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Brief paper

Composite jerk feedforward and disturbance observer for robust tracking of flexible systems ☆

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ABSTRACT

In this paper, a novel feedforward controller for flexible motion systems is proposed based on both the rigid-body mode and lump-sum of the flexible modes. It only requires the reference trajectory to be defined up to its jerk, and gives well-behaved high-pass feedforward sensitivity. For systems with severe low-frequency disturbance, an additional disturbance observer is introduced, using the same jerk feedforward as the inversion of the nominal model. Remarkably, with such proposed control scheme and novel loop-shaping criteria, the improvement of disturbance rejection and profile tracking does not result in obvious degrading of the noise attenuation.

1. Introduction

The design of the lightweight stage is becoming a trend to increase the throughput and improve the energy efficiency. However, poor settling performance may occur due to presence of flexible dynamics which is not considered in conventional high-inertia systems. Thus, the concept of "beyond-rigid-body control" is introduced, referring to the control design strategies which deal with the flexible modes in more rigorous manner compared with conventional rigid body control (Lunenburg, Bosgra, & Oomen, 2009). For continuous systems with a large number of flexible modes, the dynamics can be represented by the partial differential equations (PDE), where the boundary control can be subsequently applied (Balas, 1978; He & Ge, 2016; He, Ge, & Zhang, 2011; He, Zhang, & Ge, 2014a,b; Krstic, Guo, Balogh, & Smyshlyaev, 2008). For certain motion systems, the PDE models can be simplified to the lump-parameter models, whether the design methods based on classical control theory are applicable (Miu, 1993). As a complement to the feedback control, the feedforward control can effectively improve the response by adjusting the control signal according to the motion profiles (Liu, Tan, Chen, Teo, & Lee, 2013). The rigid-body feedforward (RBFF) from model inversion works effectively for the rigid-body systems. However, for high-order flexible systems, the direct model inversion is generally not applicable due to unavailable high-order derivative of profiles (Lambrechts, Boerlage, & Steinbuch, 2005), non-minimum-phase (NMP) zeros (Benosman & Le Vey, 2004) in the model and existence of model uncertainties.

Unlike the simple model reduction by truncation, the snap feedforward (SFF) is developed by taking care of the rigid-body dynamics and the lump-sum of the flexible dynamics in the low-frequency band (Boerlage, Tousain, & Steinbuch, 2004). The effectiveness of SFF control has been validated in a high-speed wafer scanner (van der Meulen, Tousain, & Bosgra, 2008).

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However, due to high feedforward sensitivity (S_f) (Lunenburg, 2010) at the high-frequency bands, using of SFF requires the reference to be ultra-smooth (Lambrechts et al., 2005). Meanwhile, existing data-based tuning method for such composite control structure are generally computational intensive (Butler, 2013; Heertjes, Hennekens, & Steinbuch, 2010), and the rigorous proof of convergence of tracking error is generally difficult (Hjalmarsson, 2002).

Remarkably, in the front-end semiconductor equipment (Butler, 2011), the direct-drive actuators are widely applied where the model uncertainty is small till a relative high frequency (van Herpen, Oomen, Bosgra, & van de Wal, 2009). Meanwhile, the indirect-drive actuators are widely used in the back-end semiconductor machine design (Huang, Li, Li, Chetwynd, & Gosselin, 2004; Thompson, 1997). However, for such actuators, besides the difficulties in controlling the general light-weight stages, the significant low-frequency disturbances always exist due to the mechanical coupling forces, while the performance of the feedforward controllers greatly relies on the accuracy of the models.

Notably, the disturbance observer (DOB) provides an effective way to cancel these disturbances online in a lump-sum manner (Schrijver & van Dijk, 2002; White, Tomizuka, & Smith, 2000). Remarkably, the prior work has proposed to design the DOBs in either the process with dead-time (Zhong & Normey-Rico, 2002), or rigid-body motion systems (Tan, Lee, Dou, Chin, & Zhao, 2003). Its application has been extended to deal with specific classes of nonlinear systems (Back & Shim, 2008; Chen, Ballance, Gawthrop, & O'Reilly, 2000; Shim & Jo, 2009) with higher-order or other nonlinear disturbances (Jamaludin, Brussel, & Swevers, 2009; Kim, Rew, & Kim, 2010). The sensitivity optimizations are performed to balance the disturbance response and robustness (Tesfaye, Lee, & Tomizuka, 2000; Zhong & Normey-Rico, 2002). The incorporations of the feedforward controllers with DOB have been reported for the rigid systems (Kempf & Kobayashi, 1999; Yan & Shiu, 2008). For flexible systems, the proposition of phase-lead compensator with incorporation of DOB is proposed in Katsura and Ohnishi (2007). However, to the authors' best knowledge, for incorporation of DOB with feedforward controllers in such flexible systems, no existing literature is reported with rigorous analysis of the overall performance.

The first contribution of this paper is to propose a model-based jerk feedforward (JFF) controller with tunable artificial damping factor ζ_d, which is compatible to the third-order trajectory planning in current industry practice (Lambrechts et al., 2005). Besides, it has the following advantages.

(i) There is no surge on $|S_f|$ near the equivalent resonant frequency ω_c, ensuring the good tracking performance till a higher bandwidth.
(ii) Its S_f presents a well-behaved high-pass property with unity high-pass gain.
(iii) $|S_f|$ can be lower than that of other low-order feedforward controllers at the low-frequency band, depending on the system's equivalent resonant damping ζ_c.

Theorem 1 summarizes the rules for tuning of ζ_d in this novel JFF to achieve better performance.

The second contribution is to extend this JFF to handle a class of flexible systems impeded by low-frequency disturbance. This is done by forming a 3-DOF composite control using the same JFF as the inverse of the nominal model for the DOB. In Theorem 2, it shows that with satisfying simple loop-shaping conditions, such 3-DOF control has the following advantages.

(i) Till the cut-off frequency of the low-pass filter (LPF) in the DOB, the performance on tracking accuracy and disturbance rejection capability will be superior over that by using the 2-DOF control. Beyond this frequency, such performance is typically not worse than that by the 2-DOF control.
(ii) Throughout the entire frequency range, the performance on noise attenuation is typically not worse than that by the 2-DOF control.

Simulation and real-time experiment validates the claim and shows the practical appeal of proposed control schemes.

2. A class of flexible systems impeded by low-frequency disturbance

2.1. Model description

Consider an SISO, $N+1$-mode flexible systems, given by

$$y = P(s)(u+d); \qquad y_m = y - n; \tag{1}$$

where y is the actual position output, u is the control input and d is the low-frequency input disturbance, y_m is the output measurement corrupted by the noise n, and $P(s)$, is the identified transfer function in the *modal summation form*.

$$P(s) = \underbrace{\frac{1}{m_t s(s+b)}}_{P_r(s)} + \underbrace{\sum_{i=1}^{N} \frac{k_i}{m_t(s^2 + 2\zeta_i\omega_i s + \omega_i^2)}}_{P_f(s)} \tag{2}$$

$P(s)$ can be directly derived either from the modal decomposition, as given in Appendix A, or from the transfer function curve fitting (Zheng, Guo, & Wang, 2005). Here, m_t is the total moving mass (Miu, 1993), b is the rigid-body damping frequency, k_i, ω_i and ζ_i are ith modal gain, natural frequency, and damping accordingly. Notice that in (2), the $N+1$ modes are combined linearly, and the in-phase and out-of-phase behavior of the resonant modes are easily seen from the sign of k_i (Munnig Schmidt, Schitter, & van Eijk, 2014). This is useful to mechanical engineers throughout the design, simulation, optimization and validation process.

2.2. Timing-belt stage—a case study

Fig. 1 shows the testbed of an industrial grade vee-teeth timing-belt actuator, which is taken from a tray indexing module in a laser marking machine (Marking Solutions, 2015). Its frequency response test at a single point is given in Fig. 2, excited by pseudo-random binary sequence with different levels of maximum input torque. The phase delay at high-frequency band, due to insufficient sampling rate of the driver during Bode-plot test, is ignored in the closed-loop control experiment when higher sampling rate is used. Remarkably, the magnitude response in low-frequency band varies with the input torque. This is probably due to the frictional disturbance d, which is treated as an input multiplicative uncertainty $d = \Delta(s)u$, so that $y = [1 + \Delta(s)]P(s)u$. And, from Fig. 2, $\Delta(s) = \frac{b-\underline{b}}{s+\underline{b}}$, while $\underline{b}$, the actual rigid body damping frequency, is unknown. Since u is band-limited signal from PID feedback–feedforward (2-DOF) controller, and $\Delta(s)$ is a low-pass filter, d is indeed low-frequency disturbance. Curve fitting is performed based on the result from 50% maximum test torque, which is also shown in Fig. 2. The transfer function is given by the form of (2) with parameters $m_t = 0.0133$, $b = 8.0$, $k_1 = 3.325$, $k_2 = 0.665$, $\zeta_1 = 008$, $\zeta_2 = 0.01$, $\omega_1 = 971.32$ and $\omega_2 = 9458.3$, and the whole system, including the disturbance and measurement noise, can be described in the form of (1). We can see that although the first resonant frequency is as high as 1000 rad/s, the first anti-resonant frequency is comparably lower at 400 rad/s.

Thus, in such a flexible system, to utilize the bandwidth being lower than the first anti-resonant frequency, two fundamental issues need to be addressed. The first issue is to approximate the dynamic behavior below the first anti-resonant frequency with a more accurate, but low-order model. This is equivalent to design a