Cryocooler Disturbances Reduction with Single and MultipleAxis Active/Passive Vibration Control Systems

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ABSTRACT

Cryocoolers are well known sources of harmonic disturbance forces. In this paper two miniaturized, add-on, vacuum compatible, active vibration control systems for cryocoolers are discussed. The first, called VIS6, is an active/passive isolation hexapod and has control authority in all six degrees of freedom. This capability is desirable when reduction of all cryocooler disturbance loads, including the radial loads, is required. Each of the six identical hexapod struts consists of a miniature moving coil electromagnetic proof mass actuator, custom piezoelectric wafer load cell, viscoelastic passive isolation stage, and axial end flexures. The first five disturbance tones are reduced over a bandwidth of 250 Hz using a filtered-x least mean square algorithm. Load reductions of 30-40 dB were measured both axially and radially. The second system, called VRS1, is a pure active control system designed to reduce axial expander head disturbance loads. It works on the basis of a counter-force developed from an electromagnetic proof mass actuator. Error signals are provided from a commercial accelerometer to a standalone digital signal processor, on which a filtered-x least means square control algorithm is implemented. Over the 500 Hz control bandwidth, the 11 disturbance tones were reduced on between 14 to 40 dB.

Keywords: Vibration isolation, active control, reaction mass actuators, cryocoolers, piezoelectric wafer load cell

1.0 INTRODUCTION

Infrared sensors for earth observation, missile seekers, magnetic resonance imaging devices, advanced telecommunication circuitry, etc. can all depend on cryocoolers to maintain their extremely low operating temperatures (typically 77 Kelvin or lower). While cryocoolers offer certain advantages with regard to size and volume versus passive cooling methods, they can also have certain negative side effects. The side effect of interest to this paper is the tendency of cryocoolers to generate vibration disturbances that can perturb the operation of the system being cooled, cause blurring of images, etc. These disturbances are caused by a variety of mechanisms including the movement of the pistons in the compressor, residual misbalance between the opposed compressors, motion of the regenerator in the expander, and resulting gas movement, volume changes and local expansion of expander walls.

Cryocooler disturbances are typically tonal. Normally multi-harmonic in nature, their strength typically scales with the rate of heat removal desired. The primary tone corresponds to the frequency of cryocooler operation. These issues have traditionally been addressed through system level configuration of opposed compressors and specialized drive electronics. New cryocooler technologies, such as pulse tubes and those based on turbomachinery, also show some promise of reducing undesirable vibrations.

Other approaches have also been developed, by CSA (Anderson et al, 1999) and others, that use active control to achieve extremely low vibration levels. This particular paper focuses on two recently developed systems designed as drop in solutions to reduce unwanted cryocooler disturbances. The first, VIS6, is a full six degree of freedom active isolation system. The second system, VRS1, is a more traditional active, single axis, inertial/counter-force control system.

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2. BACKGROUND: CRYOCOOLER DISTURBANCES

As stated in the previous section, cryocoolers are well known sources of disturbance vibration. In this section, representative data will be given for a Hughes 7050 one Watt split Stirling tactical cryocooler. This cryocooler is shown below in Figure 1 with the expander head/coldfinger on the left. The cryocooler was mounted to a custom six axis dynamometer, also shown in Figure 1, and run at various drive levels that covered the operating range from 20 to 100%. Representative time traces are shown in the upper right corner of Figure 1. The dominant 46 Hz mode is clearly visible, as are the effects of multiple other higher order harmonics. The results of the six load measurements were then taken and transformed into axial and radial disturbances based on the known geometry of the dynamometer. As expected, and as shown in the lower left of Figure 1, these loads increase with cooling power. Axial loads can reach 2 Newtons and radial loads approximately 1.25 N. It is these types of disturbance levels that the VIS6 and VRS1 hardware are specifically designed to reduce. Actuators capable of handling higher disturbance loads are discussed in Flint et al. (2000).

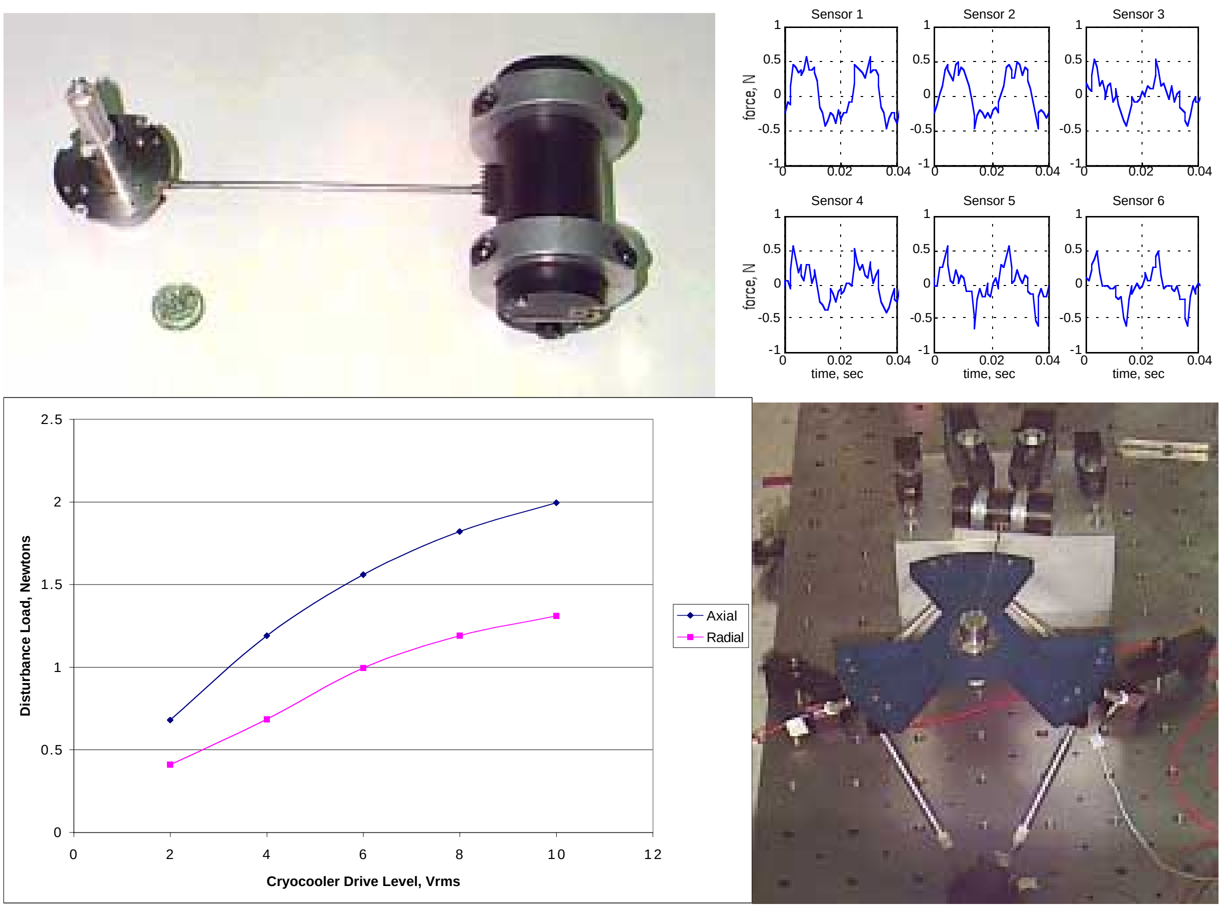


Figure 1: Clockwise from upper left are the 1 W split Stirling tactical cryocooler (with US quarter shown for scale), representative time traces of its disturbance levels for each axis of a 6 DOF dynanometer, the extracted axial and radial disturbance levels of the cryocooler versus drive level and the test set up used to characterize the cryocooler performance.

3. SIX AXIS VIBRATION ISOLATION SYSTEM (V1S6)

The Vibration Isolation System – 6 Axis (VIS6), shown in Figure 2, is designed to reduce both the axial and radial loads generated by typical cryocooler operations. This is an important improvement over purely single axis systems, since once the first few axial disturbances are reduced, the radial loads constitute a significant portion of the remaining dominant disturbances. In other cases there can also be sensors or surrounding structure that are more sensitive to radial loads than axial loads.

The developed system is arranged in a Stewart or hexapod configuration. Each of the six struts that make up the hexapod contains a miniature electromagnetic moving coil PMA, custom piezoelectric wafer load cell, and a viscoelastic based passive isolation stage. The rest of the system consists of a load cell signal conditioning unit and the primary electronic support box which provides all power conversion, signal routing, DSP board, actuator power amplification, and user interface functions. Only signals from the cryocooler drive and 18 to 36 VDC power are required. The system weighs 4.2 kg (of which approximately 70% is associated with the non-flight weight electronics, cabling and power conditioning) and requires 21.6 W to operate in standby mode. An additional 2.64 W are required when actively controlling, of which no more than 0.7 W total is estimated to be dissipated in the struts. The standby power could be further reduced through the use of customized rather than commercial grade signal processing electronics. The transducer hexapod system hardware fits within a cylinder that is 114 mm in diameter and 68 mm tall. Performance has been confirmed in detailed thermal vacuum testing. It requires minimal modification to existing cryocooler mounts, offers significant reductions in all cryocooler vibrations, and can easily be changed to accommodate the designs of cryocoolers other than the one for which it was demonstrated.

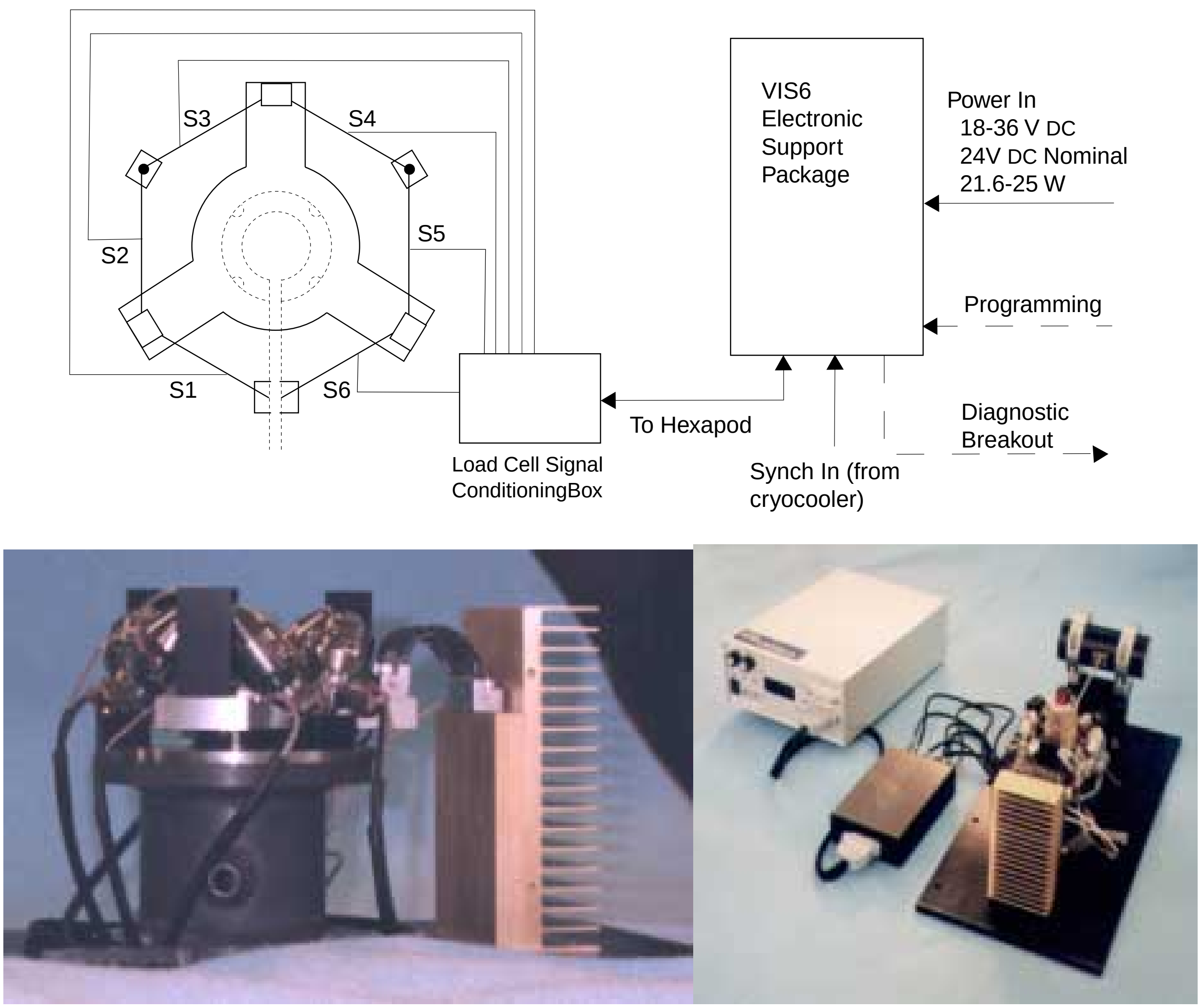


Figure 2: Schematic and pictures of the complete VIS6 system and a close up view of the hexapod.

3.1 VIS6 Strut

The core of the vibration isolation system is the repeated strut element. These miniature struts weigh 118 grams (4.6 ounces) and have a volume of 18 cc (1.1 cubic inches) in volume. The strut is dominated by an electromagnetic reaction mass actuator, which when combined with a passive viscoelastic puck isolator, load cell, and end flexures, forms the strut shown in Figure 3. Each strut is nominally 60.8 mm. long and has a maximum outer diameter of 25.4 mm. The majority of a strut's weight, 75%, comes from the actuator. Representative transfer functions from a ship set of seven such struts are also shown in Figure 3. These transfer functions correspond to the measured force generated by the strut as transmitted through the passive stage into a reference load cell. The low variability in performance among the struts eases the design of control algorithms. Further details of the subsystems that make up the active isolation strut are discussed in the following sections.

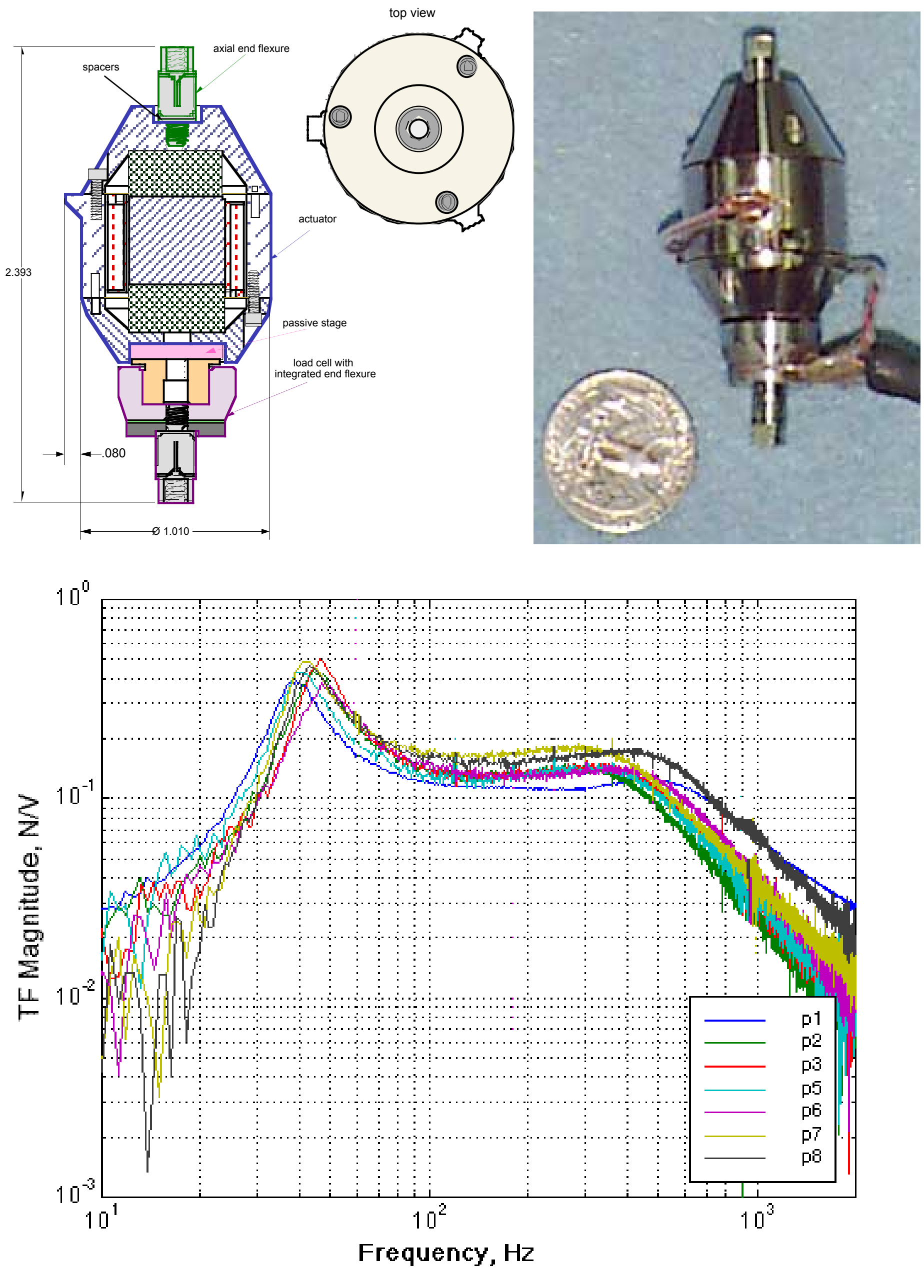
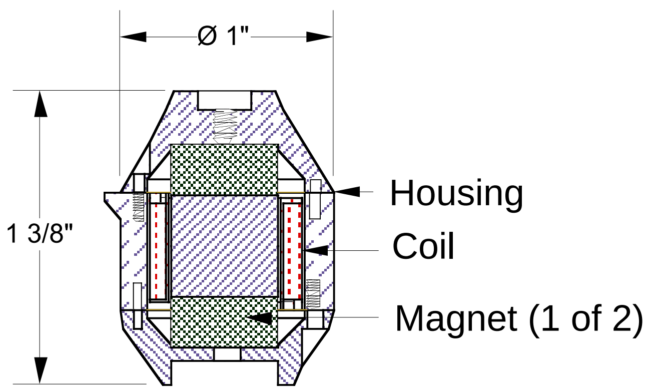


Figure 3: Schematic (dimensions in inches), picture (with US quarter shown for reference) and performance of the VIS6 strut

3.1.1 VIS6 Actuator

Due to the desire to minimize the VIS6 hexapod size a miniaturized actuator configuration based on a moving coil and fixed magnet was designed. This approach depends on the fields of opposing magnets being driven through the movable coil and returned through the actuator housing. This configuration forms a very efficient closed electromagnetic circuit will little EMI leakage. All material used was chosen to minimize outgassing.

The actuator is shown in Figure 4. Each actuator is nominally 34.9 mm long and has a maximum outer diameter of 25.4 mm. In three locations, this diameter is exceeded by 1.52 mm to accommodate optional mounting flanges. Some of the actuator length is actually extraneous, and used to form a recess for a passive isolation stage and an axial end flexure. Without these features, each actuator would be 29.7 mm long and weigh slightly less. As built, the actuator weighs approximately 88 grams (3.12 ounces) and takes up approximately 13.4 cc (0.82 cubic inches) in volume.



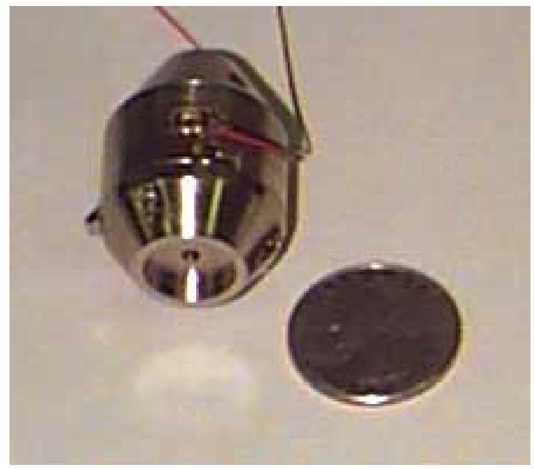


Figure 4: Schematic and picture (US quarter shown for reference) of the VIS6 actuator

In the low frequency range, the actuator output is limited by the 0.040 in. mechanical endstops required to limit the flexure stroke to the linear ranges. These limitations are most noticeable in the performance near the 40 to 50 Hz internal resonance. Above the resonance, force output is not displacement limited and 0.14 N/V can be achieved. For power limited cases (0.17 W dissipation allowable) approximately 0.5 N can be generated. If the power limitation is removed and the strut is exercised to its rated capacity, at resonance the actuator can produce 0.61 N (set by gravity induced sag limited end stop limits), and 1.0 N at 4-5 times the resonance as limited by self induced heating.

3.1.2 Passive Vibration Isolation Stage

A passive isolation stage, shown in Figure 5, was integrated into the strut in order to ensure good high frequency performance. It consists of a single 2.03 mm thick, 11.11mm diameter puck of 3M ISD 113 viscoelastic material nestled into a 2.54mm deep, 12.7 mm diameter cavity in the bottom of each VIS6 actuator. Approximately 0.79 mm is left as diametrical clearance for the viscoelastic material to expand under compressive loads. The pressure sensitive adhesive material holds itself in place through a bonding layer on either side. All loads in the strut pass through this element into the passive stage interface which serves as a mounting stud for the wafer load cell unit. The entire system is only 4 grams with the majority in the mounting stud.

The effect of the passive isolation stage can be seen in the previously shown strut only transfer functions of Figure 3. The resonance in the 40 to 50 Hz range is due the dynamics of the disturbance actuator used. Between 200 and 500 Hz the passive stage introduces an unavoidable small amplification in response levels followed by dramatic decreases in transmitted force vs. that of the hard mounted actuator for all higher frequencies. It should be noted the actual frequency at which the resonance occurs is a complex function of both the frequency and temperature dependent stiffness of the viscoelastic isolation stage and the load that it supports.

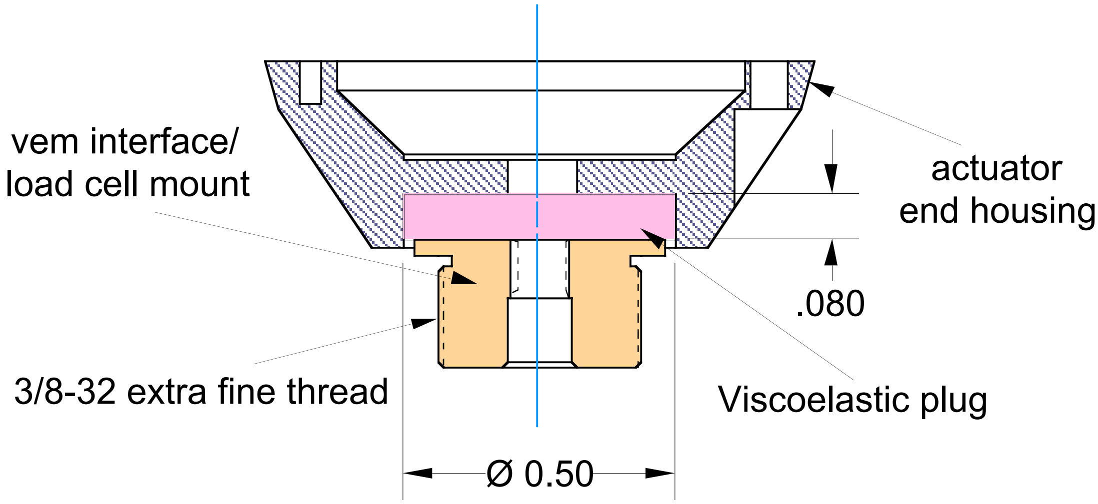


Figure 5: Schematic of the passive isolation stage (dimensions in inches)

3.1.3 Load Cell and Axial End Flexure

Due to the extreme limits placed on allowable volume for the VIS6 system, and lack of a reliable commercially available tension/compression load cell, it was necessary to develop a custom sensor. The resulting load cell is shown in Figure 6 along with details about the axial end flexures that form an integral part of the load cell pre-stress mechanism. The core of the load cell is less than 12.7 mm in diameter and 5.08 mm thick. The remaining portions of the load cell serve as interfaces to the passive isolation stage and to the axial end flexure and are slightly larger. In order to minimize total strut length, the axial end flexure was made integral to the load cell as the pre-stress mechanism.

The primary element of the load cell is a circular wafer with a through hole made from electroded piezoelectric material. When strained, the piezoelectric material develops an electrical field and correlated voltage. This voltage level corresponds directly to the level of strain. By statically pre-stressing the wafer in compression, it is possible to dynamically measure both tension and compression loads passing through the wafer. Without the static pre-stress it would only be possible to measure compression loads. The static pre-stress was realized by sandwiching the piezo wafer between the passive stage interface and a load distribution washer. These two components were then held together with the mounting threads of an axial end flexure. The sensitivity of the load cell can be adjusted by varying the applied pre-stress.

Two axial end flexures are used in each strut in order to help ensure that the majority of the loads passing through the struts are those which are axial relative to the strut. A flexure, also shown in Figure 6, weighs 1.7 grams, is 5.1 mm wide, and is 13.1 mm tall. The required off-axis flexibility was achieved with narrow slots manufactured with an EDM procedure. Finite element modeling of the component was performed in order to understand and prevent possible fatigue failure modes. The required limits were enforced with integral hard stops. The flexure can rotate 0.6° in each plane and is also very flexible in torsion.

3.2 VIS6 System Electronics

The operation of the hexapod is enabled through the use of a user friendly laboratory/industrial grade electronic support package (ESP) and a separate load cell signal conditioning system, both shown in Figure 7. The ESP contains all required power converters, digital signal processor, I/O support, actuator power amplifiers, power and signal routing, and user interface support. The load cell signal conditioning box acts as a conduit to the hexapod and provided local signal conditioning close to the load cells to limit the generation of undesirable line noise.

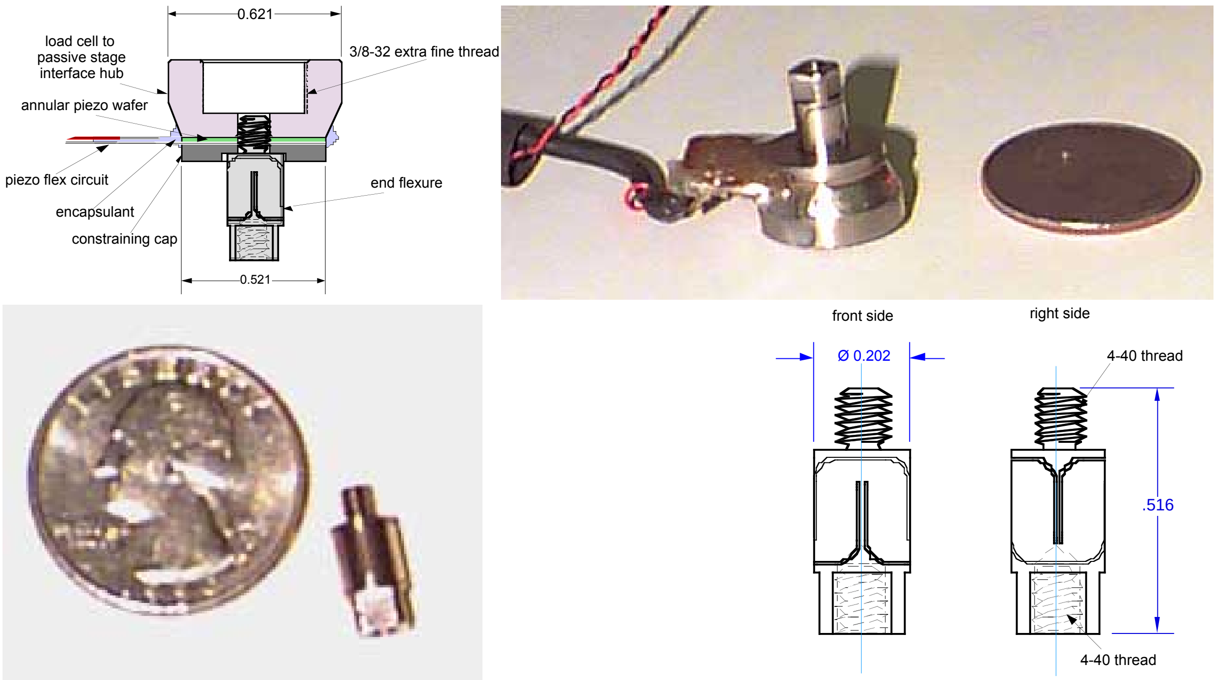


Figure 6: Schematic and picture of piezoelectric wafer load cell and axial end flexure

The foundation of the VIS6 ESP is a single board computer, the core of which is a Texas Instruments TMS320C6201 fixed point digital signal processor (DSP) clocked at 160 MHz providing 32 bits of digital I/O. Modular sites allow the integration of two analog I/O cards, each providing 4 channels of 16 bit D/A and A/D conversion. Six channels of A/D and D/A and one digital I/O channel were used in the core of the control algorithm. Additionally, two channels of D/A and seven digital I/O channels were used to support the user interface. Software embedding is done via either RS232 serial, or in line memory scan and programming (JTAG) for up to 128 K of ROM.

Figure 7: VIS6 electronic support package and load cell signal conditioning support box shown to the same relative scale (US quarter used for reference)

The VIS6 electronic support package is housed in an aluminum enclosure that is 8.37 inches in width, 3.47 inches in height, and 12 inches in depth. It weighs 2.45 Kg. The enclosure power source is a board mountable 60 Watt DC/DC converter. This switch mode converter is capable of providing 5VDC at 6.0 A, and ± 15VDC at ± 1.0 A from an 18-36VDC supply.

The VIS6 also has a front display meter that is a 3 1/2 digit true RMS meter capable of measuring combined AC and DC voltage to ± 20 volts. The display allows the user to examine the RMS level of the signal from each load cell, or the calibrated and transformed axial and radial loads. A diagnostics breakout is also provided so that a user can monitor and or measure the signals from the load cells and the drive signals to the actuators directly.

The central component of the actuator power amplifier circuit is an op-amp capable of providing up to 500 mA of continuous output current. Additional features of this amplifier include a programmable Enable/Disable pin allowing remote shutdown for power conservation, as well as adjustable output current limiting though an indirect sense resistor. The amplifier circuit is designed to deliver a current proportional to the output of the corresponding strut drive channel. The gain is 6.67 mA/V.

The signal conditioning enclosure provides local amplification of the load cell signals to minimize the effect of interference on the piezoelectric's high impedance signal. The enclosure houses the charge to voltage load cell conditioning, and resetable fuses in series with the actuator. The enclosure weighs 350 g. Miscellaneous cables add another 95 grams of mass.

Signal conditioning for the piezoelectric load cells is performed in two stages. The first stage is a charge amplifier that produces an output voltage in proportion to the charge generated by the piezoelectric. The second stage removes the high frequency signal content with a state variable active filter. The first three amplifiers and surrounding components of this IC form an inverting second order low pass Butterworth filter. This filtered signal is then sent to an A/D converter corresponding to the strut channel.

3.3 Control Approach

A feed forward filtered-x least mean square (FXLMS) algorithm was implemented as the basis for the active vibration control. FXLMS is well known within the active noise control community. A detailed description is found in Kuo and Morgan (1996). FXLMS is a version of the LMS algorithm in which the reference signal, designated by x, is filtered through an estimate of the secondary path dynamics before it is used in the output and primary plant update.

The reference signal x is assumed to be filtered by an unknown set of primary path dynamics represented by $P(z)$. The result of this filtering is the signal d. This signal corresponds to the vibration produced by the cryocooler in response to a drive signal. There is also a secondary path, $S(z)$, corresponding in the present case to the transfer function between the control actuator and the error sensor. Although $S(z)$ is not known perfectly, a good model, $\hat{S}(z)$, can be found through system identification. That model is used to filter the reference signal, and feed the LMS adapter to adapt the weights that describe the filter $W(z)$. The result is the minimization of the error, e.

The case of a multi-tonal input is a special one for which it is possible to streamline the FXLMS algorithm (Figure 8). The response, d, through the primary path is the sum of the responses $d(1)$, $d(2)$, …, $d(m)$ to m different tones. A reference or synch signal feeds the controller. A sinusoid at the fundamental drive frequency is extracted from the reference. This allows the creation of a cosine for that tone as well as sines and cosines for integer multiples of the fundamental. Only two weights, $w0$ and $w1$, are necessary to characterize the response at each frequency.

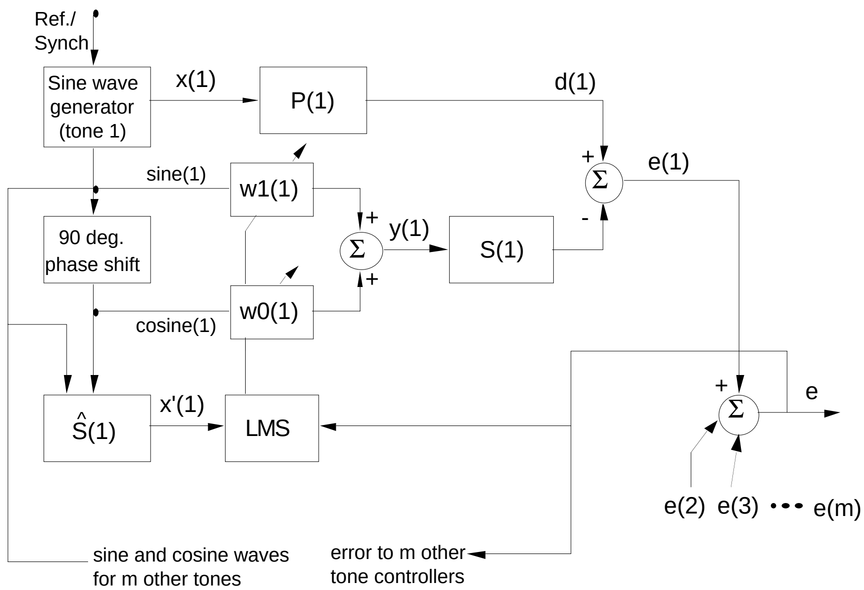


Figure 8: Specialized multi-tonal format of the FXLMS algorithm

3.4 VIS6 System Performance

The feed forward filtered-X least mean squares control algorithm, as implemented on a TI TMS320C6201 fixed point DSP, was executed at 2.5 kHz and reduced five disturbance tones (the primary and four harmonics). Depending on a user decided configuration, at start up the control system either enters a system ID mode or a sensor only mode. If system ID mode is chosen, the algorithm drives each actuator sequentially with a progression of sinusoids at the cryocooler disturbance tones to be controlled (5) and measures the response of the system with the local strut load cell. This information is then used to form an estimate of the secondary path transfer function. After this system ID is complete, the main control mode is enabled and can be started at a user controlled time. Once control is initiated, a period of gradual convergence, the disturbance tones are generally reduced by more than an order of magnitude (usually more than 26 dB) over the bandwidth of operation (240 Hz) as shown in Figure 9. Over two orders of magnitude (40.7 dB) in disturbance force reduction was observed for the fundamental tone. The code uses 350 of the 400 microseconds (µs) available per clock cycle. Of this time the code currently uses 350 µs. Of this, about 25 µs are used for the overhead operations. As approximately 65 µs are required per tone to be controlled, the remaining 325 µs of used ISR time is needed to implement the weight update and output calculation routines for the five tones being controlled.

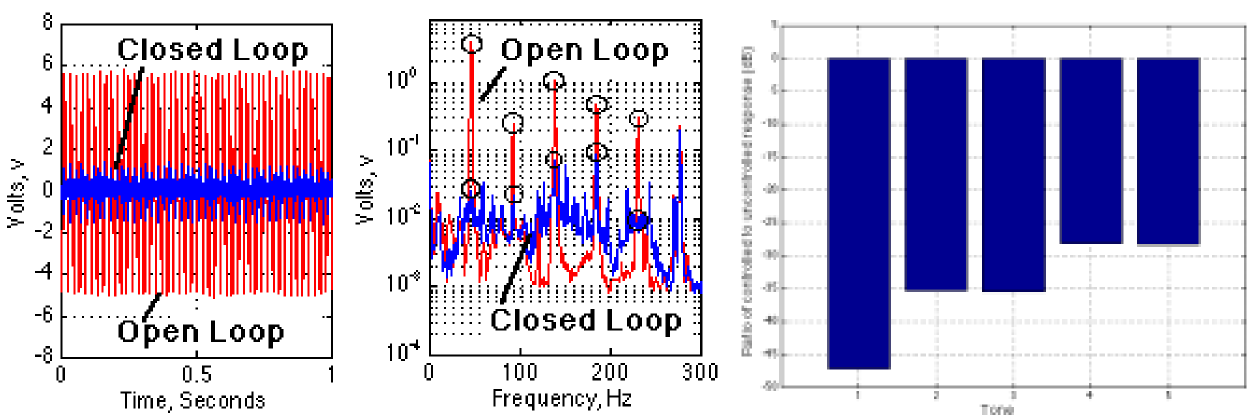


Figure 9: VIS6 system performance in the time, frequency and tonal dB down domains.

4. VRS1

Another system, called Vibration Reduction System - 1 Axis (VRS1), was also developed. It is a pure actively controlled vibration reduction system designed to eliminate axial expander head disturbance loads. This system was developed as an inexpensive, lower-end, vibration reduction system as compared to the VIS6. It consists of an electromagnetic reaction mass actuator, actuator/cryocooler expander head interface, error sensor (in this case a commercial off the shelf accelerometer) transducer cable, electronics support box (containing sensor signal conditioning, TMS320C32 DSP, actuator power amplifiers and cryocooler drive signal pick-off). The only input it requires is a clock signal from the cryocooler drive and power. A schematic along with a picture of the actual hardware of the system is shown in Figure 10. The system weighs 1.3 Kg (of which 88% is associated with the electronics, cabling, and power conditioning) and requires 9.6 W in full-up standby mode. An additional 3.4 W are used when actively controlling. These power levels are small compared to the 60-80 Watts the total cryocooler system consumes when operating. It is easy to retrofit onto existing systems and only requires access to the cryocooler drive signal and 18-36 V DC power.

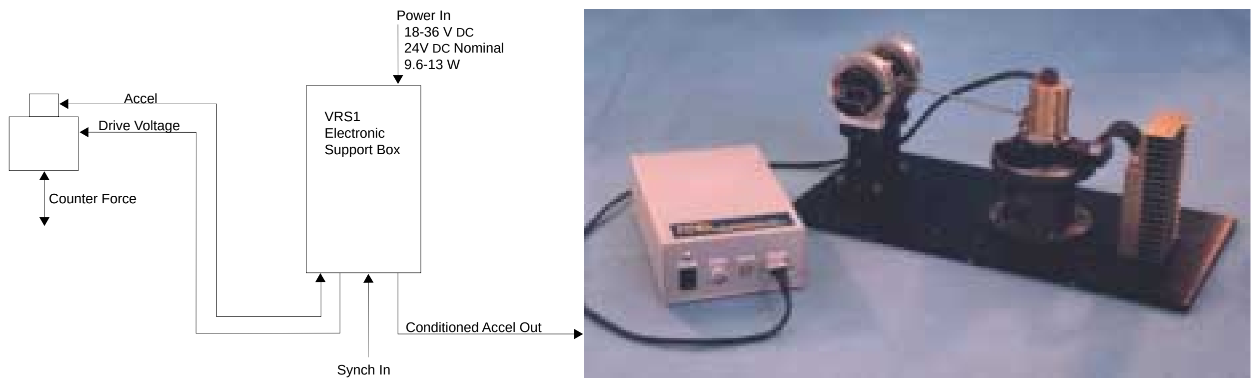


Figure 10: Schematic and picture of the VRS1 hardware mounted on the trial cryocooler expander head and compressor support structure

4.1 VRS1 Actuator

For the VRS1 system a rugged, compact, efficient moving magnet reaction mass actuator that was 1.39 cubic inches in volume weighing 3.8 ounces (107 grams) was developed. The outer diameter of the actuator housing, 1.305 in., was set by the diameter of the cryocooler expander head. The height of the actuator, 1.00 in., was set by the force requirements. A schematic and picture of the actuator is shown in Figure 11. Experimental measurements of the actuator transfer function, at various drive levels are also shown in Figure 11.

The actuator primary suspension resonance, occurring around 38-40 Hz, is clearly visible in the data as is the linear behavior over a wide range of drive levels. Test data is not shown above 300 Hz due to the presence of test system dynamics. In operation the actuator is used at control bandwidths exceeding 500 Hz.

Figure 11: Schematic, picture (in mounting sleeve with US quarter shown for reference), and performance of the VRS1 actuator

5.2 VRS1 System Electronics

The VRS1 ESP, shown in Figure 12, is housed in a steel and aluminum enclosure that is 5.57 inches wide, 2.52 inches high and 8.95 inches in depth. The internal power source is provided by two DC-DC converters which provide up to 6.0 watts of output via 5VDC at 1.2 A and 25 watts of output via ±15 VDC at 1.23 A from an input of 18 to 36VDC.

Figure 12: VRS1 electronic support package

The foundation of the ESP is a single board computer employing a Texas Instruments TMS320C32 floating point digital signal processor (DSP) clocked at 60 MHz. On board peripherals provide 16 bits of digital I/O and up to 4 channels of 16 bit D/A and A/D conversion. For the VRS1, 1 channel of the digital I/O, one D/A, and one A/D channel were used. Software embedding is done via either RS232 serial, or in line memory scan and programming (JTAG) for up to 512 K of ROM.

The implemented actuator power amplifier circuit is a unity gain inverting power op-amp, capable of providing up to 3 A of continuous output current. Additional features of this amplifier include a programmable Enable/Disable pin allowing remote shutdown for power conservation, as well as adjustable output current limiting though indirect sense resistors. The output voltage is applied across the PMA to ground. The VRS1 error sensor is a commercial off the shelf (COTS) internal charge pump (ICP) accelerometer with integral local signal conditioning. The required constant current supply and gain amplification was performed inside the ESP.

4.3 VRS1 System Performance

The same control algorithm implemented for the VIS6 system was implemented in the VRS1 system. Upon start up, the control system enters a system ID mode where it drives the actuator with a progression of sinusoids at the cryocooler disturbance tones to be controlled (11) and measures the response of the system with the sensor. This information is used to form an estimate of the secondary path (i.e. the actuator to sensor) transfer function. After this is complete the main control mode is entered and remains activated until power down. This system reduces 11 disturbance tones (the primary and the first 10 higher order harmonics), as shown in Figure 13. The primary and second harmonics, the dominant axial disturbances, were reduced by over two orders of magnitude (42.5 and 40.5 dB). Over the bandwidth of operation (550 Hz), the measured axial RMS acceleration disturbance levels were reduced between 14 to 40 dB.

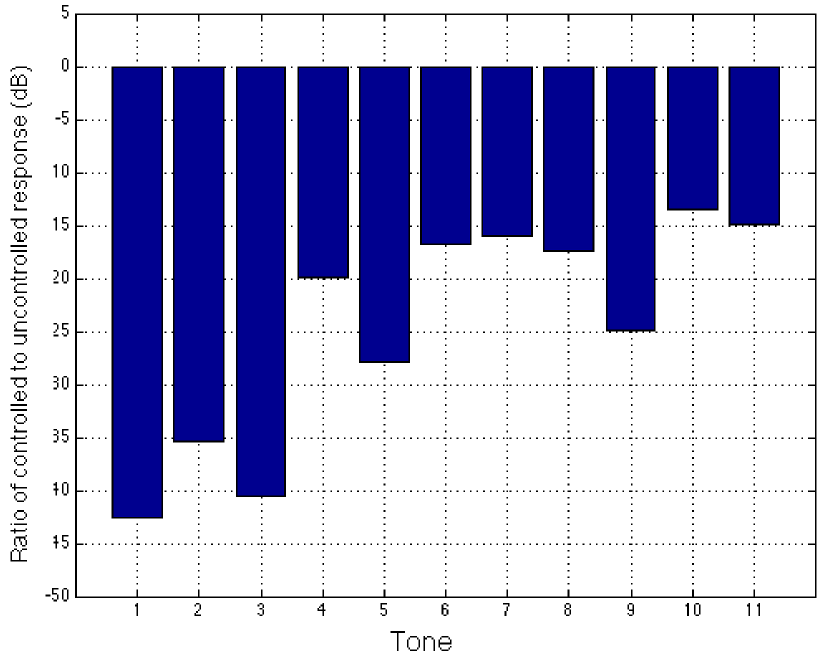


Figure 13: VRS1 system performance in the time, frequency and tonal dB down domains

5. SUMMARY

Two complete drop-in, standalone, active vibration control systems for reducing cryocooler disturbance levels were discussed. The first system tackles all six possible disturbance degrees of freedom and the second system concentrates on the axial disturbances. Both systems achieved reductions of more than two orders of magnitude with power requirements well below that needed to run the cryocooler itself. The systems show considerable promise for reducing the effects of cryocooler disturbance loads, and can also be extended to the isolation of reaction or momentum wheel disturbances. Being outgassing, vacuum, and clean room compatible, the developed technology can easily be applied to specific needs in the semiconductor manufacturing industry. The robust nature of the actuators, and the wide range of stand-alone or PC supported control hardware, allows it to be readily considered for use in a host of industrial manufacturing situations.

ACKNOWLEDGEMENTS

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