DAMPING OF VIBRATION IN BELT-DRIVEN MOTION SYSTEMS USING A LAYER OF LOW-DENSITY FOAM

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ABSTRACT

We study the dynamics of servomechanisms in which power is transmitted from the motor to payload using a flat steel belt. The bandwidth of control in such systems is usually limited by a resonance in which the payload and motor oscillate out of phase and the belt undergoes longitudinal strains. In this paper, we conduct experiments on a linear positioning system and show that significant damping of the drive resonance can be attained by attaching a layer of low-density, low-wave-speed foam to the belt.

1. INTRODUCTION

Flat steel belt drives are attractive for high-speed, high-acceleration, precision positioning systems because they can incorporate a drive reduction with low inertia and very smooth power transmission (e.g., Anon[1]). But because of belt "creep" or "microslip," (e.g., Johnson[2]) it is almost always necessary to employ a feedback sensor located on the "driven" or "output" component of the system. This makes the system susceptible to instabilities at high feedback gains. In particular, the longitudinal (axial) compliance of the belt gives rise to a resonance in which the driving and driven components of the system oscillate with different phases (e.g., Abrate[3]). This resonance imposes severe constraints on the closed-loop bandwidth. In this paper, we show that by coupling a lossy, low-wave-speed foam (Varanasi and Nayfeh[4,5]) to the belt, significant damping can be introduced into the axial and transverse modes, thereby substantially improving the closed-loop performance of the system.

2. EXPERIMENTS

The belt drive shown in Figure 1 moves a carriage weighing approximately 22.1 kg through a travel of 1270 mm. The carriage is guided along a linear rail via New Way porous graphite air bearings.[6] The drive

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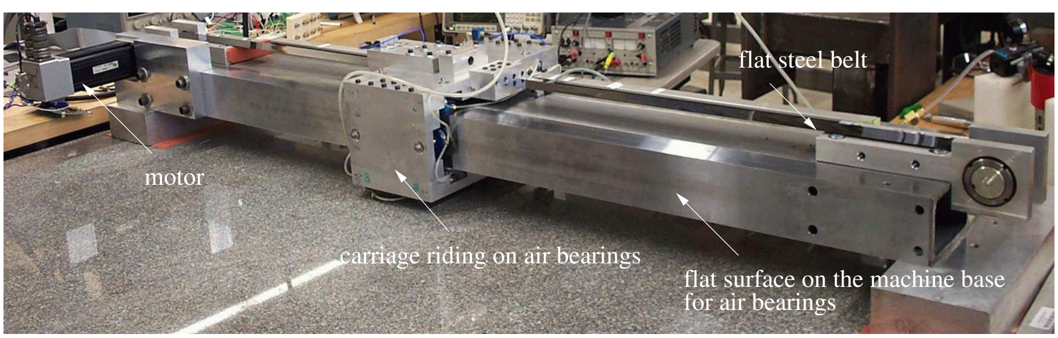

Figure 1. Photograph of the belt drive showing the machine base, carriage, motor, belt, and pulleys

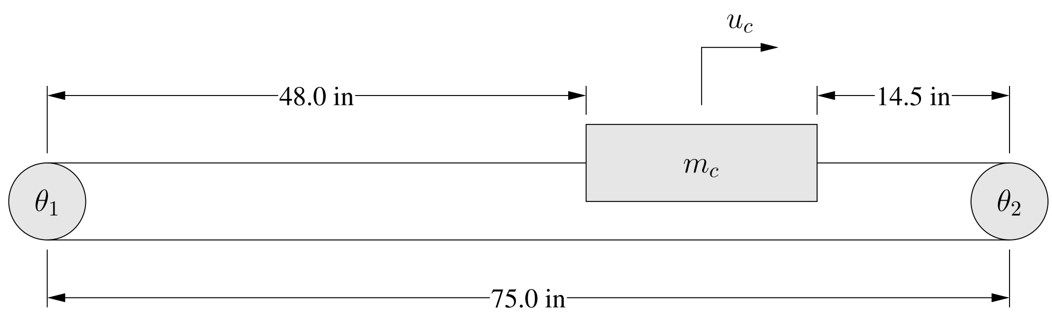

Figure 2. Schematic of the belt drive shown in the configuration used in the measurements

Table 1. Important parameters of the belt drive shown in Figure 1

belt thickness	0.10 mm
belt width	50.8 mm
mass of carriage	22.1 kg
inertia of motor	13.9×10^{-5} kgm^2
inertia of pulley	1.2×10^{-4} kgm^2
radius r of the pulleys	28.5 mm
torsional stiffness of coupling	6800 N·m

includes a spring-steel belt of thickness 0.10 mm and width 50.8 mm, the ends of which are clamped to the carriage. A brushless servomotor (Aerotech BM500[7]) is attached to the shaft of the drive pulley by means of a bellows-type coupling (R+W BK2[8]). The drive and driven pulleys are mounted onto the machine base via angular contact bearings. A Heidenhain LS403 linear encoder measures the position of the carriage and a rotary encoder built in to the motor measures its rotation. The important details of the stage are given in Table 1 and Figure 2.

2.1. Transfer Functions

We detail sine-sweep experiments for the measurement of the transfer functions from the motor torque to motor and carriage positions. In all of these experiments the carriage is located near the end of its travel at a distance of 1220 mm from the axis of the drive pulley. The transfer functions from the motor torque

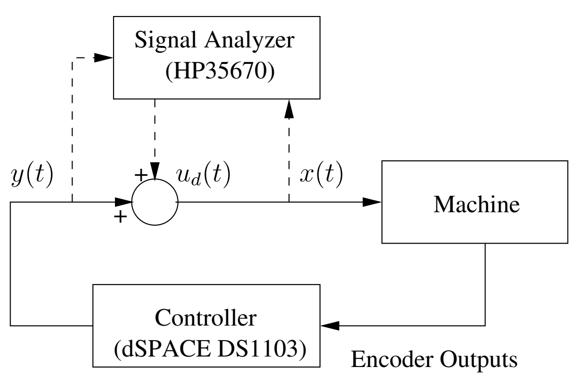

Figure 3. Schematic of sine-sweep experiments. The signals $x(t)$, $y(t)$, and $u_d(t)$ are the power amplifier input, DAC output, and swept sine disturbance, respectively. The required transfer function is $Y(s)/X(s)$.

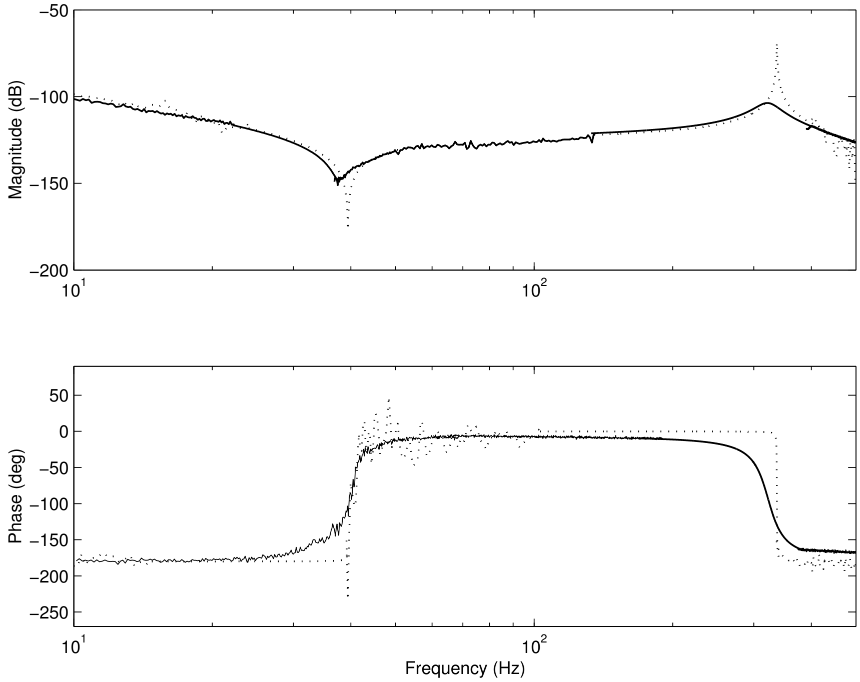

Figure 4. Measured frequency response from motor torque to motor position for the belt drive of Figure 1: without foam (dotted) and with foam (solid)

to motor and carriage positions are measured by exciting the motor with a sinusoidal torque and measuring the response from the motor encoder and linear encoder, respectively. A schematic of the set-up is shown in Figure 3. The stage is set up under closed-loop control using a PC-based DSP board (dSPACE DS1103[9]), and a disturbance signal is input to the system at a summing junction formed at the input terminal of the power amplifier of the motor. An HP35670A signal analyzer is used to generate a sinusoidal disturbance, measure the input and output signals, and determine their relative magnitude and phase.

In Figure 4, we plot the frequency response from motor torque to motor encoder with and without a foam layer attached to the belt. In the absence of the foam layer, the resonant mode arising from the compliance of the belt combined with the inertias of the carriage, pulleys, and the motor exhibits little damping (damping ratio $\zeta = 0.001$). Likewise, the complex zeros arising from the compliance of the belt and the inertia of the carriage are also lightly damped. But when a layer of 12.7 mm EAR C3201 foam is attached to the belt as shown in Figure 5, we observe significant damping in the resonant and zero modes. The damping ratio of the resonant mode is 6.0%.

Next, in Figure 6, we plot the frequency response from motor torque to linear encoder with and without foam layers. As in the previous measurement, we observe that addition of the foam layer results in significant damping ($\zeta = 6.0\%$) in the drive resonance. The second visible mode (which corresponds to a yaw mode of the carriage) is also well damped.

3. MODEL

High damping ratios result from dynamic interaction of the foam and steel belt. The foam can be viewed as a distributed (and very lossy) dynamic vibration absorber, and strong interactions occur at frequencies at or above those where standing waves can be set up in the foam. Varanasi and Nayfeh showed that relatively high damping of flexural vibration can be obtained by coupling a vibrating structure to a low-density, low-wave-speed medium, such as a foam.[5] They found that the foam used in these experiments can be modeled accurately as a lossy and isotropic continuum in which waves of dillitation and shear can propagate.

Figure 5. Photograph of the belt drive with 12.7 mm C3201 EAR[10] foam attached to the belt

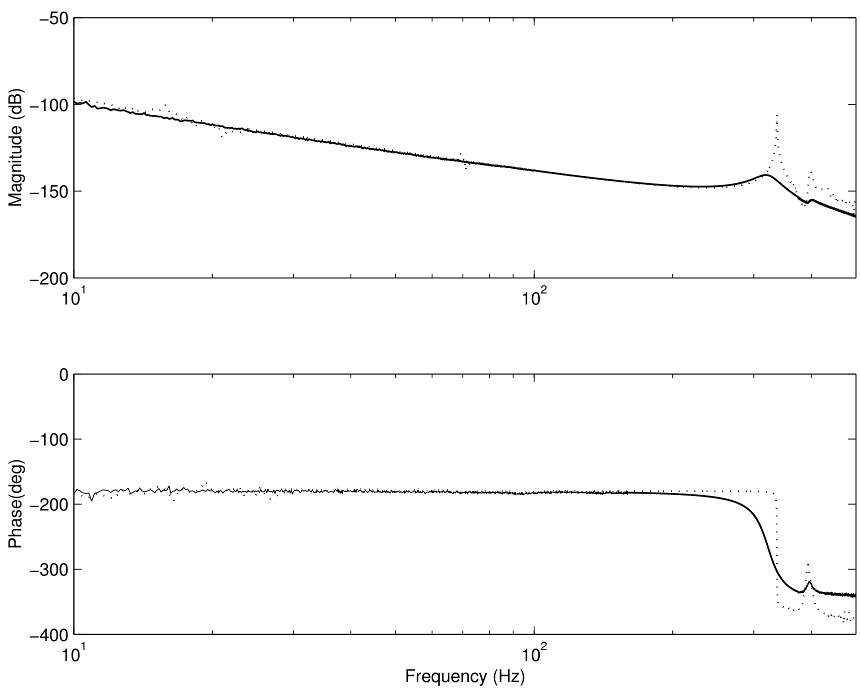

Figure 6. Measured frequency response from motor torque to carriage position for the belt drive of Figure 1: without foam (dotted), and with foam (solid)

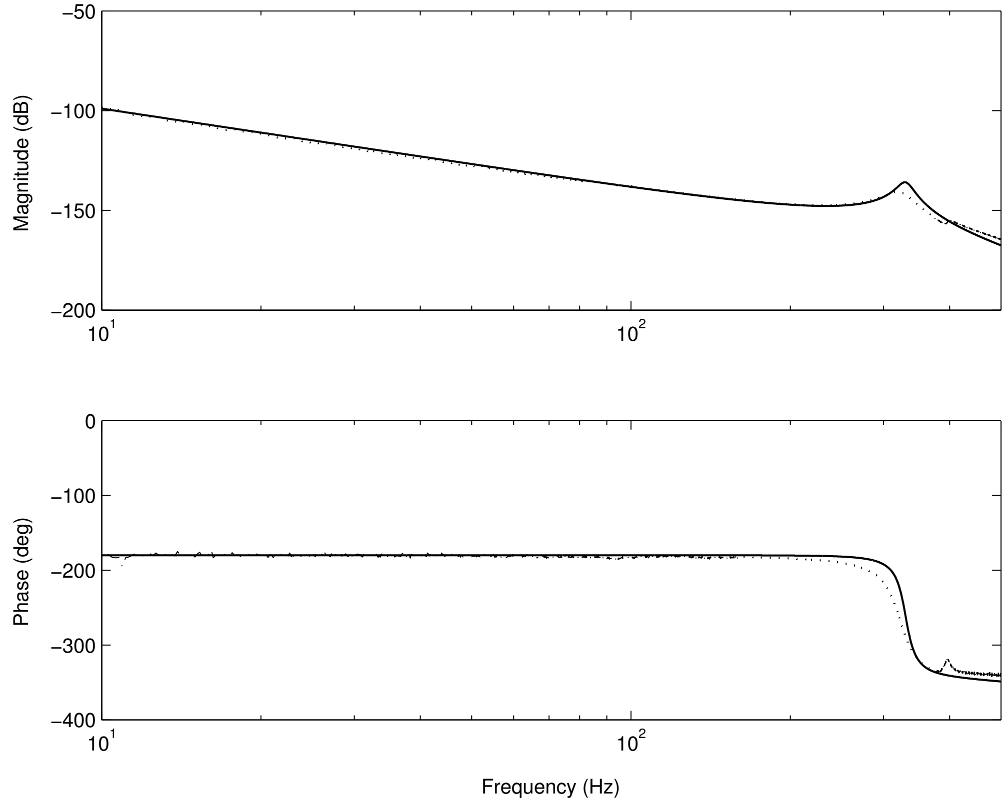

Figure 7. Comparison of the measured and predicted frequency responses from motor torque to carriage position with foam: measured (dotted) and predicted (solid)

Because the frequency of the drive resonance is much lower than that of longitudinal wave propagation in the belt, we can approximate the longitudinal strain in the belt as uniform, solve for compatible wave propagation in the foam, and thence obtain the shear stress exerted by the foam on the belt. Combining the results with the axial dynamics of the stage, we obtain the frequency response between the motor torque and motor and carriage positions. The predicted and measured transfer functions for the drive with foam are compared in Figure 7. Details of the analysis are given by Varanasi and Nayfeh.[11]

4. CONCLUSION

Attachment of a low-density, low-wave-speed foam to a steel belt is a low-cost method for attainment of predictable and relatively high damping in belt-driven positioning systems. Such foams can accommodate very large strains and are very compliant. A thin steel belt coupled to even a thick layer of foam can therefore be wrapped around a small pulley. The results presented here focus on damping of longitudinal vibration of the positioning system. Transverse vibration of the belt can also cause significant errors and even instability in belt-driven systems; this vibration is also well damped by the introduction of the foam layer, as has been studied in detail by Varanasi and Nayfeh[5] for beam bending.

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