Control System Techniques—Dampers (Part 2)

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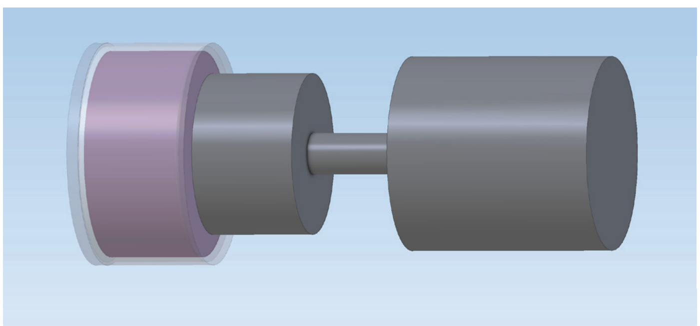
Closed-loop control systems can handle a wide range of motions with a wide range of loads if the control system and the mechanics of the system are properly designed for the task. A couple of the more difficult combinations to design for are high inertial mismatches and backlash with hard gearing. The question is not just how to make the system stable but also how to get the desired performance.



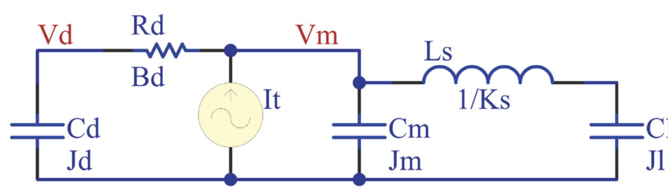


Figure 1—Damper attached to motor inertia, shaft, and load inertia (left). Electrical model of damper with motor inertia, torsion spring and load inertia (right).

Physical Viscous Inertial Damper

In "Control System Techniques—Dampers (Part 1)" [*PTE*, Vol. 18, No. 7; October 2024] we showed how adding a mechanical damper to a motor/load mass section reduces the resonance peaking in the system. In addition to reducing the peaking, a damper raises the system's phase margin. The combination of reduced resonance peaking with more phase margin allows the gain of the system to be increased significantly. We will get back to this, but more gain allows for wider bandwidth and tighter control of the load. More gain/bandwidth is also one of the fundamental ways to deal with stiction.

We will continue with the motor/damper/high inertia system. A physical viscous inertial damper (Figure 1) is shown mounted to the motor inertia (stiff compared to the viscous coupling of the grease), with the 100:1 inertia attached to the motor via a shaft. The impedance of the damper is to the left of the current (torque source), while the motor inertia, shaft torsion, and load inertia are shown to the right of the torque source in both the schematic and physical representations.

Whenever the motor velocity is greater than the damper inertia velocity, shear occurs in the viscous oil coupling them inside the damper, and the net torque available to accelerate the motor and load is reduced. Similarly, whenever the damper inertia velocity exceeds the motor velocity, the damper supplies torque into the system. Rapid changes in velocity, such as due to resonance, cause more shear and greater dissipation of vibrational energy into the viscous grease, removing the vibrational excitations from the system.

The improved gain and phase margins of the system allow for higher gains, reduced error and higher speed operation (wider bandwidth). The undesired part is that the damper is typically on the order of half or more of the size of the motor, and often a couple of times more expensive!

Synthetic Inertial Damper

This improvement in system performance without the added cost was the basis of wanting to simulate the viscous inertial damper in software. In electronics, there is a concept of Norton to Thevenin Equivalent circuits. In this transformation, a current source (or torque in the mechanical system) and the impedance representing the motor and load inertias, and shaft spring constant, can be converted to a voltage (mechanical velocity) with the impedance in series. When an additional load is then added, the current through the load (torque coupled to the mechanical damper) can be calculated as the voltage (velocity) divided by the sum of the motor/shaft/load series impedance and the added damper impedance (Thevenin equivalent circuit). In software, we can (real-time) simulate the torque that would be needed to accelerate the damper inertia to the motor velocity, given the measured motor velocity. The torque so estimated can then be subtracted from the commanded torque to the motor (from the rest of the control loop) so that the motion of the motor with the synthetic (simulated) viscous inertial damper closely approximates that of the motor and load with the physical inertial damper attached. This simulated damper gives the same improvements in gain and phase margins of the system as would the physical inertial damper but without the size and cost disadvantages.

Of course, nothing is quite free. The stepwise output of a rotary encoder and the time lag involved in processing reduce some of the margins and require a bit more complexity, but in many cases, the approximation is very good and the improvements are substantial.

In the previous article, we showed that a 100:1 inertial mismatch resulted in significant peaking at resonance (motor inertia Jm= 1e-5 kg-m^2, Ks= 100 Nm/radian ≥ Ls=1/K = 10-2 radian/Nm, load inertia $J1$= 1e-3 kg-m^2):

The damper inertia was selected as three times the motor inertia Jd=3e-5 kg-m^2, and the damping constant of the viscous oil was adjusted in the simulation to give a nice overall damping with Bd = 20N/Rad/sec.

The resulting system of the motor and load and damper improved the phase margin just above the resonance from -90 (for the velocity, and -180 for position) by about 120 degrees! It also reduced the peaking from 64 dB to 30.4 dB (gain of 1631 to a gain of 33.2) at resonance. We still have a phase margin of 40 degrees at 1000Hz, so the bandwidth of the system can be significantly improved.

In the system modeling, we take the voltage (motor speed) of the motor with 100 times the load inertia, and we divide it by the impedance of the (motor + inertia) plus the damper. The current (torque) transmitted by the viscous coupling in the damper is the same in this topology (which is an electrical model, as inertia is always modeled as a capacitor connected to a ground node) as in the parallel version with the current (torque source). This model is not physically realizable in the mechanical design but gives an easy method to calculate the torque absorbed by the mechanical damper attached to the motor. Note: The voltage source labeled as Vm in the series circuit model is the Thevenin equivalent voltage representing the speed of the motor/shaft/load without the damper present.

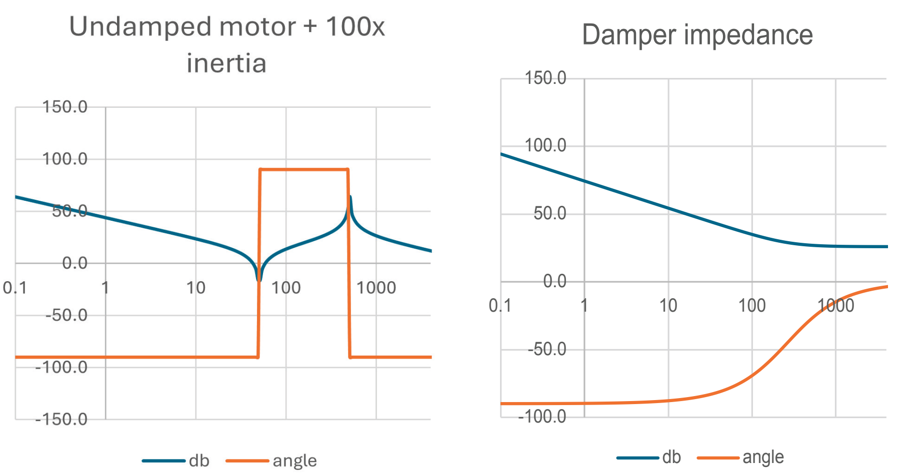


Figure 2—Impedance (velocity response to applied torque) of motor, shaft torsional spring, and 100x load inertia (left). Impedance of viscous inertial damper (right).

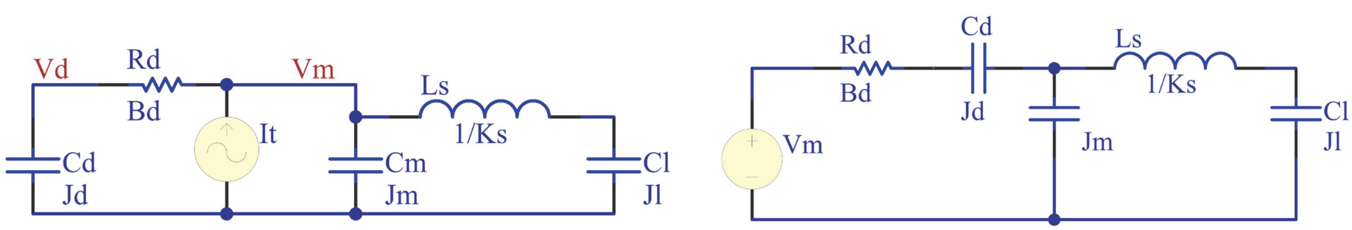


Figure 3—Electrical model of damper attached to motor inertia with motor shaft torsion spring and load inertia (left). Electrical model with Thevenin equivalent damper portion to show how spread sheet calculations were derived (right).

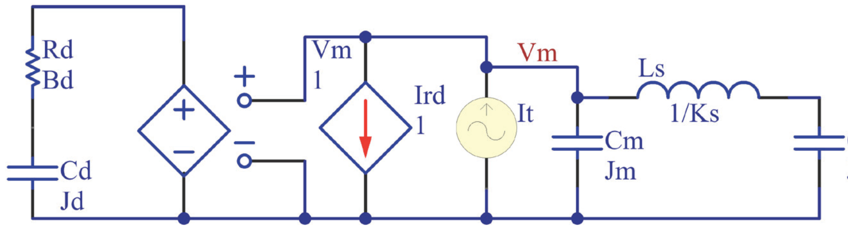


Figure 4—Electrical model with model of damper interconnected to the model of the motor, shaft and damper to implement a synthetic inertial damper. Calculated damper torque based on measured motor speed is injected into motor drive torque to produce same transfer function as an attached physical damper.

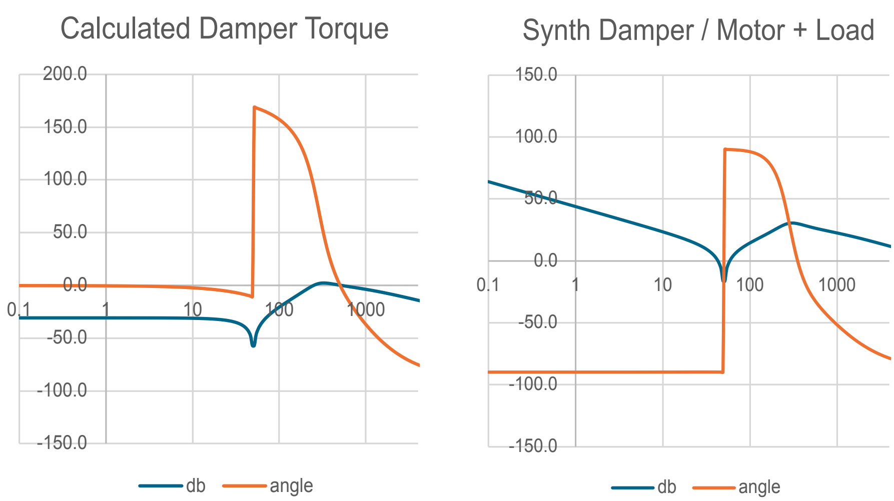


Figure 5—Synthetic inertial damper torque injection calculated from motor speed for motor plus shaft plus load inertia (left). Total system speed response to commanded torque including synthetic inertial damper torque injection (right). The system response to the synthetic inertial damper is shown equivalent to the response with a physical inertial damper.

The model works as the voltage source is modeled as a zero impedance, so the damper is essentially connected across Jm. Although this model is not realizable in a mechanical system, it does allow us to easily calculate the torque that a physical damper would incur if attached. While this model makes it easy to calculate the damper torque, it may be harder to picture how it works.

Another way to think about this model is to have the motor velocity (modeled as a voltage) drive the damper (Rd, Cd) with the same velocity (voltage) as the motor model. The torque (modeled as current) drawn by the damper is measured through Rd. This torque is subtracted from the original control torque It. The resultant net torque to the motor/shaft/load is identical to the torque that reaches the motor/shaft/load when a physical damper is in the system. Not surprisingly, the resulting system then produces the same response using the synthetic damper as it did with the physical damper!

In an actual system, this damping torque calculation would be done by measuring the actual motor position as the input, estimating the velocity, and calculating the equivalent synthetic inertial damper torque term. This damper torque term is then subtracted from the commanded torque (after some scaling for torque units used and for motor torque constant) to result in a very helpful improvement in the system dynamics. The actual system has some additional filtering terms to reduce the effects of encoder resolution with its stepwise output. These calculations are done in real time with minimized delay and are performed in the time domain, so they are a little more complicated than the simple impedance calculations in the spreadsheet, but they produce a very similar response.

Why Adding a Damper Improves Performance

In a basic system, the system gains, typically a position gain, velocity estimate (or simply derivative) gain, and integral gain, are limited by the gain/phase present in the physical system. A large inertia mismatch causes significant resonance peaking in the system. The velocity response of the system has a lag of 90 degrees as it is the integral of the acceleration. The position response of the system lags the velocity by 90 degrees as it is the integral of the velocity. This means if the motor and control system response were perfect, the position feedback would still be 180 degrees out of phase, meaning position-only gain would always be ready to oscillate (given enough gain to overcome friction). We add the velocity feedback to improve the phase margin of the system by anticipating when we will need to apply the brakes to avoid (or minimize) overshot. At just above the resonant frequency (about 510Hz for this model) of the motor/shaft/load without damper, we have a significant rise in gain to 64 db or about 1631, with a -90-degree phase, which means the gain must be set quite low to avoid having the system oscillate. With the damper added, however, the peak gain near the mechanical resonance has been reduced to 30 db (or about 33), while the phase at this maximum gain near resonance is 14 degrees positive rather than -90 for the motor velocity response. The system with a damper does not drop to a -45-degree angle until almost 754 Hz, a substantial improvement. Note that the non-damper system would need to have the gain significantly reduced (with the resulting bandwidth reduced) to avoid oscillating at the resonant frequency. The damper (either physical or synthetic) allows a much higher gain which also extends the bandwidth to allow for faster responses and much tighter control of a high inertia system. The phase boost also significantly helps even nominally low inertia systems and makes the tuning of the system much easier, often allowing the same tuning constants for an open shaft to five times motor inertia or larger with little change in the resulting motion when the load is varied.

Let's look at a couple of other example systems, starting with a geared or chain-fed system with backlash. These systems change their transfer function as the system is moving. That is, the load is only reflected to the motor when the teeth of the gear (or sprocket and chain) are in contact. When the motion reverses, the driving gear (sprocket) can rapidly accelerate while the teeth are disengaged (i.e., the load is decoupled). The teeth then slap, and according to the materials used, may significantly rebound. If the gain is high enough, or the load is positioned such that little torque is needed to keep it in position, a limit cycle oscillation may continue with the teeth bouncing off the adjacent teeth in both the clockwise and counterclockwise directions. This oscillation can quickly damage the gears while making much-undesired noise! When a physical damper is present, the inertia of the damper slows the acceleration of the decoupled motor. Upon contact between the teeth of the drive and load gears, the damper inertia continues at a higher rate than the motor for a short period, causing the teeth to not bounce off, or to have significantly less bounce, which allows the system to settle in without the limit-cycle oscillation. The damper effect allows the system gain to be significantly improved for better performance. The synthetic damper performs very similarly but without the added size and cost of a physical damper.

In some animatronic applications, for example, synthetic inertia can be made significantly larger than the physical motor inertia to help smooth out the motions in mechanisms having some degree of backlash. The damper then acts as a flywheel but with viscous damping. The flywheel action eliminates most of the high-speed vibrations which would otherwise make the motions look artificial.

In pumping applications, such as those involving a syringe-type pump, stiction may be a very significant problem. Stiction describes the fact that static friction is normally higher than dynamic friction, sometimes by a considerable degree. Stiction effects in a pump are noticed when the motion slows to a point where the seal on the moving piston begins to form mechanical bonds to the cylinder walls. This higher static friction coefficient may completely stop the piston until the control system can build up enough force to overcome the higher static friction coefficient, and then the piston lurches forward due to the lower dynamic friction coefficient. The resulting fluid flow is anything but smooth. The corrective action for this is to have sufficient gain and bandwidth in the system to rapidly adjust the forces so that the cylinder is not allowed to slow down. Rather, it can maintain the desired motion even in the presence of rapid variation in the frictional forces. Looking again at the physical damper, one might imagine a very stiff coupling grease and a large inertia that acts as a flywheel to prevent friction from stalling the motion. But this is only part of the solution, as the improved phase margin of the system with a (synthetic or physical) damper allows the gain to be significantly increased, allowing for both wider bandwidth and more powerful control system response to the friction variations, resulting in very smooth liquid dispensing even in the presence of stiction.

Note that the margins described here are just the plant torque to velocity forward transfer function wth a (synthetic) damper. Additonal installments will cover additional closed loop control techniques which are not available in a PID system which significantly benefit the performance of the improved control system.

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