d}{k_p} = 2.94\cdot10^{-3} \tag{17}$$

which almost equals the value for τ_d. The frequency responses and the bandwidth ω_b of both designs are indeed the same, although they are obtained from different starting points. We obtained a smooth transition from conceptual design to detailed design, *i.e.* a design in the frequency domain, by reusing the bandwidth ω_b. Now, we add the integral action by continuing the design procedure:

6. *Determine the integral time constant τ_i in order to obtain a desired gain at low frequencies.*

$$\tau_i \le \frac{s_l K}{\omega_l} = 3.5\cdot10^{-2} \tag{18}$$

7. *Choose the time constant of the roll-off filter $\tau_h < \beta\cdot\tau_d$, but as large as possible to limit the controller bandwidth.*
   We choose:

$$\tau_h = 1.5\cdot10^{-4} \tag{19}$$

Similarly, we can design a PID compensator (Fig. 5a), starting with step 3 of the design procedure.

3. *Select the parameter β, i.e. the tameness factor.*
   It is chosen equal to the default value 0.1.

4. *Determine the value of the derivative time constant τ_d to obtain extra phase lead.*
   For the compensator we use:

$$\tau_d = \frac{1}{2\omega_b\sqrt{\beta}} = 5.93\cdot10^{-3} \tag{20}$$

5. *Determine the proportional gain K to obtain the desired bandwidth ω_b.*
   For the compensator we use:

$$K = 2\omega_b^2 m\left(\frac{\omega_b^2\tau_d^2\beta+1}{\omega_b^2\tau_d^2+1}\right) = 4.5\cdot10^5 \tag{21}$$

6. *Determine the integral time constant τ_i in order to obtain a desired gain at low frequencies.*

$$\tau_i \le \frac{s_l K}{\omega_l} = 1.4\cdot10^{-2} \tag{22}$$

7. *Choose the time constant of the roll-off filter $\tau_h < \beta\cdot\tau_d$, but as large as possible to limit the controller bandwidth.*
   We choose:

$$\tau_h = \beta\tau_d = 3.0\cdot10^{-4} \tag{23}$$

3.3 Evaluation

In Fig. 6, the frequency responses of the PID controller and PID compensator are shown, with the standard fourth-order plant model.

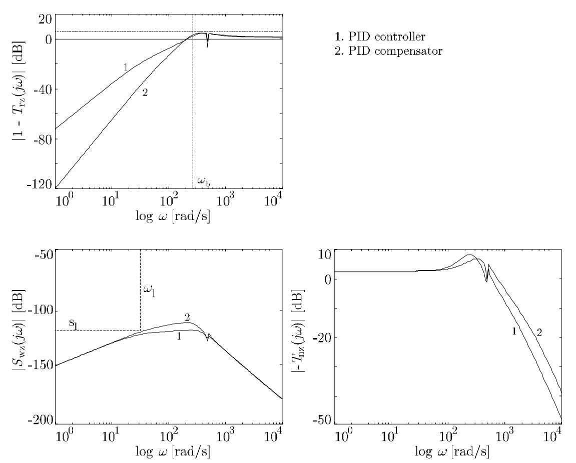

Fig. 6 Frequency responses of the PID controller and PID compensator.

Application of the assessment method indicated that the stability margin requires extra attention. Here, we see that the peak of $|1 - T(j\omega)|$ is smaller than 6 [dB], thus providing a sufficient stability margin. The desired low-frequency disturbance attenuation can be achieved with both designs. The controller configuration does not require an integral action to meet this requirement, as the suppression of the PD controller is already –120 [dB], but we maintain the integral action as it does provide infinite gain at zero frequency. The proportional gain of the compensator is much lower than the proportional gain of the controller, which results in a lower cost of feedback. The roll-off can start at a lower frequency, thus the amplification of measurement noise, is much smaller for the compensator configuration. We will also compare the two control configurations by means of simulations, using the extended plant model of Fig. 4. The controller configuration allows a large error during the motion time. Therefore, the integral action will only be switched on, once the end-effector is near the end position. The specifications of the PM module indicate that the switching moment is 8 [ms] after the motion time. The same switching moment is used here.

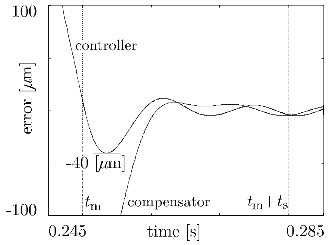

Fig. 7 Positional error of closed-loop systems with PID controller and PID compensator.

In Fig. 7, it can be seen that both designs fulfil the specification after the settling time. The maximal error after the motion time t_m is much smaller for the PID controller (40 [μm]) than for the compensator (210 [μm] at t = t_m). This illustrates that a PID controller is attractive in point-to-point motion systems with only a feedback component.

4. PRACTICAL APPLICATION

We will apply and test the different control configurations to the real PM module. The position of the end-effector is measured over a small range by means of a Position Sensitive Detector (PSD), such that we can verify the behaviour of the variable of interest. The PSD output is not used for feedback.

We start with the PD-type controller of section 2 that has been designed with the assessment method. In Fig. 8, the experimental error plots are shown that correspond to the simulations of Fig. 3. The maximal positional error after the motion time is 37 [μm], which is 6 [μm] larger than in simulation, but corresponding well with the positional error of 39 [μm] predicted by the assessment method.

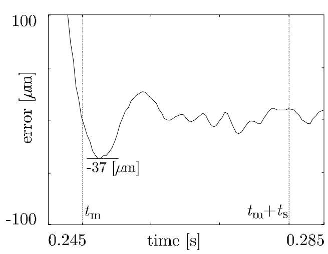

Fig. 8 Measured error of PM module with the PD-type controller from the assessment method.

Next, we consider the PID controller and PID compensator of section 3, where the integral action is switched on 8 [ms] after the motion time.

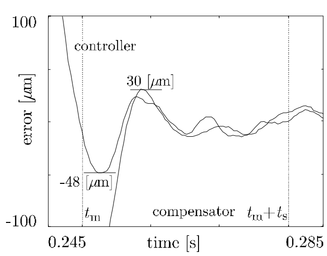

Fig. 9 Measured error of PM module with PID controller and PID compensator.

In Fig. 9, the experimental error plots are shown that correspond to the simulations of Fig. 7. The characteristics of the error plots are similar to those in simulation, although the actual values of the positional errors are again a little larger.

5. CONCLUSIONS

By means of application to an industrial motion system, the design enhancement presented in (Coelingh, 2000) was illustrated. The focus was mainly on evaluating the design enhancement and not on maximisation of the performance of the controlled system. Assessment learned that the specification of a maximal positional error of 100 [μm] can be met; a maximum error of 39 [μm] was indicated as feasible. Simulations with an initial plant model, as well as practical experiments, confirm these results. This illustrates that, using minimal plant knowledge, the assessment method provides the designer with relevant knowledge about the design problem, early in the design process. During detailed design, the conceptual design is reused relatively easy, by using the bandwidth of the conceptual design as the initial specification for the design of the feedback component. This component now also has to provide low-frequency disturbance suppression. Simulations and frequency analysis confirm that the specifications are feasible. The main benefit of the structured design method is that it leads to a short design process, while providing insight in the design problem. The recommended controller settings proved to be relevant in practice, that is, as long as the parameters of the initial model are about correct.

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