PDFF: An evaluation of a velocity loop control method.

George Ellis
Kollmorgen
201 Rock Road
Radford, VA 24141, USA

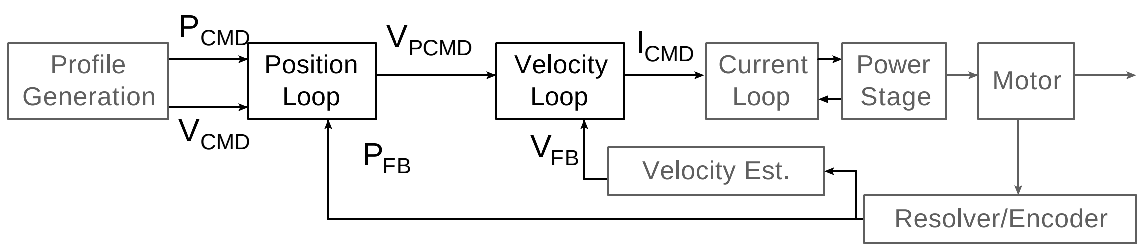
Email: gellis@kollmorgen.com

Abstract

High-performance electric motors and their associated controllers are widely used in motion control applications. Feedback devices are employed to measure position and estimate motor velocity. The control system closes position and velocity loops using these feedback devices. A number of control schemes have been employed in industrial applications including closing a velocity loop within a position loop. Here, the velocity loop is often a Type I (integrating) loop that is enclosed within a proportional position loop. In this case, proportional-integral (PI) compensation is commonly used for the velocity loop. PI control has two advantages: it is simple and it supports rapid motion. However, it also has two shortcomings: it has a tendency to overshoot and it provides low DC stiffness. This paper details a compensation algorithm called PDFF which modifies PI, allowing the user to eliminate overshoot and provide much more DC stiffness than PI.

Introduction

High-performance electric motors and controllers are widely used in servo motion control applications such as machine tools, packaging, printing, web handling, robots, textiles, and food processing. Feedback devices such as encoders and resolvers are employed in such systems to measure position and estimate motor velocity [Br]. The control system usually closes position and velocity loops using these feedback devices. A number of control schemes have been employed in industrial applications including cascading a velocity loop within a position loop. Here, the velocity loop is often a Type I (integrating) loop that is enclosed within a proportional position loop, usually with a velocity feed-forward term. A proportional-integral (PI) velocity loop compensator is commonly used for the velocity loop. PI control has many advantages: it is simple and able to support rapid motion. However, PI velocity control has several shortcomings including a tendency to overshoot and low DC stiffness.



Typical Motor Control System
Figure 1

This paper describes a method of velocity loop compensation called PDFF that extends PI by modifying the control algorithm to reduce overshoot; this allows greater DC stiffness, although it comes at the cost of reduced responsiveness. This paper begins with a brief discussion of PI control and then provides theoretical development of PDFF.

A tuning procedure is developed which yields responsive, stable performance in a non-iterative process (that is, each gain is adjusted just once). This procedure is used to set each of three velocity loop gains, one by one. The only equipment required is an oscilloscope capable of measuring velocity command and feedback. Note that many modern digital drives include a software oscilloscope so that frequently no hardware is required. This paper will also present simulation and laboratory results for the PDFF controller exercised over a broad range of operation. The goals of this paper are to provide a theoretical development and quantitative analysis of PDFF and to guide potential users in the selection of and expectations for the method.

Background

Closed-loop motion control systems employ a variety of loop configurations. One common configuration is to enclose a cascaded velocity loop within a position loop, as shown in Figure 1. The velocity loop output feeds an independent closed-loop current controller which creates torque in the motor. Position is fed back from a position sensor such as an encoder or resolver. Most modern systems estimate velocity from the position signal.

Several alternative configurations are also used in industrial applications. Sometimes the velocity loop is non-integrating. PID position control eliminates the explicit velocity loop although these systems use a term based on the derivative of the position error which is, of course, velocity error. These alternative configurations are used in industry and are generally equivalent. For the remainder of this paper, the focus will be cascaded-loop systems: integrating velocity loops contained within proportional position loops.

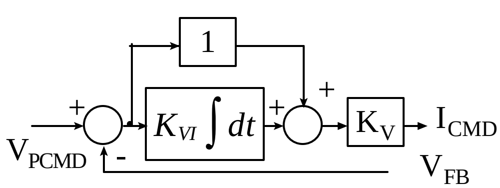
Velocity Loop

The velocity loop resides between the position and current loops as shown in Figure 1. The most popular velocity-loop configuration is the proportional-integral (PI) loop as shown in Figure 2. Note that in some cases, the integral and proportional terms are scaled independently and then added. These two methods are mathematically equivalent although the units for K_{VI} change. The output equation here is:

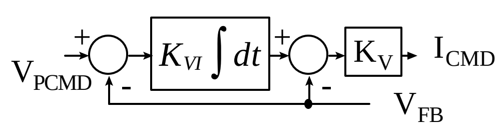
$$I_{CMD} = (K_{VI}\int (V_{PCMD} - V_{FB})dt + V_{PCMD} - V_{FB}) * K_V \qquad \text{Equation 1}$$

An alternative velocity loop configuration used in split-loop systems is the Pseudo-Derivative Feedback (PDF) loop [Ref. 4] as shown in Figure 3. The output equation is:

$$I_{CMD} = (K_{VI}\int (V_{PCMD} - V_{FB})dt - V_{FB}) * K_V \qquad \text{Equation 2}$$



PI Controller
Figure 2



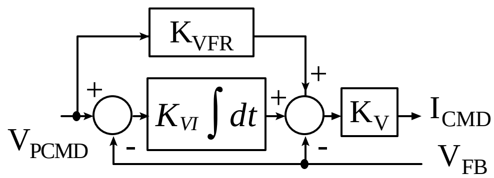
PDF Controller
Figure 3

PI and PDF are similar in that both scale the integral of velocity error and both have an output scaling term (K_V). The key difference is that PDF forces the entire error signal through an integration. This makes PDF less responsive to the velocity command than PI. However, the change in structure allows PDF to have higher integral gains and thus responds better to low-frequency torque disturbances than PI.

PDFF Velocity Loop

PDF can be augmented with a feed-forward term to create PDFF as shown in Figure 4. The feed-forward term, scaled by K_{VFR}, injects the command ahead of the integral making the system more responsive to commands. The output equation is:

$$I_{CMD} = (K_{VI}\int(V_{PCMD} - V_{FB})dt + K_{VFR} * V_{PCMD} - V_{FB}) * K_V \qquad \text{Equation 3}$$



PDFF Controller
Figure 4

Upon inspection of Equations 1, 2, and 3, it can be seen that PDFF is a general controller that includes PDF and PI control as two cases (when K_{VFR} is 0% and 100%, respectively.) While K_{VFR} can be set to these two extremes, it can also be set anywhere in between. The question that will be taken up in this paper is: How does one set K_{VFR} for a particular application?

Analysis of PDFF

This analysis will provide side-by-side comparisons of PDFF for multiple values of K_{VFR}. It is possible to determine some characteristics by inspection. For example, the PDF loop (Equation 2, or Equation 3 with K_{VFR} = 0%) passes the command entirely through an integrator, where the PI loop (Equation 1, or Equation 3 with K_{VFR} = 100%) transfers the command without integration through the proportional gain. Because the error avoids the integral, one would expect that PI would be more responsive to the command and that is the case. In fact, the most responsive applications are normally satisfied with K_{VFR} = 100%, which is equivalent to simple PI.

The responsiveness provided by PI comes at a cost. Generally, PI velocity loops cannot be tuned to provide high DC stiffness. DC stiffness is a characteristic demanded in applications that must hold position or follow command in the presence of low frequency disturbances such as gravity or friction. For example, DC stiffness is valuable when the application involves a worm gear, a mechanism that usually has high friction. High DC stiffness here allows the controller to pull into position quickly, even in the presence of the frictional load. DC stiffness is proportional to the integral gain. Unfortunately, PI control suffers from high overshoot to a step when the integral gain is high. When K_{VFR} is low, this problem is cured. Because most of the error signal is filtered by the integral, overshoot is reduced and the integral gain can be raised without causing overshoot to a step. With PDFF, the user has the choice of making the system very responsive (K_{VFR} = 100%), very stiff at low frequencies (K_{VFR} = 0%), or anything in between.

Many other characteristics of servo systems are unaffected by the selection of K_{VFR}. For example, mechanical resonance, a problem caused by compliant coupling between the motor and load inertias, is almost entirely unaffected by the integral gain. The loop gain K_V is the only servo gain that has much impact on resonance. This is intuitive as resonance is a high frequency effect, well beyond the influence of the integral gain in most systems.

Another characteristic of servo systems is the acoustical noise generated, especially by the most responsive systems. This noise is usually caused by one of two problems: either there are problems in the system wiring or the feedback device does not have sufficient resolution. Both problems are exaggerated by high loop gains. In either case, noise usually couples through the position sensor into the velocity estimation, then through the loop gain K_V and to the current command. Like resonance, noise from poor wiring or from inadequate resolution is a high frequency effect and is also outside the influence of the integral gain. In fact, comparisons in the laboratory revealed that the selection of K_{VI} or K_{VFR} have virtually no impact on noise or resonance.

A method for quantitative comparison

In order to evaluate the effects of K_{VFR}, many sets of tuning gain were derived according to a consistent set of criteria. The criteria were selected based on experience with velocity controllers:

- Limit overshoot to a step command of 10%.
- Have no noticeable effects of mechanical resonance.
- Produce equivalent effects of resolution and wiring noise for all tuning gain sets.

While the value of 10% overshoot is somewhat arbitrary, it is consistent with current industry practices. While few motion control systems are actually subjected to a step command, this measure is still important to help quantify stability. In addition, the results of experiments do not change significantly if the acceptable overshoot is adjusted over a reasonable range and all tuning sets are required to meet the same limit.

Resonance and resolution noise affected by K_V

As expected, laboratory experiments demonstrated that mechanical resonance and resolution noise were caused almost solely by K_V. Effects of resonance were evaluated with a load designed to allow changes in mechanical compliance (Figure 5). Because of this, all tuning gains are based on the same K_V. This keeps the effects of resonance and noise the same for all sets of tuning gains. The value of K_V was selected by converting the control loop to a proportional loop (K_{VFR} = 100%, K_{VI} = 0) and raising K_V until there was noticeable overshoot. The selected value was 5600 for all experiments. Other means of selecting K_V (for example, allowing some overshoot) have little bearing on the results as long as the value of K_V is reasonable and all tuning gain sets use the same value.

Stiffness

Stiffness rates the ability of the system to follow a command in the presence of a disturbance. We have already discussed DC stiffness, a special case which considers only the response to low-frequency disturbances such as friction. Other disturbances occur at higher frequencies. For example, in machine tools, a disturbance is created each time a cutting surface contacts a work piece; this often occurs at frequencies above 50 Hz, well above the effects of the integral gain. A broad study of dynamic stiffness is beyond the scope of this paper; however, simulations and analysis reveal that at low frequencies, stiffness is affected by the integral gain, and at frequencies beyond the bandwidth of the controller, the stiffness is provided by the system inertia. In the middle-range frequencies, stiffness is provided predominantly by the loop gain, K_V. Since our comparison will only impact the integral gain, all tuning sets provide approximately the same mid- and high-frequency stiffness, and DC (or "low frequency") stiffness is well known to vary in proportion to the integral gain.

Adjusting K_{VFR} and K_{VI}

What remains is selecting a variety of values for K_{VFR} from 0% (PDF) to 100% (PI), raising K_{VI} as much as possible without exceeding the 10% step response overshoot limit. Then each tuning gain set can be evaluated for DC stiffness and responsiveness. The measure of responsiveness was selected as the bandwidth, the frequency of sine wave command at which the velocity loop response falls to 70% (-3dB) of the very low frequency gain.

Laboratory Experiments and Simulations

Analysis of PDFF relied on a combination of laboratory experiments and simulations. For the experiments, the Kollmorgen SERVO**STAR**® amplifier and GOLD**LINE**™ motor system were selected (Figure 6). A model was developed in a time-based modeling environment called ModelQ. A stand-alone executable version of this model is available from the author (e-mail gellis@kollmorgen.com and put "PCIM-99 ModelQ" in the message title). The ModelQ environment provides time-based and frequency-based information to allow thorough evaluation of the simulated system.

Validation

Before simulation data was accepted, the model was verified using a motor/drive system. Measurements were done using an oscilloscope internal to the drive, and a SigLab Model 20-42 frequency analyzer by DSP Technologies (www.dspt.com). Note that in Figure 6, the drive and motor set are on the left; the SigLab frequency analyzer is on the right beneath a laptop computer.

The tuning constants for a typical velocity loop were entered into the motor drive and into the model. The closed-loop command response was measured from the actual system and from the model and then compared. The gains of the two plots did not differ by more than a few tenths of a dB below 100 Hz; the phase did not differ more than two degrees over that same range. Between 100 and 200 Hz, gains were equal within about one dB and phase was accurate to within 350 μsec. The model was judged valid and was used for system measurements.

Motor with variable-compliance load used to verify that K_V is primarily responsible for resonance
Figure 5

*Photograph of test setup including Kollmorgen SERVO**STAR**Òamplifier and GOLD**LINE**™ Motor and SigLab 20-42 Dynamic Signal Analyzers*
Figure 6

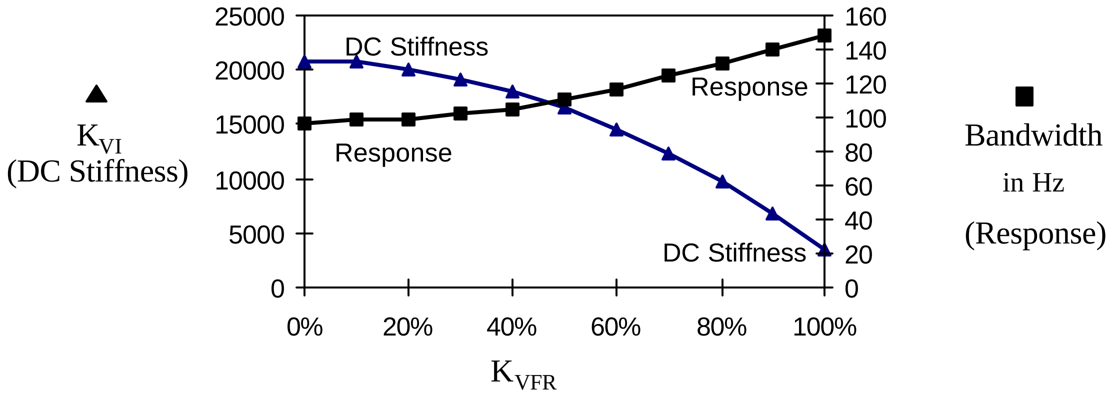
Results

Given that K_V = 5600 as stated above, the remaining step was to adjust K_{VI} from 0% to 100% in 10% increments and then adjust K_{VI} for 10% overshoot. Bandwidth was measured in the simulation. Table 1 provides the results of this test:

K_{VFR}	0% PDF	10%	20%	30%	40%	50%	60%	70%	80%	90%	100% PI
K_{VI}	20800	20800	20000	19200	18100	16600	14600	12400	9700	6800	3500
Bandwidth (Hz)	96	99	99	102	105	111	117	125	132	140	148

Table 1. Tuning gains with resulting bandwidth for various values of K_{VFR} with K_V=5600.

Table 1 reveals two important characteristics of PDFF control. First, as K_{VFR} increases, the responsiveness increases. Here, the bandwidth increased more than 50% over the range of K_{VFR}. Second, the stiffness of the controller declines as K_{VFR} increases; here the stiffness declined by a factor of almost six. The two popular methods (PI and PDF) are extremes of this continuum. With PDFF, the designer can select either extreme or any position in the middle. PDFF allows users to select K_{VFR} according to the needs of the application. This is plotted in Figure 7.



Responsiveness increases and DC stiffness declines as K_{VFR} increases.
Figure 7

PDFF and Position Control Systems

The benefit of PDFF, that designers can choose between responsiveness and DC stiffness, appears to extend beyond the velocity loop to the full positioning system of Figure 1. While quantitative analysis is not available as to how PDFF impacts the position controller, the author's experience has been that velocity loops with higher responsiveness or greater DC stiffness transfer those qualities to the position controller.

Conclusions

The conclusions resulting from this work are:

- K_{VFR} impacts both response and DC stiffness. Over the entire range of K_{VFR} , response increases in a velocity loop controller by about 50% and DC stiffness declines by about six times.
- PI is superior for highly responsive systems. PDF is ideal for applications requiring the maximum DC stiffness. PDFF allows the user to select for applications that are not satisfied by either extreme.
- Resonance and noise from resolution are exaggerated by high values of K_V. They are not impacted significantly by integral gain.

References

1. Biernson, G., Principles of Feedback Control, Volume I, Wiley, 1988, p 369.

2. Brown, H., Schneider, S., and Mulligan, M., "Analysis of Algorithms for Velocity Estimation from Discrete Position Versus Time Data," IEEE Transactions on Industrial Electronics, Vol. 39, No. 1, Feb 1992.

3. Schmidt, P.B., and Lorenz, R. D., "Design Principles and Implementation of Acceleration Feedback to Improve Performance of DC Drives," IEEE Trans. on Ind. Appl., May/June 1992, pp 592-599

4. Phelan, Richard M., Automatic Control Systems , Cornell University Press, 1977.

5. Vukosavic, S., Stojic, M., "Suppression of Torsional Oscillations in a High-Performance Speed Servo Drive," IEEE Transactions on Industrial Electronics, Vol. 45, No. 1, Feb. 1998.