The Effect of Time-Delay and Discrete Control on the Contact Stability of Simple Position Controllers

John Fiala, Ronald Lumia
National Institute of Standards and Technology
Bldg. 220, Rm. B127
Gaithersburg, MD. 20899, USA

Abstract

By analysis of the driving-point admittance, it is shown how time-delays and discrete control can create instabilities for a simple position controller in contact with the environment. The lowest frequency of contact instability due to time-delay or sampling is determined analytically. It is shown how mechanical compliance between the motor and point of contact can eliminate these instabilities. To achieve the best relative stability when contacting arbitrary environments, the mechanical/control design of manipulators should maintain a critical relationship between the frequency of the compliant mode and a frequency associated with contact instability.

1. Introduction

The simple proportional-derivative (PD) controllers used for controlling most robots show a remarkable robustness in a number of tasks including those which involve contact with the environment. Recently, some authors have noted that the time-delays and sampling in these controllers should have a detrimental effect on stability during contact with certain environments [1,2]. However, since such instabilities are not often observed in practice, these authors conclude that inherent mechanical compliance ultimately stabilizes these systems. The idea that the robustness of simple position controllers in contact is due to mechanical compliance is analyzed in detail in this paper.

First, consider a force generator (motor) $u$ acting on a damped mass, as shown in Figure 1. An ideal proportional-derivative position control law

$$u = K_p (x_d - x) - K_v$$

(1)

with $K_p, K_v > 0$, would produce a stable system, both in free-space and in contact, since it has a passive physical equivalent [1]. That is, the same control could be achieved by attaching a physical spring and damper to the mass, and physical systems are always passive or energy dissipating. Alternatively, one could examine the passivity of the driving-point admittance of the mechanical system. If the driving-point admittance does not have more than 90° of phase shift, then there is a passive physical equivalent and the system will be stable in contact with any passive environment [1,2]. To investigate stability it is only necessary to look at the admittance due to feedback [2], so for (1), the driving-point admittance is

$$Y(s) = \frac{-s}{s + (JwB + K_v s)}$$

(2)

The phase of $Y(\omega)$ for this system is between 90° and -90° for all $\omega > 0$, and the system is stable in contact with arbitrary environments.

Figure 2. Effect of time-delay on the passivity of the damped mass system.

Without friction, the frequencies delimiting the passivity violations are the intersections of the line $k_v o k_p$ with $\tan(\omega)$, as depicted in Figure 2(b). The frequency of the first such intersection is the frequency $\omega$ at which passivity is first violated. The presence of friction may stabilize the system. The amount of damping required to completely stabilize the contact instability can be used as a measure of relative stability. For the system of (3), damping can stabilize the system only at unreasonable levels. Suppose substantial friction of $B=K_v$ exists. Then (4) can be written

$$\frac{k_v}{k_p} \geq \frac{\sin(\omega)}{1 + \cos(\omega)} = \frac{\omega}{2}$$

(5)

which indicates that contact instabilities still exist but for stiffer environments. Note also that as $B$ increases, so does $\omega$. Only when $B \gg K_v$ can the system be completely stabilized. Such high levels of friction clearly limit performance.

For most robots there is some compliance between the motor and the point of contact [3]. The presence of this compliance may help stabilize the system during contact. The damped mass model can be
modified to include a compliant transmission modeled as a damped spring [1], as shown in Figure 3. To simplify initially, let $B=0$. Computing $\text{Real}(Y(j\omega))=N(\omega)/D(\omega)$, yields
\begin{equation}
N(\omega) = 16\epsilon_l^2 K_p^2 \cos (T\omega) - 16\epsilon_l^2 K_p \sin (T\omega) + 16\epsilon_l^2 \omega^2
\end{equation}
Since $D(\omega)>0$ for $\omega=0$, as before, the system will be stable in contact with arbitrary passive environments provided $N(\omega)\geq0$. This produces condition (4) again, and apparently the system with compliance will suffer the same type of passivity violations as before. However, the effect of the compliance is to improve the relative stability of the system by requiring less damping at certain frequencies.

The undamped natural frequencies of the system are
\begin{equation}
\omega_{n1} = \frac{2K_p}{J}, \quad \omega_{n2} = \frac{4K_p}{J}
\end{equation}
At contact frequencies below $\omega_{n1}$ the masses move together with the spring unstretched [4], such that the transmission damping has little effect. At these frequencies the situation of (4) applies, which requires significant motor-side friction $B$ to stabilize. Above $\omega_{n2}$ the masses move out of phase and the system may be stabilized by a smaller amount of friction $B_i$ in the transmission. In order to have a system with the best relative stability, i.e., which can be stabilized by the least amount of friction, the requirement is
\begin{equation}
\omega_i < \omega_p
\end{equation}
where
\begin{equation}
\omega_p = \frac{4K_p}{J}
\end{equation}
This relation can be achieved by decreasing the time-delay, which increases $\omega_p$ or by decreasing the stiffness of the transmission $K_p$. Although the presence of friction will also affect these values, a system design maintaining $\omega_i < \omega_p$, assuming no friction, will retain good relative stability in the actual system.

As an example, Figure 4 shows the phase of $Y(j\omega)$ for the system of Figure 3 with $J=1$, controlled by (1) with a time-delay of $T=0.01$, and $K_p=360$, $K_v=27$. When $B=0$, $\omega_i=148.1$ rad/s, as obtained from the graph of Figure 2(b). When $K_p=3000$, $\omega_i < \omega_p$ and the system is made completely passive by transmission damping of $B_i=0.55$. On the other hand, when $K_p=60000$, the system requires $B_i=100$ ($B=0$) or $B_i=0$ ($B=0$) for passivity.

3. Effect of Discrete Control

A computer implementation of (1) will introduce sample and hold elements into the control law. For a discrete controller with a zero-order hold
\begin{equation}
u = \frac{1-e^{-ht}}{ht} (K_p (x_d-x) - K_v v)
\end{equation}

Here it is assumed that any computational delays are small compared to $h$. Following the same reasoning as before, the positive real condition for this controller reduces to
\begin{equation}
K_p \cos (h\alpha_0) + K_v \cos (h\alpha_0) + B h^2 \alpha_0^2 - \alpha_p \geq 0 \quad (h \neq 0)
\end{equation}
which, when $B=0$ and $\alpha < \alpha_i$, can be written
\begin{equation}
\frac{K_p \alpha_0}{K_v} \leq \cos \left(\frac{h\alpha_0}{2}\right)
\end{equation}
This shows that the effect of discrete control on system passivity is the same as the effect of time-delay. Likewise, the effect of compliance between the motor and the point of contact is to improve the relative stability provided the condition (7) is maintained.

A position controller with an analog velocity loop
\begin{equation}
u = \frac{1-e^{-ht}}{ht} (K_p (x_d-x) - K_v v)
\end{equation}
will generally not exhibit contact stability problems since the positive real condition is
\begin{equation}
K_p \cos (h\alpha_0) - 1 + h K_v \alpha_0^2 \geq 0 \quad (h \neq 0)
\end{equation}
and this condition is only violated when $h K_v > 2 K_p$. Contact instabilities in simple position controllers are principally due to delays in the velocity feedback loop.

As another example, consider the controller depicted in Figure 5 applied to the damped mass system. The positive real condition yields
\begin{equation}
\frac{2K_p^2 \cos^2 (h\alpha_0) - (2K_v + K_p) \cos (h\alpha_0) + (K_v - h K_v^2 \alpha_0^2)}{4K} \leq \cos (h\alpha_0) \leq \frac{2K_p + K_p}{4K}
\end{equation}
From which we obtain the condition
\begin{equation}
\frac{(2K_v + h K_p) - \Gamma}{4K} \leq \cos (h\alpha_0) \leq \frac{(2K_v + h K_p) + \Gamma}{4K}
\end{equation}
\begin{equation}
\Gamma = \frac{8B h^2 \alpha_0^2 K_v + (2K_v - h K_v)^2}{2}
\end{equation}
As with the previous discrete controllers, there are multiple frequency intervals of phase in excess of $-90^\circ$. The intervals above the first interval of passivity violation, however, have relatively little phase excess, as shown in Figure 6. The figure shows the phase for $h=0.01$, $K_p=360$, and $K_v=27$. From (15) with $B=0$, the first interval of passivity violation begins at the frequency
\begin{equation}
\omega_p = \frac{1}{h} \cos^{-1} \left(\frac{K_p}{2K_v}\right)
\end{equation}
The minimum damping $B$ required for passivity can be obtained from (15) as well. Large amounts of damping may be required depending on the gains and sampling rate chosen. For the example of Figure 6, $B=13$ is required to make the system completely passive.

As with the time-delay problem, the controller of Figure 5 has the same positive real condition (14) when a transmission compliance is inserted as in Figure 3. Thus, as before, the presence of transmission compliance will improve the relative stability when $\omega_i < \omega_p$.
4. Effect of Discrete Control with Torque Loop

The effect of compliance external to the controller was examined in prior sections. Some modern manipulators, however, use drive transmissions surrounded by torque loops [5]. This approach places the major mechanical compliance of the device inside the controller. To see how this affects stability, consider the model of Figure 7. A spring representing the drive compliance has been inserted between a motor mass and a damped load. The position and velocity at the load are controlled using the equivalent of a torque loop, which attempts to control the force in the spring. The analog force control law is given by

$$\gamma = K (f_d - f)$$  \hspace{1cm} (17)$$

where the desired force $f_d$ is the input $u$ from the position controller in Figure 5, and $f$ is the sensed force produced by the transmission spring $K_t$. The resonant frequency of the force loop is

$$\omega_r = \frac{K_t (K + 1)}{J_m}$$  \hspace{1cm} (18)$$

Applying the positive real condition when there is no damping, $B_m=0$, $B_t=0$, and $B_l=0$, we have

$$\omega^2 + 2J_m\omega + (K + K_t)^2 \geq 0$$

The frequency (16) is still a determining boundary of the first instability interval, but the actual frequency interval is given by

$$\omega_p \leq \omega \leq \omega_r$$  \hspace{1cm} (19)$$

or

$$\omega_r \leq \omega \leq \omega_p$$  \hspace{1cm} (20)$$

Thus, for the torque loop model, there is a prominent first region of contact instability. For the first case (19), the system may be passive in the upper part of the region provided that $\omega_0$ is sufficiently far from $\omega_p$ to hold the entire first instability interval starting at $\omega_p$. In this case, the relative stability of the system in contact is similar to that of the damped mass system—a large amount of friction is needed to stabilize the system. Damping requirements will also be large when the system is unstable at $\omega_0$, which can be a problem for (20) as well. This is due to the resonant frequency term

$$J_m\omega_0^2 - (K + 1)K_t$$  \hspace{1cm} (21)$$

in the positive real condition. For a given $\omega_0$, this is most problematic when $J_m$ is small, since $J_m$ determines the steepness of (21) around $\omega_0$.

Again, $B_l$ is most effective in stabilizing the system for (20), rather than (19). In general, some damping is needed to stabilize the higher frequency non-passive intervals. $B_l$ will work as well as $B_t$ for this. However, there is another way to stabilize the prominent first interval, provided $J_m$ is not too small, as is illustrated in Figure 8. The following base set of parameters is chosen for the figure: $B_m=0$, $B_t=0$, $B_l=1$, $J_m=2$, $K_t=12000$, $K_p=360$, $K_v=27$, $h=0.01$. With $K=7$, the situation is that of (19), and the phase of the driving-point admittance is shown as the solid line in Figure 8. Only with damping of $B_m>19$ and $B_t=0.1$, can this admittance be made completely passive. An example of (20) can be obtained by using $K=1$. The result is shown in Figure 8 as the dashed line. Note that the passivity violation for this case is due to phase in excess of 90°, rather than -90°. Thus, the system can be made passive by the addition of phase lag in the PD loop. Since the controller will typically contain some phase lag due to computational delay, it is possible to design the system so that it is completely passive without excessive damping provided $\omega_p < \omega_0$. For this example, lag due to a computational delay of $T=0.0035$ and $B_l=0.15$ makes the system stable in contact with arbitrary environments.

5. Conclusions

Simple position controllers may exhibit contact instabilities, particularly when low sampling rates and long time-delays are involved. Most commercial robot controllers are stable in contact because they are typically controlled with analog PD controllers. When commercial systems use discrete control, they use sampling rates as high as possible. This generally ensures that the frequencies of passivity violations are above the resonant mode of the mechanical structure, $\omega_P > \omega_0$. As shown in this paper, achieving this relationship for these two critical frequencies makes contact stability much easier to achieve.

6. References

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