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THE MEASUREMENT OF MESO-SCALE TORSION SPRINGS

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INTRODUCTION

The characterization of meso-scale torsion springs is an important activity for applications relying on the miniaturization of mechanical mechanisms. Designers continue to push the design limits for springs, and are faced with significant uncertainty in the quantification of the static, dynamic and fatigue behavior of small torsion springs. While static stiffness is the primary parameter specified in designs; providing in-situ measurements of dynamic stiffness, fatigue life and operational torque loads would give designers information necessary for further optimization and improved reliability. Because commercial systems do not exist that meet the requirements for range, accuracy or dynamic response, this paper will describe continued work to develop a measurement system for the characterization of meso-scale torsion springs.

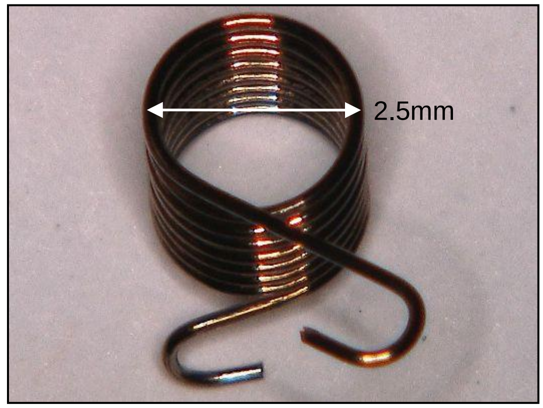

FIGURE 1. A representative meso-scale torsion spring. The wire diameter is 0.175mm, the spring length is 3.8mm, the coil diameter is 2.5mm, and the wire material is Elgiloy.

MESO-SCALE TORSION SPRINGS

Figure 1 shows an example of a spring targeted by the new measurement system. They are produced through traditional coil winding processes and typically consist of drawn Elgiloy, Hastelloy, Inconel or stainless steel wire. Coil diameters and lengths are typically 4mm or less with wire diameters spanning from 50 to 500µm. Mounting methods vary with the design application, so end geometries may have curved or straight tangs with radii of curvature and bend angles in any orientation. Operational torque loads are typically 0.5 to 2.0mN·m, with initial preloads in the range of 0.5 to 1.5mN·m. Normal mechanism motions rely on 20 to 30° of spring rotation beyond their initial preloads at cycle rates up to 100Hz.

MEASUREMENT SYSTEM REQUIREMENTS

While systems for measuring torsion springs exist[1], no commercial equipment meets the requirements for meso-scale torsion springs. Fundamental shortcomings of commercial systems are their inability to measure dynamic spring motions or to quantify torque below 10mN·m. Additional problems with measurement accuracy, with traceability and with system reliability have been observed with customized equipment delivered to SNL. The presented work details a system developed to address each of these weaknesses. The prototype system has been designed to measure in-situ torque ranging from 0.1 to 10mN·m, at frequencies up to 100Hz and with dynamic rotations of ±15°. Such requirements enable the measurement of both static and dynamic spring rates. As a result, the effects of resonance, cyclic loading, fatigue, end effecter geometry and even defects on operational performance should be quantifiable.

SYSTEM DEVELOPMENT

The prototype measurement system, Figure 2, was designed to be simple, and yet to provide the functionality necessary to meet the design requirements[2]. Recent developments have focused on system improvements thru the elimination of noise sources in the torque sensor signal, the addition of a preload flexure to the voice coil stage and the implementation of closed loop control of the voice coil actuator.

<u>Torque Sensor</u>

Torsion loads on springs under test are measured using a Kistler 9329A torque cell and

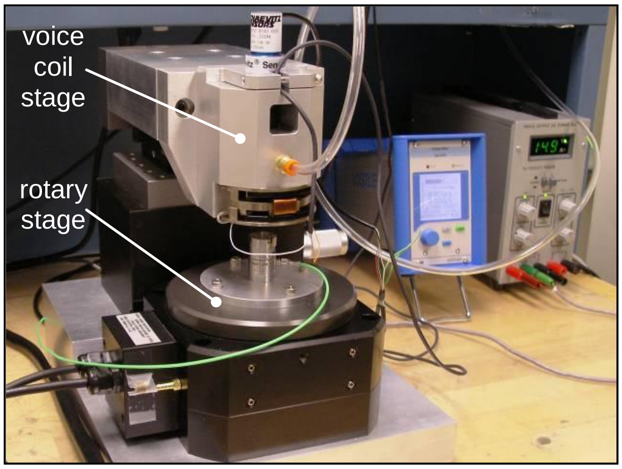

FIGURE 2. The prototype system for the measurement of meso-scale torsion springs.

5015A charge amplifier. Although the 9329A provides a measurement range up to ±1N·m, its lowest range setting (±10mN·m) is required to measure meso-scale torsion springs with adequate resolution. Cable management was found crucial in this operating regime. Without strain relief, the sensor cable stressed the torque sensor during rotary stage motion and create signal errors of 1mN·m or greater. Constraining the sensor cable to the table of the rotary stage eliminated stresses on the sensor and reduced the noise threshold during rotation to within the 0.03mN·m product specification. Initial work also revealed the ability of a magnet to introduce false torque signals, prompting concerns regarding potential errors from operation of the voice coil. Operational tests demonstrated, however, that no measureable error signal was introduced by the voice coil although an offset due to the stator magnets is nulled during spring testing.

Preload Flexure Mechanism

A significant shortcoming in the original system was inadequate dynamic response from the voice coil actuator[3]. Therefore, a flexure mechanism, Figure 3, was designed to introduce stiffness to the voice coil stage. The design combines two identical planar flexures which are flipped 180° about their central axis and pinned together using light press fit dowels. The resulting symmetry ensures a constant spring rate in either direction of rotation of the voice coil. Dowel locations are offset 20° from the mounting holes and provide symmetrical preloads in the direction of coiling to ensure that both flexures experience only coiling loads for any rotation of the voice coil stator within its

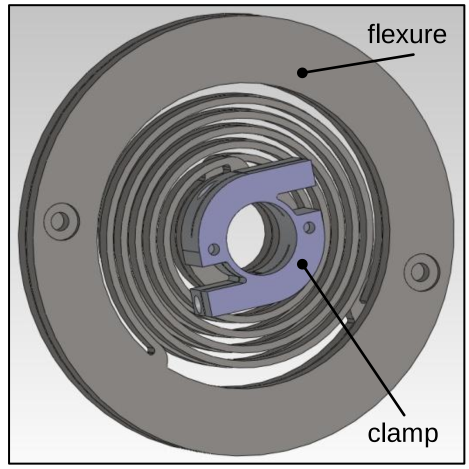

FIGURE 3. Preload flexure assembly.

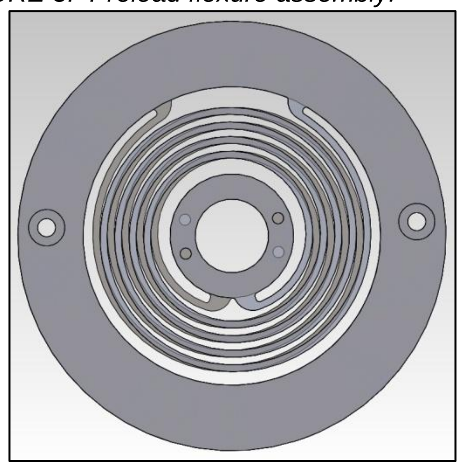

FIGURE 4. Flexure stack showing 180° flipped orientations and 20° offset dowel locations.

±15° range of motion. The assembly fits between the stage base and voice coil rotor, and utilizes a clamp for attachment to the rotating shaft. The flexures are made from 304 stainless to facilitate fabrication via wire-EDM machining.

The flexures were designed to generate an open loop system resonance of 10Hz. While well below the original specification of 100Hz, 10Hz represents the maximum that can be achieved for a ±15° range of motion with the torque capabilities of the BEI-Kimco RA29-11-002A rotary voice coil. Based on a 1.4×10^{-5} kgm^2 mass moment of inertia for the voice coil's moving mass, the subsequent torsional stiffness, K_t, for a single flexure was 0.028N·m/rad. The thickness of the flexure, b, was arbitrarily chosen to be 2mm, while a maximum allowable stress, σ_{max}, of 162MPa was selected to ensure a fatigue life greater than 10^8 cycles. Based on closed form design equations for a spiral spring[4], the corresponding strut width, w, was calculated to be 0.86mm from

$$w = \sqrt{\frac{6 \cdot K_t \cdot \theta_{max}}{b \cdot \sigma_{max}}}$$

where θ_{max} is the maximum angle of rotation in radians. The 356mm length of the spring section, l, was obtained from the relationship

$$l = \frac{\pi \cdot E \cdot b \cdot w^3}{6 \cdot K_t}$$

where E is the material elastic modulus. A start diameter, D_i, of 12.7mm was chosen to accommodate the diameter of the voice coil shaft. The subsequent 24.16mm finish diameter, D_o, was determined by

$$D_o = \frac{2 \cdot l}{\pi \left(\frac{\sqrt{D_i^2 + 1.27 \cdot l \cdot w} - D_i}{2 \cdot w} - \frac{\theta}{2 \cdot \pi} \right)} - D_i$$

Finally, the 4.96 number of turns, n, was found geometrically based on the length, initial diameter and final diameter.

Once the flexure design was developed analytically, FEA analyses were performed for validation and further investigation. A torsional stiffness of 0.033N·m/rad was predicted from the FEA for a single flexure and was within 16% of the closed-form solution. A maximum stress of 172MPa for a flexure rotation of 35° was then calculated by FEA and was within 6% of the original design target. Finally, the first resonance of the flexure was estimated via FEA to be 160Hz which was deemed satisfactory for system operation.

Closed Loop Control

A second area of effort for improving the dynamic response of the voice coil stage was to implement closed loop control of the voice coil actuator. Figure 5 shows the transfer function obtained without the preload flexure using a PID control algorithm with proportional, derivative and integral gains of 0.07, 1.0 and $1\text{x}10^{-6}$. Control was implemented using a 2kHz update rate, and achieved a -3dB crossover at 35Hz with a damping ratio of 0.75. 1.33° of overshoot and a 40msec settling time were observed for a step input, Figure 6; while a 0.35° amplitude error and a 52.3° phase error were observed for a 30Hz, ±15° sine wave input, Figure 7. A

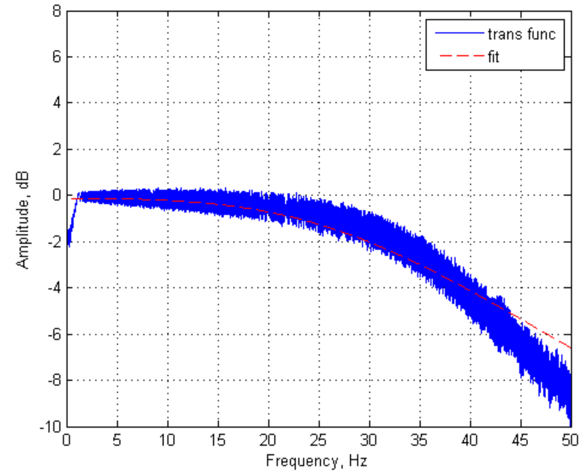

FIGURE 5. Voice coil actuator PID closed loop transfer function.

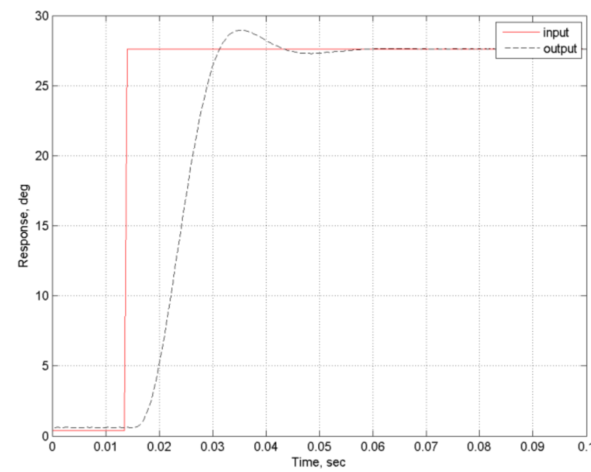

FIGURE 6. Voice coil actuator response to a 27°step input.

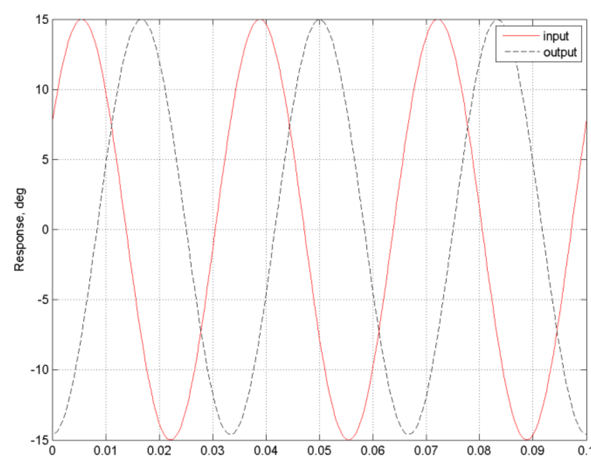

FIGURE 7. Voice coil actuator response to a 30Hz, ±15° sine wave input.

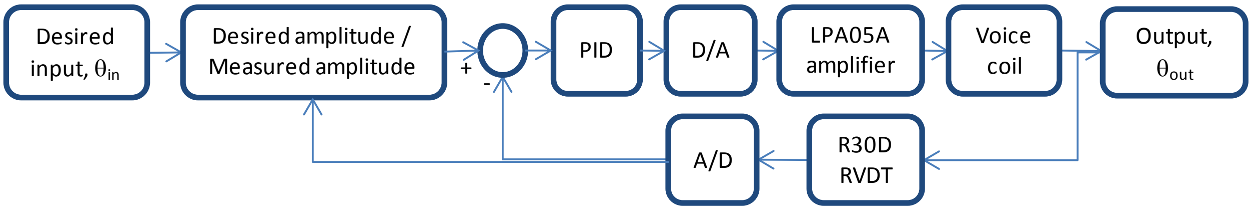

FIGURE 8. PID control diagram with amplitude control.

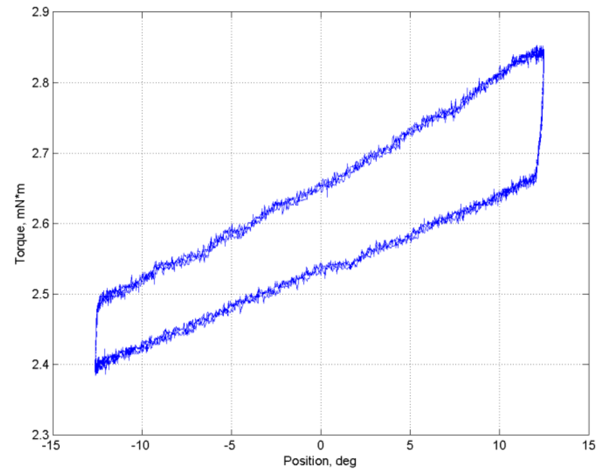

FIGURE 9. Initial torsion spring test data.

modified PID control scheme, Figure 8, was utilized to reduce sine wave errors whereby the control signal amplitude was modified based on the desired sine wave response. As a result, sine wave response up to 30Hz consisted predominantly of phase errors with amplitude errors remaining below 0.4°.

SPRING TESTING

Initial spring tests have been performed as Figure 9 shows spring data using a 82.5° initial offset superimposed with a 1Hz, ±12.5° amplitude sine wave. Torque values are within the tolerance range of the tested spring design, an early indication of system success. Initial insight into dynamic spring response has also been achieved as the internal friction between the coil windings has been observed to contribute 5-10% of the torque required to rotate the torsion springs. This internal friction is quantified by the step change in Figure 9 that occurs at the extremes of spring travel.

CONCLUSIONS AND CONTINUED WORK

The development of a prototype system for the testing of meso-scale torsion springs continues. Improvements regarding the torque sensor, a flexure mechanism and closed loop control have been implemented as initial spring data has been collected. Continued work is focused on further system improvements, understanding its performance and quantifying uncertainties in spring test measurements.

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