A Quasi-Zero Stiffness Vibrational Damper for Cryogen Free SPM

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Traditional low temperature scanning tunneling microscopes (STM) use liquid helium cryostats to cool samples to temperatures as low as 4 K. However, as the world's helium supply dwindles and prices drastically increase, such designs are made exceedingly impractical. Closed-cycle dilution refrigerators can reach comparable temperatures without a helium supply and are thus logical candidates for future-proofing low temperature STMs. However, these refrigerators operate using pumps that introduce significant, low frequency vibrational noise to the STM that cannot be damped via traditional means (Fig. 1). As STM often requires picometer-level resolution, this vibrational noise is extremely detrimental, so a novel vibrational isolation system is necessary.

We demonstrate a novel, fully passive vibration damping system designed to isolate a scanning tunneling microscope (STM) from the low-frequency noise introduced by a cryogen-free refrigerator pump.

Why Quasi-Zero Stiffness?

> Closed-cycle, dilution refrigerator pumps introduce **low frequency vibrational noise** (resonance at 1.3 Hz) directly to the STM.

> Attenuating such frequencies with traditional spring dampers would require a very low resonance frequency (<1Hz). Such **springs would be extremely long and have poor thermal conductivity**.

> **A quasi-zero stiffness vibrational isolator allows for extremely low resonance frequency in a compact package.**

Working Principle

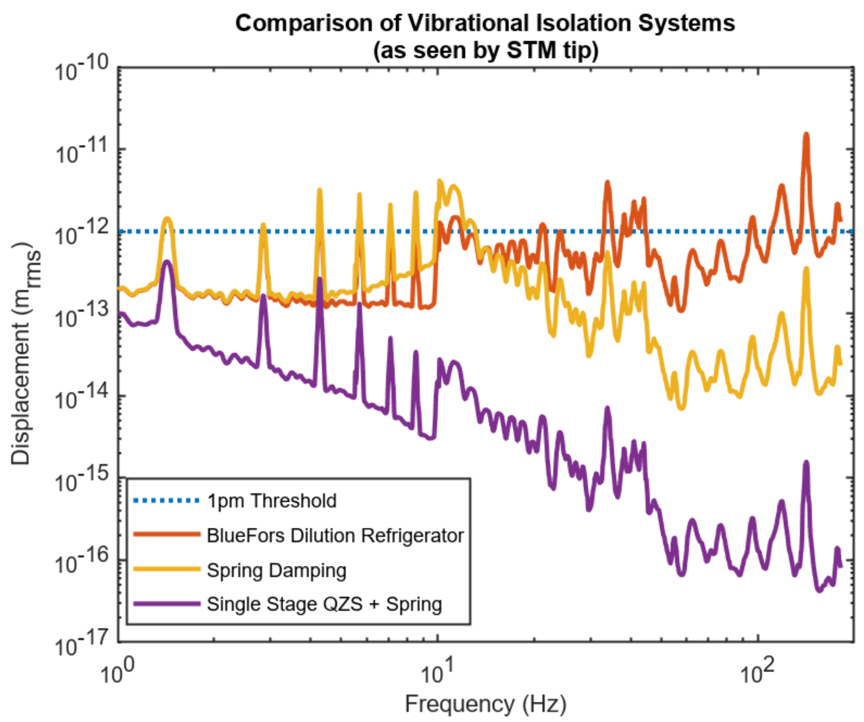

Figure 1: A comparison of the QZS vibrational isolator and the traditional spring damper applied to the refrigerator pump's vibrational spectrum. The sub-hertz resonance of the QZS is necessary in order to keep the vibrational levels of the STM tip below 1pm at low frequencies.

The lower a spring's stiffness, the lower its resonance frequency and the better it attenuates low frequencies. **A quasi-zero stiffness mechanism effectively acts as a traditional spring with an extremely low stiffness**. This is accomplished by combining positive and negative stiffness ('buckling') mechanisms (Fig. 2) to create a nonlinear force-displacement relation, with a local region of near-zero stiffness. The net effect is to shift the resonances frequency of the damper - below which vibrations are transferred - to lower frequencies Such a system requires less displacement than a linear system with the same stiffness, and is thus ideal for a compact, low frequency vibrational isolator.

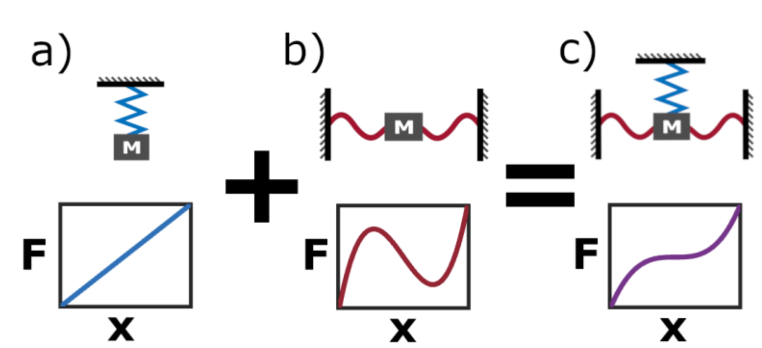

Figure 2: Positive stiffness (a) can be combined with negative stiffness (b) to create quasi-zero stiffness (c).

Design and Simulations

While traditional springs could be used for the positive stiffness element, flexures were necessary to create the negative stiffness element. To determine the behavior of these flexures, finite element analysis (FEA) using Solidworks was utilized (Fig. 3). In particular, it is necessary for the flexures to only undergo elastic deformation, as plastic deformation would introduce undesirable hysteresis and instability to the system. Force-displacement and frequency response data was also collected using FEA.

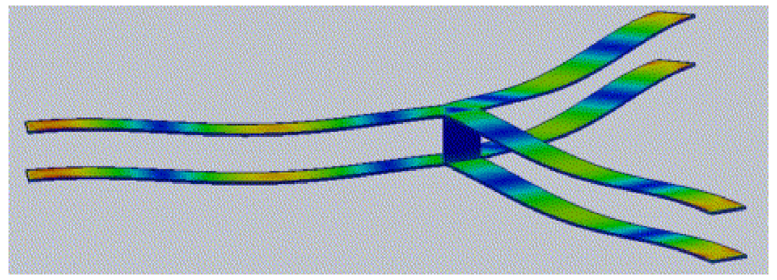

Figure 3: An example of using finite element analysis to simulate the strain on the flexures.

After finding a suitable flexure design, the complete vibrational isolator system was designed (Fig. 4).

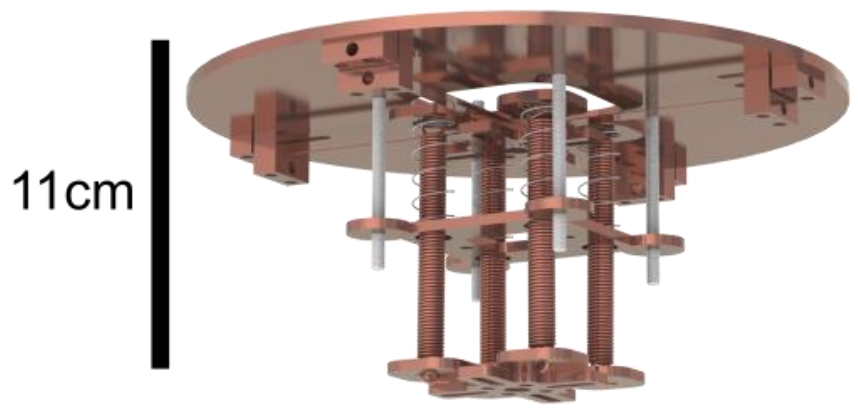

Figure 4: The current model of the system.

This design features:

- A second stage of extension springs for additional high-frequency damping.
- Beryllium copper (BeCu) construction, chosed for its high thermal conductivity and relatively constant stress/strain properties at cryogenic temperatures versus room temperatures
- Four flexures, each independently adjustable
- Adjustable vertical springs to accommodate a range of loads.
- Brackets for tunable eddy-current damping.

Procedure

The first prototype of the system was then machined out of stainless steel (which has nearly identical stress/strain properties to BeCu at room temperature). An accelerometer (A Wilcoxon 731-207) was used in conjunction with a low-noise power supply and a oscilloscope (Yokogawa DL850EV) to measure the frequency response of both long springs and the QZS system. The test mass was displaced half a centimeter, and the resulting signal was recorded and converted to the frequency domain (Fig. 5). Data was also taken with a force sensor and distance sensor to find the force vs distance relation of the QZS system.

Results

The vibrational spectrum of the QZS system shows a peak resonance at 0.68 Hz (Fig. 5), well below the 1 Hz requirement. The other resonance peaks are likely due to modes between the two stages of the system. Such peaks can likely be significantly reduced by including additional eddy-current damping on the second stage – something not included on this prototype. Traditional long, thin springs were also tested and found to have a resonance at roughly 2.24 Hz, significantly higher than the QZS system and resonance frequency of the cryopump.

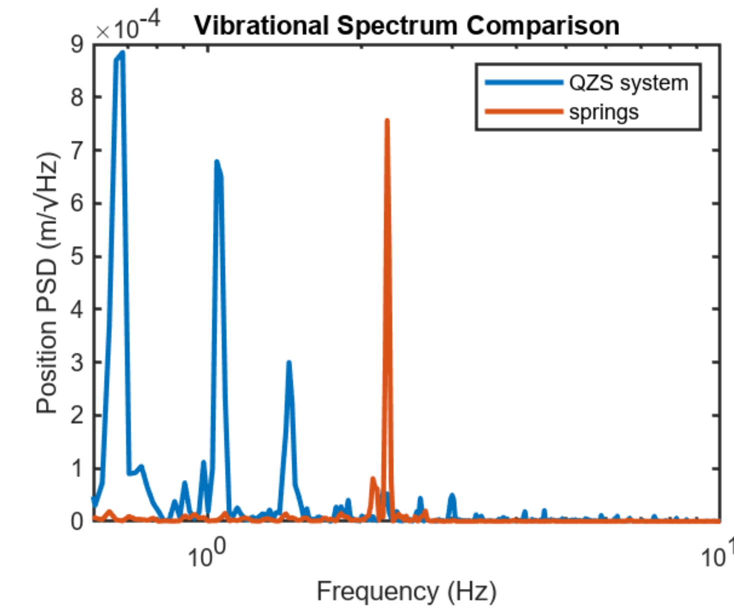

Figure 5: Resonance measurements of the two-stage QZS system (0.68 Hz) and a traditional spring system (2.24 Hz).

In order to verify that the QZS stage has a region of near-zero stiffness, force versus displacement data was taken (Fig. 6), showing a clear region of quasi-zero stiffness.

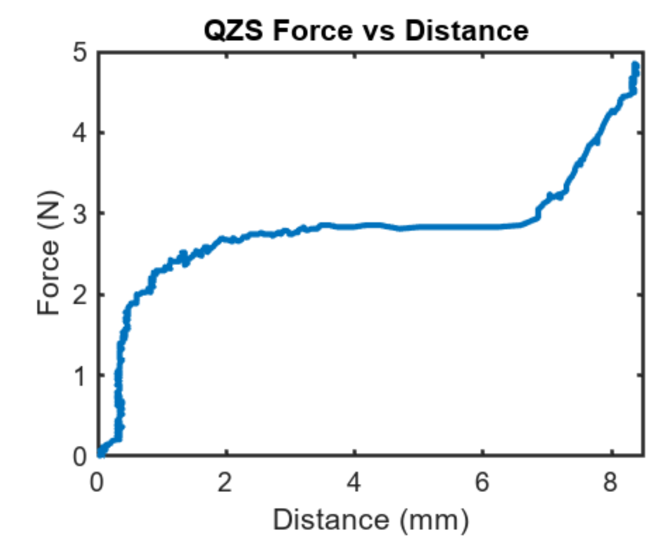

Figure 6: Force versus displacement data of the QZS stage of the vibrational isolation system.

Acknowledgements

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