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Application of Advanced Control Theory to Fluid Power Control

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ABSTRACT

This paper presents high performance control of electrohydraulic actuators for generating near periodic time varying trajectories. Such type of trajectories can be found in many industrial applications, particularly those involving master-slave type electronic cam-follower motion generation. The control algorithm includes a robust feedback control for disturbance rejection, a repetitive control to compensator for periodic signals, and a previewed feedforward control for tracking time varying signals. The three control actions are numerically solved simultaneously by formulating the control problem as a μ-synthesis problem. The μ-synthesis formulation includes practical design constraints by imposing frequency domain bounds of disturbance rejection, model matching error for tracking, unmodeled dynamics for robust stability, a periodic signal generator in the repetitive control, and a delay filter that corresponds to the preview length of the feedforward compensator. The proposed control design approach is applied to an electrohydraulic actuator to generate cam lobe profiles for cam shaft machining application. Control realization issues including model structure of electrohydraulic actuators, characterization of system model and uncertainty bounds, and controller order reduction for real-time implementation by digital signal processors are discussed. Experimental results are presented to demonstrate the design process and control performance.

KEY WORDS
Electrohydraulic Actuators, Repetitive Control, Robust Control, Preview, Feedforward Control

INTRODUCTION

Fluid power is a main source of actuation used in diverse industrial applications where high level of dynamic motion and force requirements make it as the only choice among various methods. In many applications, hydraulic actuators are used to generate motion in synchronization with a process. The motion generated by this master-slave type of electronic cam is often repetitive or near periodic. For example, in machining of non-circular engine parts, such as pistons, crankshafts, and camshafts, the cutting tool must precisely tracks a predefined profile as a function of the work piece rotational angle and axial length. The profile essentially repeats itself in every work piece rotation, but may change with respect to the tool's feed progression along the work piece axial length. Other examples include camless engine valve motion generated by electrohydraulic actuators and hydraulic ram motion for material forming (injection molding and stamping). Control of electrohydraulic actuators for generating dynamic motion is challenging because the high order dynamics and nonlinearities presented in the hydraulic system requires advanced control design approach beyond the conventional low order linear control approach. Industrial motion controllers, most of which are limited to PID (proportional-integral-derivative) plus velocity and acceleration feedforward control actions, are generally effective for controlling mechanical motion generated by electric servo motors, but they are usually ineffective in controlling high bandwidth servo hydraulics. This paper addresses high order linear control design for high performance trajectory tracking with application to electrohydraulic systems, where the nonlinearities are accounted for in characterizing the uncertainty bounds of the linear model.

In tracking or rejecting periodic signals, the internal model principle [1] states that a periodic signal generator is required in the feedback loop to make output asymptotically track or regulate this periodic signal. Several discrete-time repetitive control design techniques [2, 3, 4, 5, 6, 7] have been proposed in literature. However, they do not consider near periodic or non-periodic signals. Some published literature in discrete-time repetitive control considers both periodic and non-periodic signals. In [8] a disturbance observer is used to estimate and cancel disturbances in the inner feedback loop. In [9] the low-pass filter of the periodic signal generator in the prototype repetitive control proposed in [2, 6] is modified by frequency shaping of the sensitivity function so that the resulting sensitivity function may selectively reduce repeatable and non-repeatable runout in the computer disk drive head servo control problem. A method for designing robust repetitive control using structured singular values was proposed in [10, 11]. In this method, the non-periodic signal is accounted for by specifying a sensitivity weighting function and the high order delay term in the periodic signal generator is treated as a fictitious uncertainty so that μ-synthesis may produce a reasonably low order controller. The above repetitive control designed for rejecting disturbances are not equipped with previewed feedfoward action for tracking reference signals.

Tracking control problems are usually designed with a two-degree-of-freedom control structure, in which the feedback and feedfoward controllers are designed independently. A two-degree-of-freedom extension of H_∞ loop-shaping design [11] to enhance the model-matching properties of the closed-loop was proposed in [13, 14]. In [15] a two-step design procedure was proposed based on Youla parametrization of two-degree-of-freedom controllers. In the first step a model-matching approach is used to set the desired nominal tracking objectives and in the second step μ-synthesis technique is applied to achieve the robust performance objectives. A game theory approach was proposed to solve the H_∞ tracking control problem of a causal or non-causal reference signal [16]. These two-degree-of-freedom controllers cannot readily include the repetitive control action because including the internal model, which contains a long delay in the periodic signal generator, would substantially complicates the model matching problem and solution.

This paper presents an integrated feedforward, robust feedback, and robust repetitive control structure and its design method. The robust feedback control action provides the basic disturbance rejection and tracking function. The repetitive control action deals with periodic components of the disturbance and reference signal. The previewed feedforward addresses non-periodic time varying reference signals. The integrated control is particularly effective in tracking near periodic signals. Based on this control structure, we first present a sequential design method [17] based on zero phase error tracking control (ZPETC) approach [18]. Then a integrated design approach [19] that simultaneously solves the three control actions by μ-synthesis framework is presented

The rest of this paper is organized as follows: Section 2 presents the proposed control structure with the ZPETC design approach. Section 3 presents the presents μ-synthesis design approach. Section 4 presents the modeling of an electrohydraulic servo system for robust control design. Section 5 presents the digital control implementation and experimental results for the μ-synthesis design to the system. Finally, conclusions are given in Section 6.

ROBUST FEEDBACK REPETITIVE AND FEEDFORWARD CONTROL BY ZPETC

The control structure shown in Figure 1 consists of the three control actions: robust feedback control, repetitive control, and feedforward control. The robust control action is realized by a causal compensator K_1. The repetitive control consists of the periodic signal generator and a causal compensator K_2. The filter Q will be described later. Finally, the feedforward control consists of a F-step delay block and a causal compensator K_3. Before the F-step delay block is the previewed reference signal $r(k+F)$. A simple design approach of the repetitive and feedforward control based on the zero-phase-error tracking control (ZPETC) [18] is presented in this section to demonstrate this control structure's feature.

In robust feedback control system design the system dynamics are modeled with a multiplicative uncertainty in the form of

$$G(s) = [1 + \Delta(s)W_r(s)]\, G_o(s), \qquad (1)$$

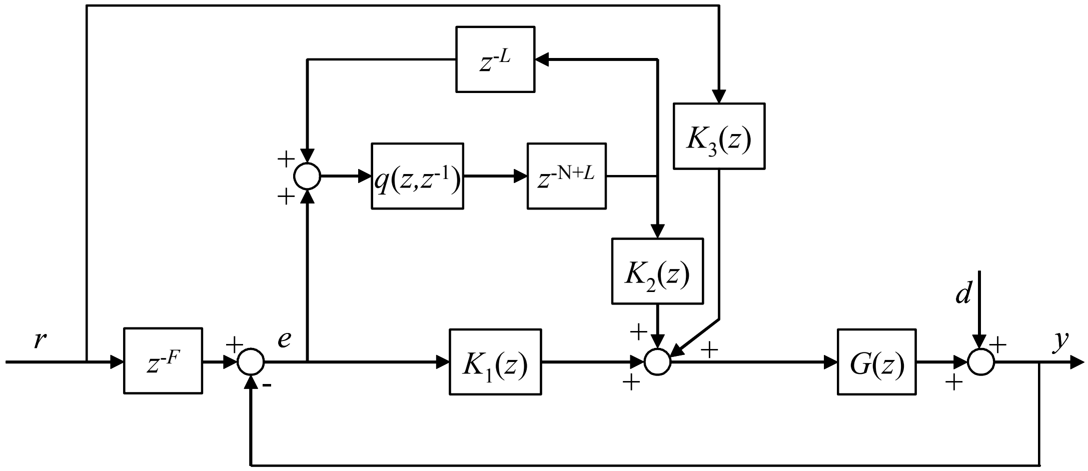

Figure 1. Block diagram of the control system.

where $G(s)$ represents the real system dynamics, $G_o(s)$ is the nominal model used for control design, and $W_r(s)$ is a fixed stable transfer function which bounds the model uncertainty. The function $\Delta(s)$ includes all stable transfer function satisfying

$$\| \Delta(s) \|_\infty := \sup_\omega \; | \Delta(j\omega) | \le 1. \qquad (2)$$

The robust performance is specified in the frequency domain by a weighting function $W_p(s)$, which typically has larger magnitudes at lower frequencies, indicating desired tracking at these frequencies:

$$\| W_p S \| < 1 \text{ , where } S = \frac{1}{1+GK_1} \qquad (3)$$

A necessary and sufficient condition for robust performance [13] is (in frequency domain)

$$\| \; | W_p S_o | + | W_r T_o | \; \| < 1 \qquad (4)$$

$$S_o = \frac{1}{1+G_o K_1}, \quad T_o = \frac{G_o K_1}{1+G_o K_1}$$

The tracking performance represented by the sensitivity function S is not satisfactory for precise dynamic profile generation because of the dynamic delays, particular phase delays, in the closed loop system. For example, 1 degree of phase error of a single frequency sinusoidal wave will generate about 1.7% maximum tracking error in magnitude. Repetitive control and feedforward control are used in a plug-in manner to the feedback controller to achieve superior tracking performance.

One effective method to design discrete-time repetitive controller is the prototype repetitive control as described in [2, 3]. This controller is designed based on the nominal closed-loop transfer function $G_o T_o$:

$$H_o(z^{-1}) = G_o(z^{-1})\, T_o(z^{-1}) = \frac{z^{-d} B_{cl}(z^{-1})}{A_{cl}(z^{-1})} , \qquad (5)$$

$$A_{cl}(z^{-1}) = 1 - a_1 z^{-1} - \dots - a_n z^{-n}$$

$$B_{cl}(z^{-1}) = b_0 + b_1 z^{-1} + \dots + b_m z^{-m} , \quad b_0 \neq 0$$

As Figure 1 shows, the prototype repetitive control action consists of the periodic signal generator, where N represents the number of data points in one period, cascaded with a compensator K_2:

$$K_2(z^{-1}) = z^{-L} \cdot \frac{z^d A_{cl}(z^{-1}) B_{cl}^{-}(z^{-1})}{b \cdot B_{cl}^{+}(z^{-1})}$$

$$b \ge \left| B_{cl}^{-}(e^{j\omega T}) \right|^2 \qquad (6)$$

Here, $B_{cl}^{+}(z^{-1})$ contains the stable zeros of $B_{cl}(z^{-1})$ to be cancelled, and $B_{cl}^{-}(z^{-1})$ must contain all the roots of $B_{cl}(z^{-1})$ outside or on the unit circle to avoid unstable pole-zero cancellation. Also $B_{cl}(z^{-1})$ is obtained by substituting z for z^{-1} in $B_{cl}^{-}(z^{-1})$.

The low-pass, zero-phase filter $Q(z,z^{-1})$ is included in the periodic signal generator to ensure robust stability for the repetitive controller. For this, the multiplicative uncertainty bound for the closed loop transfer function H_O is calculated in the discrete-time domain from the open loop plant uncertainty bound $W_r(\omega)$ for each case and is denoted as $W_{rcl}(\omega)$. The following robust stability condition can be derived [6] to determine Q:

$$|Q(\omega)| < \frac{1}{|W_{rcl}(\omega)|} \quad \text{for all } \omega. \tag{7}$$

To further enhance the tracking performance, especially for near periodic or non-periodic reference signal, feedforward compensator $K_3(z)$ is introduced. In tracking control, it is often desirable to incorporate preview action to compensate for the dynamic delay of the plant. This means that a finite number of future reference signal is available and that the feedforward controller needs not be causal. A general technique to design optimal feedforward tracking control based on different criteria and constraints are presented in [20]. A simple yet effective method is to implement the zero phase error tracking controller (ZPETC) [18] based on the open loop nominal plant model $G_O(z^{-1})$:

$$G_O(z^{-1}) = \frac{z^{-d} B(z^{-1})}{A(z^{-1})}, \tag{8}$$

$$K_3(z^{-1}) = z^{-F} \frac{z^{d} A(z^{-1}) B^{-}(z)}{B^{+}(z^{-1}) [B^{-}(1)]^2}, \tag{9}$$

where $B^{-}(z)$ and $B^{+}(z^{-1})$ are as previously described. The F-step finite preview (look ahead) of the reference signal corresponds to the plant delay d plus the order of $B^{-}(z)$.

With the above three control actions, the tracking performance characterized by the transfer function from the reference input to the tracking error (r → e) for the nominal plant model is

$$S_{tot} = S_{ff} \cdot S_{rep} \cdot S_o, \tag{10}$$

where

$$S_{ff} = (1 - G_O K_3) \tag{11}$$

$$S_{rep} = \frac{1 - Qz^{-N}}{1 + Q(K_2 H_o - 1)z^{-N}} \approx 1 - Qz^{-N} \tag{12}$$

The multiplying effect of adding the plug-in feedforward and the repetitive controllers to the feedback controller make it easy to identify the contribution of each control action in the tracking error reduction.

Now let T be the sampling time and substitute $z=e^{j\omega T}$, then it is easy to see from Eq. (12) that as long as the zero-phase low-pass filter $Q(z, z^{-1})$ is close to unity, the magnitude of sensitivity function S_{rep} is close to zero at integer multiples of $1/NT$, which represents the Fourier harmonic frequencies of the periodic signal. The magnitude may be as large as 2 between these frequencies. Thus the ZPETC-type prototype repetitive control may amplify non-periodic disturbances.

ROBUST REPETITIVE AND FEEDFORWARD CONTROL BY μ-SYNTHESIS

μ-synthesis is a powerful control design technique for a system subjected to structured, linear fractional transformation (LFT) perturbations. While designing a controller via by μ-synthesis, it is convenient to enforcing the robustness (both stability and performance) of the system by providing appropriate weighting functions. It is a well known fact that modern control design methods such as H-infinity control and μ-synthesis produce controllers of order at least equal to the plant, and usually higher because of the inclusion of weighting functions. With regards to computational complexity and practical implementation, the order of z^{-N+L} in the periodic signal generator in Figure 1 is too high to be directly included in a discrete-time μ-synthesis formulation. One useful technique [10, 11] is replacing z^{-N+L} by a fictitious uncertainty Δ_f and then applying μ-synthesis to design a robust repetitive controller. In addition, a reference model M is introduced and is to be matched by the overall transfer function from the reference r to the output y. The overall μ-synthesis control design structure is shown in Figure 2.

In Figure 2, the three uncertainty blocks Δ_r, Δ_p, Δ_d, in addition to the aforementioned uncertainty Δ_f , are accompanied by the respective weighting functions. The weighting function $W_r(z)$, as previously described, specifies the bound of the plant uncertainty. The weighting function $W_p(z)$ specifies the bound for the model reference matching error (r → e_m). The weighting function $W_d(z)$ specifies the bound for disturbance rejection (d → e_m). The reference model $M(z)$ may be a zero-phase low-pass filter with unity gain. Notice that a non-causal $M(z)$ can be used, as long as $z^{-F} M(z)$ is causal.

When all the uncertain perturbations are pulled out into a block-diagonal matrix, the final augmented block structure of the perturbations is

$$\hat{\Delta} = \begin{bmatrix} \Delta_r & 0 & 0 & 0 \\ 0 & \Delta_f & 0 & 0 \\ 0 & 0 & \Delta_p & 0 \\ 0 & 0 & 0 & \Delta_d \end{bmatrix} \tag{13}$$

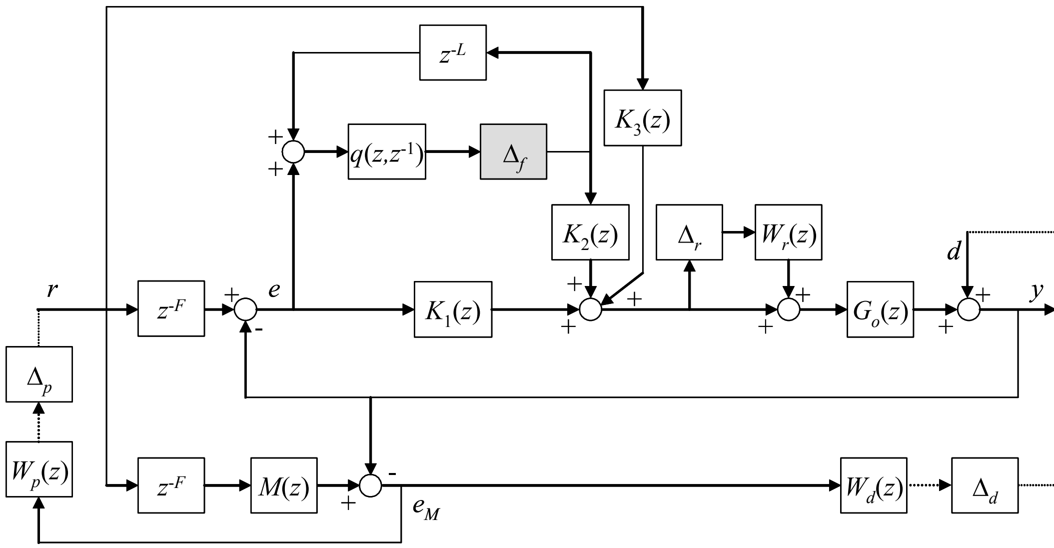

Figure 2 μ-synthesis control formulation

When the controllers [K_1 K_2 K_3] are pulled out, the remaining part is the generalized plant P, as shown in Figure 3. Assume that the nominal stability is achieved such that

$$U = F_l\left(P, K\right) = P_{11} + P_{12}K\left(I - P_{22}K\right)^{-1} P_{21} \qquad (14)$$

is (internally) stable, then robust performance is obtained if the following structured singular value (μ-value) is satisfied [21]:

$$\mu_{\hat{\Delta}}\left(U\right) < 1 \qquad (15)$$

Unlike the ZPETC-type repetitive control system, this method does not require the plant G(z) to be stable.

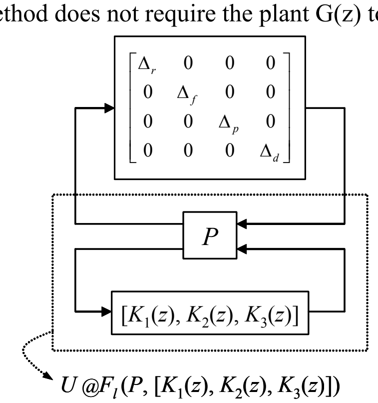

Figure 3 LFT form of the Control System

Further, the design of the repetitive controller K_2 is not based on closed loop transfer function H_o, which includes the robust feedback controller K_1.

MODELING OF ELECTROHYDRAULIC SERVO SYSTEM FOR ROBUST CONTROL DESIGN

In this section, we present a design example for an electrohydraulic system used in the noncircular turning process. Figure 4 shows a "two-dimensional" cam-shape profile used in the experiment. This profile could correspond to variable cam timing or lift in contrast to the conventional "one-dimensional" profile, in which the cross section is fixed. θ- and x-directions are defined as shown in the figure.

Now consider machining this twisted cam-shape by direct turning process with a constant spindle speed and feed rate. The tool motion should follow the reference given by the two dimensional profile $r(\theta,x)$. Notice in the figure that the cam profile remains fixed from z = 0 to 20 mm (i.e. periodic), then its magnitude and phase change in the range 20 to 36 mm, and finally there is only phase change from 36 to 60 mm. The cross sectional view and its normalized power spectrum of the two-dimensional cam-shape profile at x = 0 mm are shown in Figure 5. The spectrum appears only at the fundamental frequency and its harmonics because the cam profile is periodic when the spindle rotates at a constant speed. The cross sections at x = 36 and 60 mm are rotated 60 and 36 degrees, respectively, clockwise from the cross section at x = 0 mm. In the experiment, the real-time reference signal was generated by

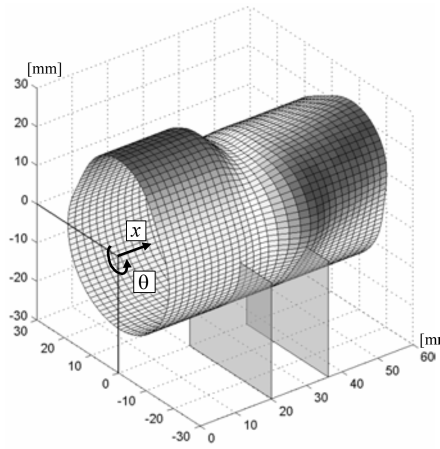

Figure 4 A two-dimensional cam-shape profile

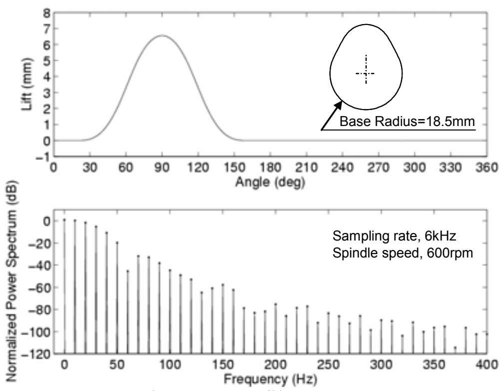

Figure 5 Profile at x = 0 mm

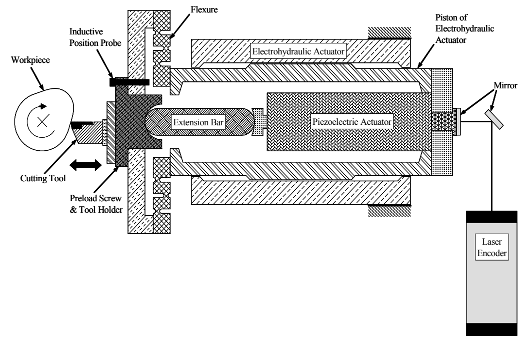

Figure 6 Electrohydraulic Actuator

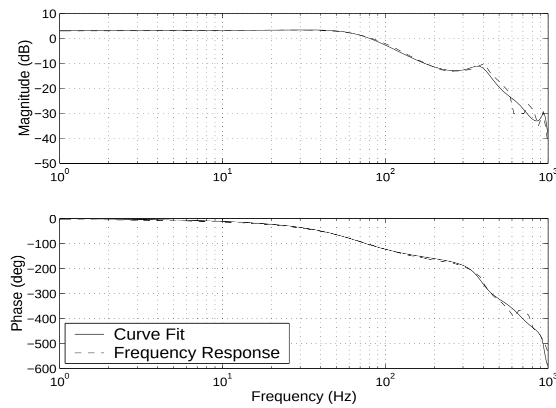

Figure 7 Frequency Response of the Hydraulic Servo

traversing this cam profile at a spindle speed of 600 rpm and a feed rate of 0.2 mm/sec, and 600 samples per spindle revolution, which corresponds to digital control sampling rate of 6 kHz.

Figure 6 shows the cross sectional view of the electrohydraulic actuator developed for the non-circular turning application. The actuator has a double acting and equal area piston and tapered hydrostatic bearings to support side loads applied to the actuator. The actuator is driven by a flapper nozzle type servovalve. Inside the hollow piston is a piezoelectric actuator used for dual-stage actuation [11]. Piezoelectric actuation will not be discussed in this paper. The actuator has an analog proportional feedback loop with an internal LVDT sensor. The actuator motion is measured by a laser encoder having a 0.63 μm resolution.

The servo valve and hydraulic actuator dynamics are slightly nonlinear due to flow-pressure drop relation through orifices. However, linear model $G_o(s)$ and its uncertainty bound $W_r(s)$ are required for the controller design. The effect of nonlinearity for different flow rates may be considered as perturbations to a nominal linear model. In [17] an effective method to determine $G_o(s)$ and $W_r(s)$ was proposed. A suitable range of input magnitudes, which cause various flow rates, were applied to the system to obtain outputs and corresponding frequency responses. These experimental frequency responses are then used to determine a nominal frequency response. A least squares fit of this nominal frequency response to a transfer function model renders $G_o(s)$, and the maximum residual errors with respect to all the frequency response data render $W_r(s)$.

According to the linearization of the servo valve and actuator's nonlinear dynamics, the nominal transfer function $G_o(s)$ has one unstable zero and eight stable poles [21]. The unstable zero comes from the servovalve, where a mechanical feedback spring connects the torque motor armature and the valve spool. The experimental averaged frequency response and its nominal model shown in Figure 7 have very close agreement by confining the model to this structure. This accurate dynamic model is critical to achieving high performance in the subsequent model based control system design.

The discrete nominal model $G_o(z)$ is computed from the continuous-time model using a zero-order-hold transformation. The magnitude Bode plots of $G_o(z)$, $M(z)$, $q(z,z^{-1})$, $1/W_p(z)$, $1/W_d(z)$, and $W_r(z)$ used in this design example are shown in Figure 8. N was 600 and L was chosen to be 10. The reference model M is selected as the following zero phase low-pass filter form:

$$M(z)=(0.25z+0.5+0.25z^{-1})^n. \qquad (16)$$

Its bandwidth reduces as n increases. The value n was 2 in the design and the same zero filter was used for the filter $q(z, z^{-1})$.

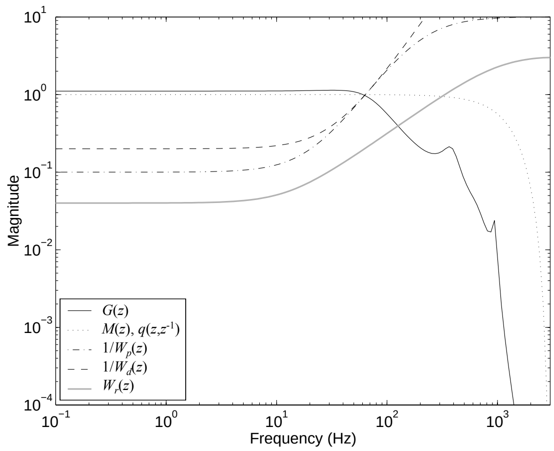

Figure 8 Weighting filters

Several sets of controllers based on μ-synthesis are designed with different preview length (F values). The error transfer functions e/r are compared for F = 2, 4, 6, 8, 10 in Figure 9. We can see deep notches whose ends are marked at the fundamental frequency and its harmonics. The notches are a distinctive characteristic of repetitive control providing high gain at these frequencies. It clearly shows that tracking error reduces as the preview length F increases. While increasing F theoretically and intuitively continues to reduce the tracking error, unmodeled dynamics and noise in the real system prevents reduction beyond a certain F value. In the experiment presented next $F=8$. Notice that all the tracking error transfer functions are below the upper bound $1/|W_p(e^{j\omega T})|$. Also, the disturbance rejection transfer functions d/e are all bounded by $1/|W_d(e^{j\omega T})|$ although the corresponding plots are not presented here. A two parameter robust repetitive control (TPRRC) using the above structure with the same design parameters described above but without the feedforward controller K_3 was also designed for comparison with the present controller. Figure 10 shows the resulting tracking error transfer function of TPRRC.

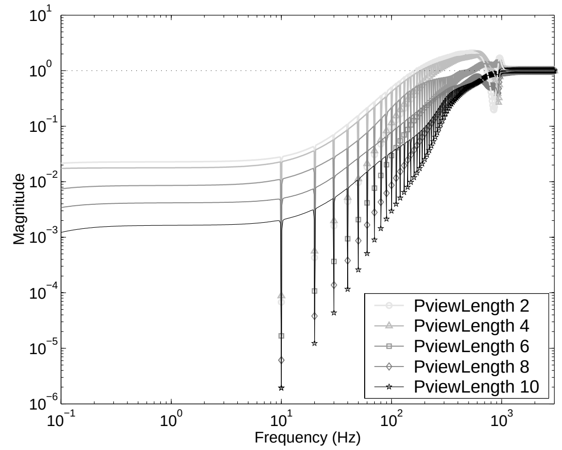

Figure 9 Transfer function of error/reference

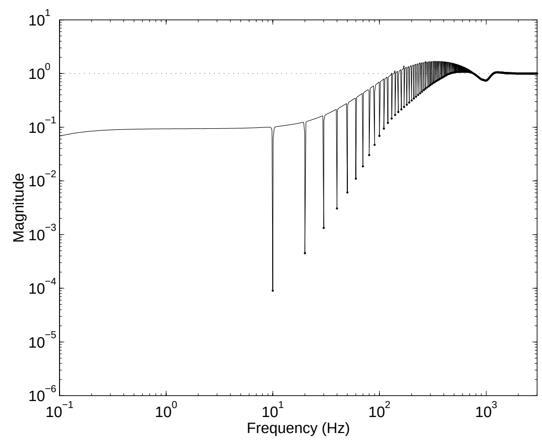

Figure 10 TPRRC error/reference transfer function without feedforward control

CONTROL IMPLEMENTATION AND EXPERIMENTAL RESULTS

The controller with the three control actions designed by the μ-synthesis method with a preview length of eight was initially in a 65th order state space form. For real-time implementation, controller order reduction was performed for each of the three input channels independently. The controller order was educed to 9th, 8th, and 13th for K_1, K_2, and K_3, respectively. The μ values from the reduced order controller were almost the same as those from the full order controller.

All the designed real-time controllers were implemented by a 32 bits floating point digital signal processor (TMS320C32). Finite word length (FWL) truncation error is another important factor in the fast sampling rate digital controller implementation. The 32 bits floating point DSP provides single precision floating point number representation, which has a finite precision of approximately 7 significant decimal digits and a finite range of 10^{-38} to 10^{+38}. The original high order controllers not only consume much computational time but also are vulnerable to FWL errors. Controller order reduction can be employed to overcome this situation. The output from $K_3(z)$ may be calculated off-line because it does not use feedback signal. The controller order reduction and numerical truncation effects on $K_3(z)$ was checked by comparing the off-line 64 bit double precision computation for the initial 65th order controller with the on-line single precision computation of the reduced order controller. The experimental tracking performance for these two cases was almost the same and justified the controller reduction approach. Similarly, a 45th order controller from TPRRC was reduced to a 9th and 8th order for K_1 and K_2, respectively. The details of the model reduction procedure adopted here can be found in [11].

The reference signal was generated from the two-dimensional cam-shape profile in Figure 4 with a spindle speed of 600 rpm and feed rate of 0.2 mm per spindle revolution. The experimental results of the control system are shown in Figure 11 in terms of RMS errors calculated per spindle revolution (600 samples). A few leading seconds were given to the actuator system until it reached its steady-state at x = 0. Due to the hardware limitations, the whole two-dimensional reference could not be imported into the C32 DSP. The cross sections of Figure 4 were specified at every 0.6 mm and linear interpolation was used to generate the real-time reference for the controller. The effect of the linear interpolation appeared as small ripples in the RMS error curve distinctively in the range of x = 20 to 36 mm. where both phase and magnitude of the cam profile varies.

Both controllers generated very small tracking error, where their RMS errors are less than 20 μm for the entire cam profile, which has over 6 mm lift. The integrated controller with all three control actions has consistently superior performance than the similarly designed TPRRC, which does not have feedforward action. The reference is near periodic at x > 20 mm and as such the previewed feedforward control action is very effective in reducing the tracking error as it is compared with TPRRC. Figure 12 shows the tracking errors at x = 10, 25, 50 mm of the two control design methods. The abscissa of each plot represents time in seconds. Note that at x = 10 mm the reference is purely periodic, at x = 25 mm the magnitude and phase change, and at x = 50 mm only the phase changes. As shown in the figure, the method with all three control actions has peak-to-peak errors well below ± 20 μm.

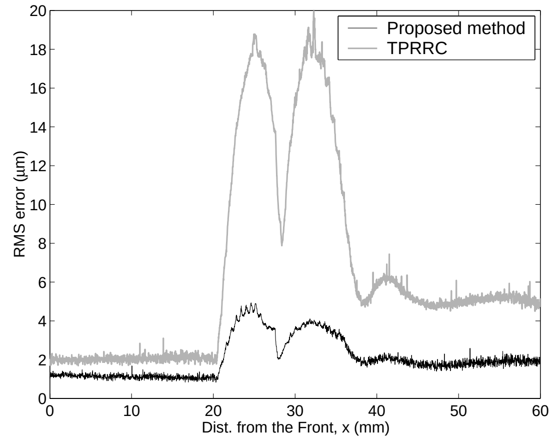

FIGURE 11 Tracking error (RMS value for every rotation) of the controller and the TPRRC

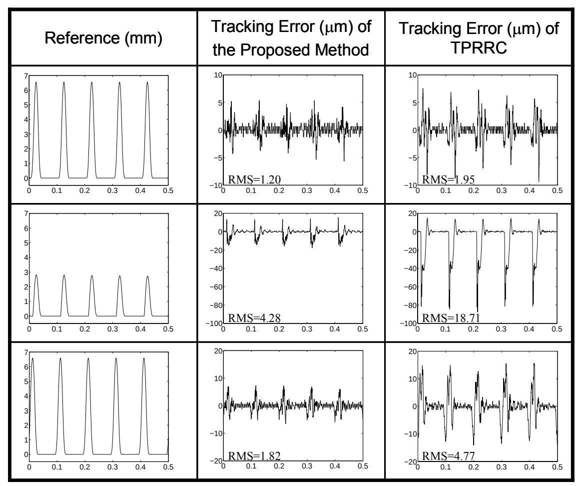

FIGURE 12 Tracking error of the controller and TPRRC at x = 10 (top row), 25 (middle row), and 50 (bottom row) mm

CONCLUSIONS

The control design problem for feedback, repetitive, and previewed feedforward control actions has been formulated in the LFT form and solved by μ-synthesis. In this way, the desired upper bounds of disturbance rejection and tracking error transfer functions, and the repetitive controller internal model, are explicitly included in the design process. Method to obtain the system dynamics and the uncertainty bound for electrohydraulic systems is also presented to facilitate μ-synthesis. Controller order reduction is essential for the real-time implementation in order to reduce computational time and the effect of finite word length truncation error. Experimental results on an electrohydraulic actuator have demonstrated the effectiveness of the design method and implementation technique.

There are situations where the nonlinearity in the system is too significant to be treated as uncertainty. In this case, the nonlineariy may be modeled and compensate for by nonlinear feedback approach. One such case occurs in dynamic material testing of nonlinear materials. In [23] a back stepping nonlinear feedback with repetitive control was developed and successfully implemented on an electrohydraulic actuator loaded with a nonlinear test material to precisely generate periodic motions. Another situation is that the system dynamics may change slightly due to hydraulic fluid temperature and trapped air content variations. Although the robust feedback control is insensitive to such changes, the feedforward control is. Adaptive control [2, 6, 24] has been successfully to electrohydraulic systems to maintain precise dynamic tracking performance under such changes.

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