Vibration suppression of advanced space cryocoolers — an overview

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ABSTRACT

Mechanical cryocoolers represent a significant enabling technology for precision space instruments by providing cryogenic temperatures for sensitive infrared, gamma-ray, and x-ray detectors. However, the vibration generated by the cryocooler's refrigeration compressor has long been identified as a critical integration issue. The key sensitivity is the extent to which the cooler's vibration harmonics excite spacecraft resonances and prevent on-board sensors from achieving their operational goals with respect to resolution and pointing accuracy. To reduce the cryocooler's vibration signature to acceptable levels, a variety of active vibration suppression technologies have been developed and implemented over the past 15 years. At this point, nearly all space cryocoolers have active vibration suppression systems built into their drive electronics that reduce the peak unbalanced forces to less than 1% of their original levels. Typical systems of today individually control the vibration in each of the cryocoolers lowest drive harmonics, with some controlling as many as 16 harmonics.

A second vibration issue associated with cryocoolers is surviving launch. Here the same pistons and coldfingers that generate vibration during operation are often the most critical elements in terms of surviving high input acceleration levels. Since electrical power is generally not available during launch, passive vibration suppression technologies have been developed. Common vibration damping techniques include electrodynamic braking via shorted motor coils and the use of particle dampers on sensitive cryogenic elements.

This paper provides an overview of the vibration characteristics of typical linear-drive space cryocoolers, outlines their history of development, and presents typical performance of the various active and passive vibration suppression systems being used.

Keywords: vibration suppression, cryocoolers, particle dampers, electrodynamic braking

1. INTRODUCTION

Space cryocoolers are miniature refrigerators designed to cool sensitive components to cryogenic temperatures. Typical temperatures range from the boiling point of liquid Nitrogen at 77 K (-321°F), down to the boiling point of liquid Helium at 4 K (-453°F). Cryogenic temperatures enable the operation of many space-science devices such as infrared detectors and focal planes, solid-state gamma-ray detectors, and emerging superconducting technologies.

Many near-term and future space-instrument programs within NASA and the DoD depend upon the successful application of long-life, low-vibration space cryocoolers.[1,2] Cooler-generated vibration has long been identified as a particularly important parameter for most space applications, and research on cryocooler vibration suppression reached a high level in the early 1990s after the first long-life Oxford Stirling coolers became available.[3]

The key issue is the extent to which cooler-generated vibration excites structural resonances that prevent on-board sensors from achieving operational goals. The lowest vibration levels are achieved with cryocoolers such as Joule-Thomson (J-T) or turbo-Brayton coolers that operate at frequencies far removed from spacecraft structural resonances. J-T coolers involve a very-low-flow-rate, steady (zero frequency) stream of gas that can be routed to cool sensors highly remote from the compressors themselves. At the other extreme, Turbo Brayton coolers incorporate tiny high-speed turbines running at 200,000 to 800,000 rpm, well above spacecraft resonant frequencies. However, most space coolers are Stirling and pulse tube coolers that have compressors with fundamental drive frequencies in the range of 30 to 60 Hz; they generate vibratory forces in a series of discrete harmonics at their operating frequency and multiples thereof.

Figure 1 illustrates the classic Oxford-style Stirling cooler[4,5] whose design is the basis of nearly all long-life space Stirling and pulse tube cryocoolers. This original unit weighs approximately 3.9 kg, is about 20 cm in length, and draws approximately 30 W of electrical power to produce 0.8 W of cooling at 80 K. The key feature of the Oxford-style cooler is the use of linear-motion spindle assemblies within the compressor and displacer that are suspended on

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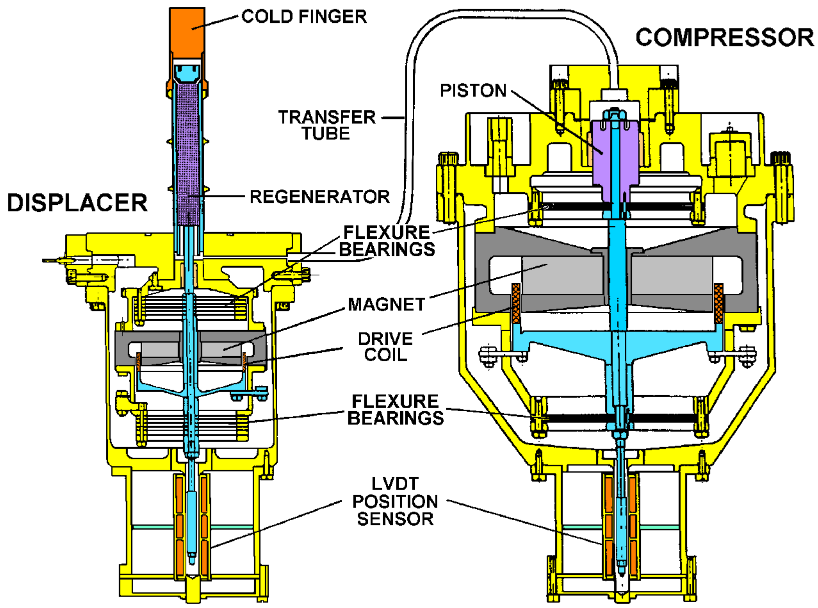

Figure 1. Cross-section showing the internal construction features of the classic Oxford 80 K Stirling cryocooler.

spring flexures to avoid any rolling or sliding contact. The absence of friction, wear, and lubricants is critical to achieving the multi-year operational life required of space cryocoolers.

As noted in Fig. 1, the moving piston and displacer shafts are driven via moving coils in a permanent-magnet field, similar to the drive of a conventional loudspeaker. The required fixed-frequency, variable-stroke mechanical motion is generated by applying an alternating current through the coils at the selected drive frequency—typically a frequency in the range of 30 to 60 Hz; the frequency is chosen to optimize the thermodynamic performance of the cooler. To achieve high motor drive efficiency, the moving piston assembly is tuned to have its mechanical resonance at this operating frequency.

To achieve acceptably low vibration levels, most Oxford-style space cryocoolers incorporate some sort of vibration suppression system based on momentum cancellation using head-to-head compressors and displacers, or electrically driven active counterbalancers. Figure 2 presents an example vibration spectra of the first commercially available Oxford-style linear compressor, both with and without momentum cancellation. Note that cancellation can be quite effective at the fundamental drive frequency, but tends to be less effective at the higher harmonics.

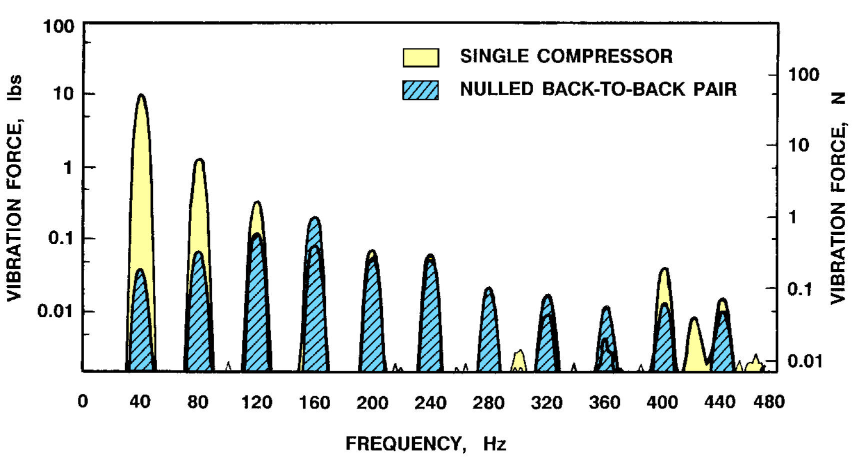

Figure 2. Vibration spectrum generated by a BAe 80K Oxford-style, linear-drive compressor with a 40-Hz drive frequency.

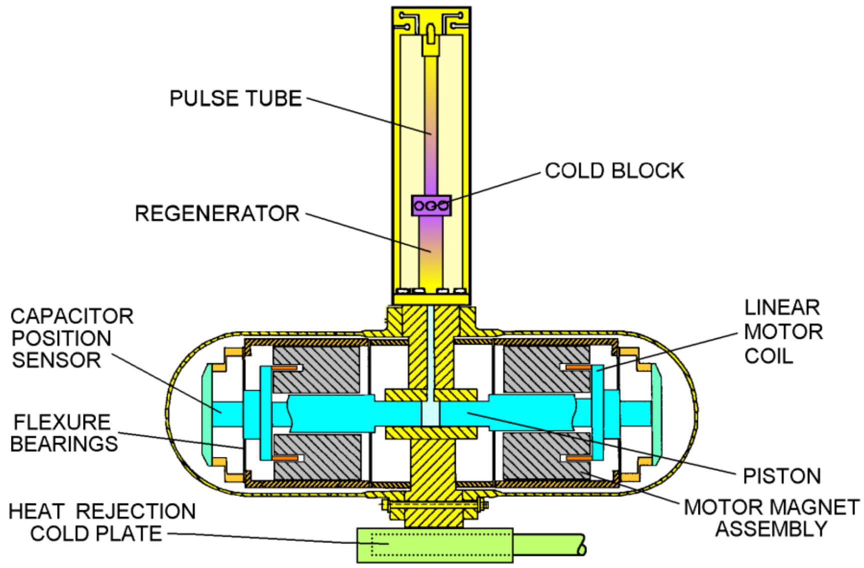

Figure 3. Schematic of modern, dual-piston pulse tube space cryocooler.

To achieve high levels of suppression over a broad frequency range has been found to require the use of active vibration control incorporated into the cooler drive electronics. With this approach the momentum cancellation is actively managed, using feedback based on measured residual vibration forces or accelerations. Because of this need for momentum cancellation, most modern-day cryocoolers involve integral head-to-head compressors such as that shown schematically in Fig. 3.

1.1 Cryocooler Drive Electronics

Although a cryocooler is often thought of as a mechanical device, a major fraction of its design and cost is in its drive and control electronics. In addition to providing power to drive the cooler, space cryocooler electronics are generally tasked to provide a number of functional capabilities such as closed-loop vibration control, closed-loop temperature control of the cryogenic stage, and acquisition and conversion of cooler performance data into digital form for telemetry to the ground. Incorporation of these diverse functions into the electronics often involves embedded processors, digital logic, digital communication interfaces, and sensitive analog circuits.

Figure 4 is a photograph of a typical space cryocooler drive electronics unit, this one for a space pulse tube cryocooler with head-to-head pistons in a single compressor housing as illustrated in Fig. 3. For this cooler, the electronics are tasked with driving two compressor motors and carrying out the many control and data acquisition functions just described.

Figure 4. Typical space cryocooler drive electronics.

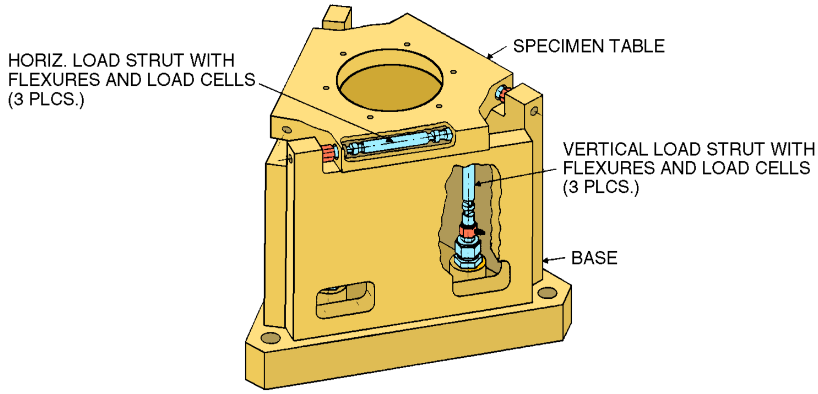

Figure 5. Example cryocooler force dynamometer.

2. MEASURING COOLER-GENERATED VIBRATION

Before examining the details of cryocooler vibration control it is useful to understand how vibration is defined and measured for space cryocoolers. Cooler-generated vibration is defined in terms of the peak vibratory force imparted by the cooler into its supports when rigidly mounted. This force is the reaction force to the moving masses within the cooler that undergo peak accelerations during various phases of the cooler's operational cycle. The accelerations can be from controlled motion, such as the reciprocating motion of the compressor pistons, or natural vibratory resonances of the cooler's elastic structural elements. Although the level of vibratory force that is acceptable is a strong function of the specific application, a value on the order of 0.2 N (0.05 lbs) has gained acceptance as a reasonable design goal.

To measure cooler-generated vibration, specialized vibration dynamometers have been developed, such as the six-degree-of-freedom unit illustrated in Fig. 5.[6,7] Most cryocooler dynamometers use high-sensitivity piezoelectric load cells to measure vibratory forces in a frequency range from 10 to 500 Hz with a typical force sensitivity of 0.005 N (0.001 lb), and a full-scale force capability of 445 N (100 lbs). During operation, the forces (Fx, Fy and Fz), generated in each of the cooler's three axes are simultaneously recorded in real time using a spectrum analyzer.

The results of vibration measurements can be summarized for a variety of operational conditions including different compressor or displacer strokes, coldblock temperatures, and drive frequencies. For space coolers with advanced closed-loop vibration suppression systems, characterizing the effectiveness of these systems is a common objective.

Single-Piston Compressor. Figure 6 displays the measured forces of a second-generation version of the single-piston, Oxford-style compressor shown earlier in Fig. 1. Because the vibration output is a series of discrete harmonics at the cooler drive frequency and multiples thereof, Fig. 6 displays these harmonic levels in all three axes and the vibratory moment about the spindle axis (Mz). Note that the vibration is highest in the piston drive axis (Fz), but that the vibration in the lateral (x and y) cross-axes is just as high or higher for the dominant lateral modes in the 200 to 400 Hz range; this is typical for linear-drive cryocoolers.[8,9] Normally, single-piston compressors such as the one illustrated here would be used in head-to-head pairs to cancel the primary drive-axis harmonics.

Dual-Piston Compressor. Figure 7 displays the measured forces of a representative dual-piston compressor, this one with an integral pulse tube expander as was illustrated in Fig. 3. Notice that the fundamental harmonic is well suppressed in the drive axis, but otherwise the vibration signature is similar in level to the single-piston compressor shown in Fig. 6. To achieve higher levels of vibration attenuation than that shown in Fig. 7 requires active vibration control that is closed-loop on either force or acceleration measurements.

3. CRYOCOOLER VIBRATION CONTROL

Some of the first work on cryocooler vibration suppression started in 1989 with the development of the cryocooler dynamometer shown in Fig. 5. This tool provided the ability to understand and quantify the cooler's vibration, which was a critical first step to developing effective suppression techniques. The first measurement of the cooler's harmonic structure, shown in Fig. 2, illuminated the ease with which high levels of attenuation could be achieved at the fundamental drive frequency, but also highlighted the modest levels of off-axis response and the more difficult challenge associated with achieving good nulling of the higher harmonics.[7]

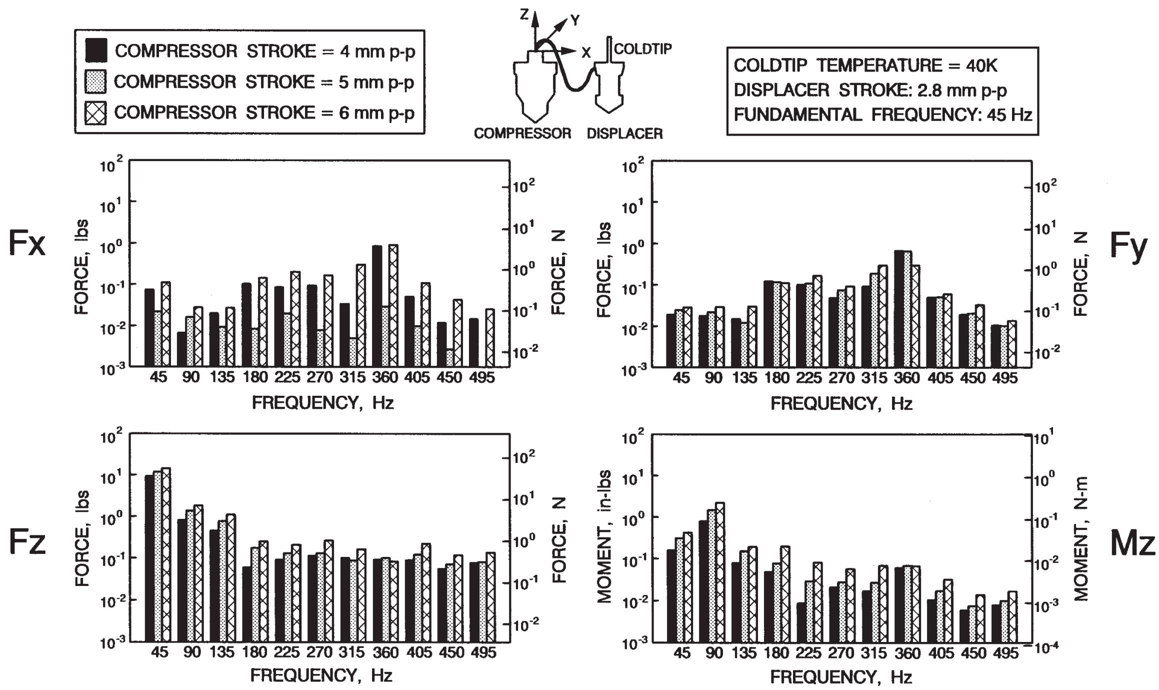

Figure 6. Generated vibration forces of a single-piston Oxford-style compressor.

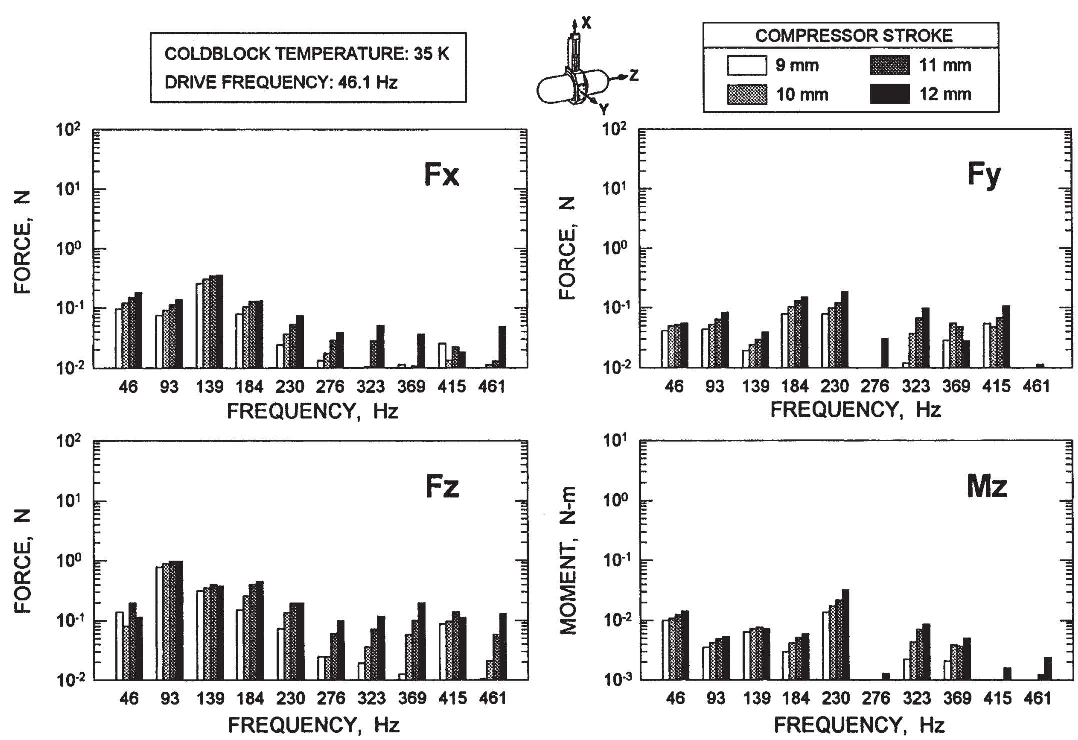

Figure 7. Representative vibration forces of a 10-cc dual-piston compressor.

3.1 Active Vibration Control

In 1990, work by von Flotow[10], Satcon, and others on active vibration suppression became visible to the cooler community, and space coolers were adopted as an ideal application for this emerging technology. Coolers not only had a strong need for vibration control, but they also already contained the voice-coil driven masses and drive electronics needed to implement vibration control algorithms. Thus, minimal added hardware and cost were required to implement active vibration control in cryocoolers.

In the 1991-1997 timeframe a wide variety of vibration suppression algorithms were developed and tested. Key players included Lockheed Palo Alto[11], Satcon[12,13], NASA/GSFC[14,15,16], JPL[17], Hughes[18,19], Rockwell[20], Ball[21], TRW, and MIT[22]. The developed algorithms ranged from step-wise programming of the manual techniques used in the lab to achieve harmonic nulling, to sophisticated adaptive feed-forward algorithms using dedicated DSP processors. Although most algorithms focused only on vibration in the cooler drive axis, at least two investigations developed three axis vibration control.[17,22] A key finding was that the cryocooler plant is quite stable, and a fast, computationally intensive algorithm is not really required for effective suppression.

Starting in the 1994 timeframe, mature versions of the adaptive feed-forward algorithms were introduced into flight cryocooler drive electronics.[23,24,25] Although the first Oxford cryocoolers in space lacked vibration suppression, by 1997 nearly all space cryocoolers contained active vibration suppression as a standard feature.

Experience with the algorithms over the years has been quite positive, with current work centered on driving cooler costs down by further reducing the software and hardware complexity associated with the cooler drive electronics. Other recent research has focused on further reducing the off-axis vibration by refining the internal design features and tolerances used in the manufacture of the coolers.[26]

Figure 8 illustrates the excellent suppression that has been achieved in the drive axis with adaptive feedforward algorithms designed to null various numbers of harmonics. Although these data are for nulling six harmonics, the number of harmonics addressed varies from three or four for the simplest systems, to 16 for the most complex. Most systems are digitally based with digital waveform generation and control.

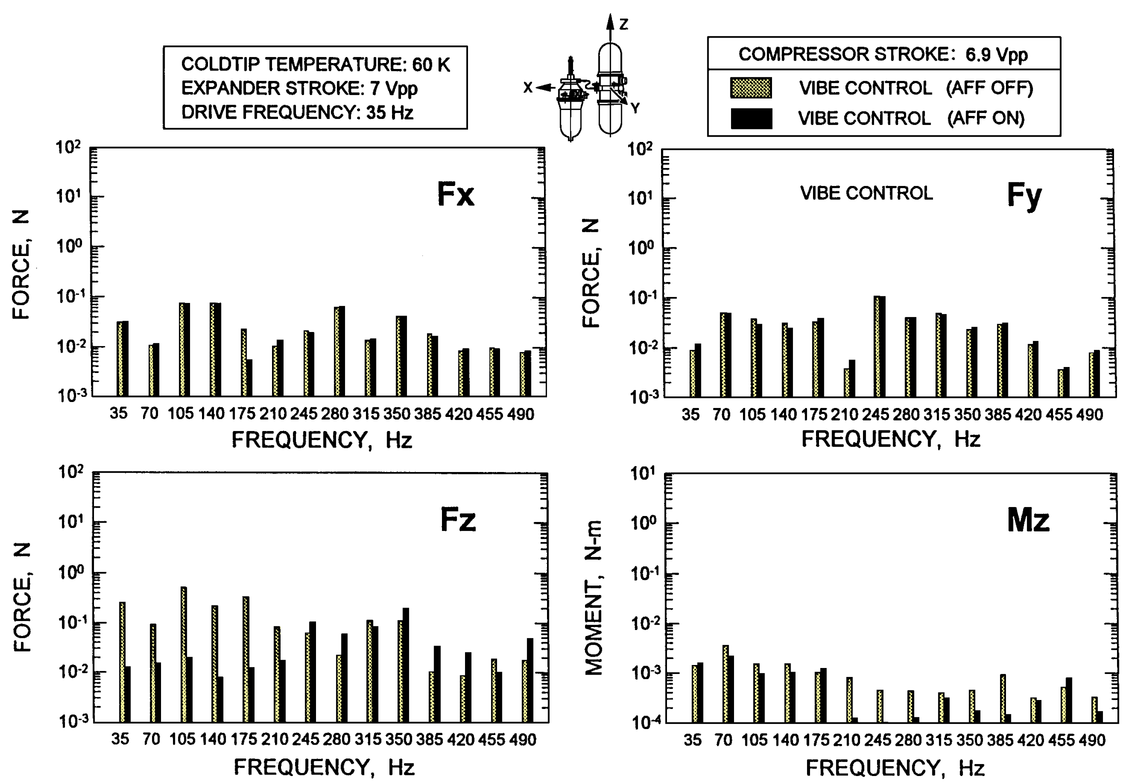

Figure 8. Vibration forces of a dual-piston space Stirling compressor with and without active closed-loop vibration suppression of the first six harmonics.

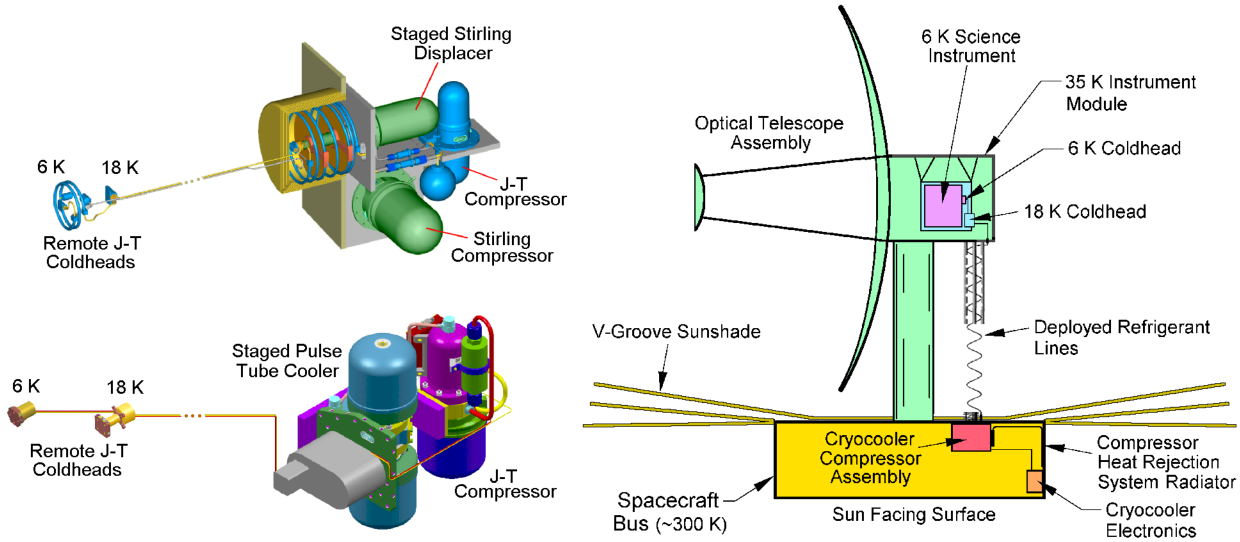

Figure 9. Hybrid J-T cryocooler concepts from Ball Aerospace (top) and Northrop Grumman (bottom) under development for missions such as the James Webb Space Telescope (right) and Terrestrial Planet Finder.

3.2 Vibration Suppression for Large Space Observatories

The current focus of cryocooler vibration suppression research within NASA is on the needs of future large space observatories planned as follow-ons to the Hubble Space Telescope. Two such missions, the James Webb Space Telescope (JWST) and Terrestrial Planet Finder (TPF), plan to use infrared detectors operating between 6-8 K combined with cryogenically cooled infrared telescopes from 3 to 6 meters in diameter. These applications have the room-temperature spacecraft widely separated from the cryogenic telescope and will demand extremely low levels of vibration within the telescope. To address cryocooler vibration for these next-generation missions, NASA has included this low-vibration requirement as an important focus of its recently initiated Advanced Cryocooler Technology Development Program (ACTDP).[27]

Two cooler concepts emerging from the ACTDP effort address this issue of near-zero vibration by using the hybrid Stirling/J-T or Pulse tube/J-T systems illustrated in Fig. 9. Although the compressors of such systems are expected to have vibration signatures similar to those in Fig. 8, the ultra-low-vibration J-T coldheads are designed to be remotely located as far as 25 meters away from the room-temperature compressors. This integration concept allows additional spacecraft-mounted vibration suppression systems to isolate the sensitive telescope from low-level vibration sources, such as the cryocooler compressors, that are on board the spacecraft.

4. DESIGNING FOR AND SURVIVING LAUNCH ACCELERATION LEVELS

A second area where vibration suppression is important to space cryocoolers is in surviving launch acceleration levels. In general, space cooler compressors and displacer bodies have little difficulty meeting representative qualification-level random vibration launch environments of 0.3 g^2/Hz. However, a classic problem with cryocoolers is the fragility of the coldfinger when exposed to launch loads and the sensitivity of the flexure-supported pistons to low frequency launch excitation.

4.1 Minimizing Coldfinger Dynamic Loading by Providing a Bumper or Damping

The classic problem of coldfinger fragility is driven by the fact that the structural robustness of the coldfinger is in direct competition with achieving minimum thermal parasitic loading from conduction down the coldfinger. In addition, without added damping, coldtip amplification factors (Qs) often range as high as 50 to 80. Due to emphasis on thermal optimization, most cryocoolers require an added launch restraint or added damping to survive launch when the typical coldlink interface mass is attached to the coldfinger; this added mass is often in the range of 50 to 100 g.

There are two common means of limiting launch loads on the coldfinger: coldfinger bumper assemblies, and add-on damper assemblies.

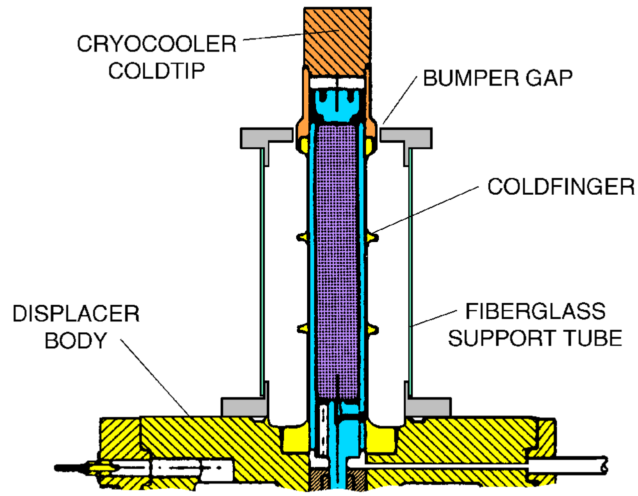

Figure 10. Example coldfinger bumper assembly supplied by BAe as an attachment for its Stirling cryocoolers.

Coldfinger Bumper Assemblies. A coldfinger bumper assembly is a separate structural support designed to limit the maximum dynamic deflection of the coldfinger during launch vibration. To avoid imparting a static deflection or parasitic thermal conduction path to the coldfinger, the redundant structure provides bumpers that are separated away from the coldfinger by a very small gap, typically around 0.1 mm (0.004 - 006 inch). The gap is sized by the maximum deflection that can be withstood by the particular coldfinger without risking fatigue cracking during vibration testing and launch. Because the bumper assembly has to be in close proximity to the coldfinger, it invariably must become an integral part of the coldlink and cryogenic thermal insulation implementation. Figure 10 illustrates a typical coldfinger bumper assembly that was supplied by British Aerospace (BAe) with their line of space Stirling cryocoolers in the early 1990s. The disadvantage of such a bumper assembly is that it can complicate the installation of multilayer thermal insulation (MLI) used to thermally insulate the cryocooler coldtip.

Coldfinger Damper Assemblies. As an alternative to the coldfinger bumper assembly, one can limit the dynamic response of the coldfinger to launch vibration inputs by adding damping to its motion. One significant source of damping is the thermal flexible braid or S-link assembly commonly used to attach the coldfinger to its load. These assemblies provide a modest degree of damping ($Q \approx 20$) due to the internal rubbing that occurs between the assembly's many wires and foils; often this is enough to allow the coldfinger to survive the required launch vibration levels.

If a higher level of damping is required, an ideal means of providing it is via a particle damper. Such a damper consists of a small container of lead or tungsten shot attached to the coldfinger tip. Figure 11 illustrates the dynamic performance of the particle damper used on the AIRS[25] cryocooler. Notice that this damper drops the maximum amplification (Q) of the pulse tube coldfinger from 40 to 10, a factor of four reduction. Note also that the resonant frequency was not appreciably affected, implying that the shot mass "floats" during dynamic excitation and does not add directly to the first-mode modal mass of the pulse-tube cold end.

5. ELECTRODYNAMIC DAMPING OF COMPRESSOR PISTONS

Most cryocooler-sized space assemblies have their resonant frequencies well above 100 Hz so that they do not couple with low-frequency spacecraft and launch vehicle vibration modes in the 10 to 50 Hz frequency range. In sharp contrast, cryocoolers have their fundamental drive frequency tuned in the range of 30 to 60 Hz, and commonly have highly resonant piston and balancer vibration modes as low as 20 to 30 Hz. A particularly sensitive vibration mode is the in-phase piston response of a dual-piston linear compressor when it is unpowered. In this mode the two pistons travel in the same direction at the same time and do no gas compression. This mode has a strong coupling to launch excitation and has a high amplification factor ($Q \approx 30$).

During system qualification it is important for the cryocooler's low-frequency vibration modes to be specifically addressed, both in terms of establishing low-frequency vibration test requirements, and introducing special launch-vibration latches or dampers for the cryocooler internal drive assemblies. For assemblies with motor drives, a favored

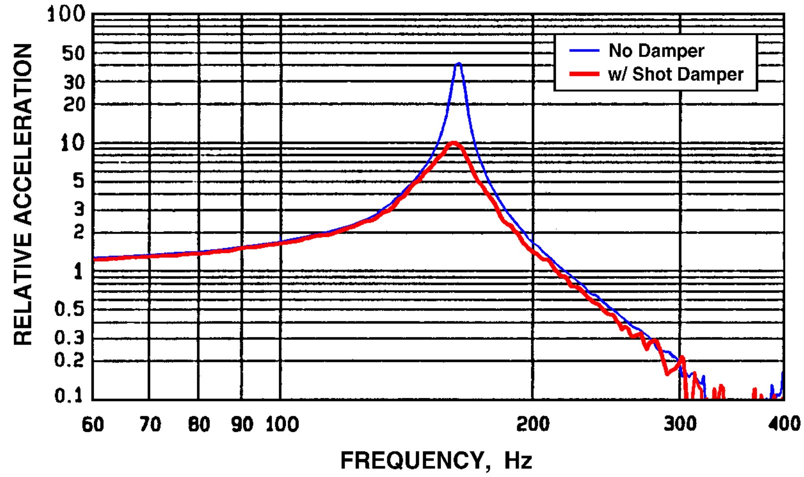

Figure 11. Performance of shot damper used on the AIRS pulse tube to reduce the launch induced acceleration response by a factor of four.

means of introducing launch restraint is to short the drive motor coils during launch. Alternatively, some space systems power the coolers during launch and use closed-loop piston servo control to maintain the pistons in a centered position.

5.1 Computing Piston Response to Low-Frequency Launch Acceleration

The mechanical elements of the compressor closely approximate a classic single-degree-of-freedom spring-mass system with its natural frequency f_o (Hz) defined by its total spring stiffness K (N/m) and moving mass M (kg). The degree of damping is described by the ratio ζ of the damping coefficient C (N·sec/meter) to the critical damping coefficient C_c (N·sec/meter), which is defined by

$$C_c = 2\sqrt{KM} = 4\pi M f_o \tag{5.1}$$

The fundamental equation describing the amplitude of vibration for a given launch acceleration level ($\ddot{x}$) is given by

$$x/x_o = 1\Big/\sqrt{[1-(f/f_o)^2]^2+[2\zeta(f/f_o)]^2} \tag{5.2}$$

where x = piston/displacer motion amplitude (o-p), meters; x_o = deflection amplitude (meters) under a static acceleration of $\ddot{x}$ (i.e., $x_o = m\ddot{x}/K = \ddot{x}/(2\pi f_o)^2$); $\ddot{x}$ = launch acceleration level (o-p), m/s^2; and f = launch acceleration frequency, Hz. Equation 5.2 is plotted in Fig. 12 in terms of the damping ratio ($\zeta = C/C_c$) and the frequency ratio (f/f_o). Given Eq. 5.2, what is needed is a means of predicting the damping ratio ζ associated with shorting the cryocooler drive coils.

5.2 Computing the Damping Ratio Associated with Shorted Drive Coils

When the compressor or displacer drive coils are electrically shorted, the back emf voltage developed by the drive coil generates a current proportional to the coil velocity as limited by the coil circuit resistance. This current develops a damping force (F_c) that opposes the coil motion and is given by[28]

$$F_c = C_e\dot{x} \tag{5.3}$$

where F_c = electrical damping force, N; C_e = electrical damping coefficient, N-s/m; and $\dot{x}$ = coil velocity, m/s.
The components of the electrical damping coefficient are defined by

$$\begin{aligned} C_e &= (BL)^2 \cdot R \\ &= (\text{Newton/amp})^2 \cdot R \end{aligned} \tag{5.4}$$

where B = magnetic flux density in the gap (Tesla); L = length of coil wire in the gap (m); and R = total coil resistance including external wiring, ohms. Note that BL is the motor force constant (N/amp), which can be measured

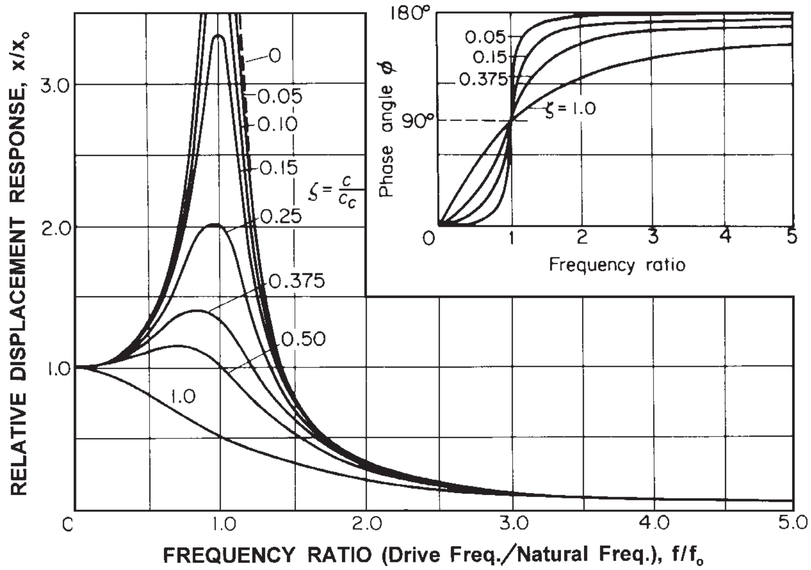

Figure 12. Frequency response of 1-d.o.f. system, such as a cryocooler piston or displacer, when excited by a sinusoidal launch acceleration.

for a particular cooler drive motor. The result of shorting the drive coils thus leads to a new damping coefficient for the cooler defined by $\zeta_{total} = \zeta + C_e/C_c$, where C_e is defined by Eq. 5.4, C_c is defined by Eq. 5.1, and ζ is the measured damping ratio of the unshorted cryocooler.

6.3 Launch Motion Example

As an example, consider the BAe 50-80K cooler for which the resonance properties are summarized in Table 1. For this cooler, the electrical damping coefficient obtained by shorting the 0.77-ohm coil of the compressor, which has a motor force constant of 17 N/amp, is given by

$$C_e = (17 \text{ N/amp})^2 \times 0.77 \text{ ohm} = 375 \text{ N·s/m} \qquad (5.5)$$

When combined with the existing compressor damping value ($\zeta = 0.14$ and $C_c = 169$ N·s/m), this gives a damping ratio of $\zeta_{total} = \zeta + C_e/C_c = 0.14 + 375/169 = 2.4$. The resulting piston amplification at the cooler's resonant frequency ($Q_o = (2\zeta)^{-1}$) during launch is thus $(2\times2.4)^{-1} = 0.2$.

To validate the predictions, the cryocooler was rigidly mounted to a launch vibration test shaker with the cooler axis parallel to the shaker motion. The piston motion was then recorded as a function of acceleration level and frequency using the cooler's built-in position measuring transducers (e.g. LVDTs). Figure 13 presents the piston amplitude measurements for a constant 3-g, 2-oct/minute sine sweep. Three curves are presented in the plot, corresponding to coils shorted, 1.0 ohm shorting resistance, and open circuit (infinite resistance). Notice the dramatic effect of shorting the drive coils on the resulting piston motion.

Table 1. Summary of drive resonance parameters for BAe 50-80K cooler.

PARAMETER	VALUE	
	Compressor	Displacer
Natural Frequency, Hz	50	35
Motor Force Constant, N/amp	17	6
Moving Mass, g	270	37
Spring Stiffness, N/mm	26.7	1.8
Damping Ratio, C/C_c	0.14	0.17
Critical Damp. Coef. (C_c), N-s/m	169	16.3

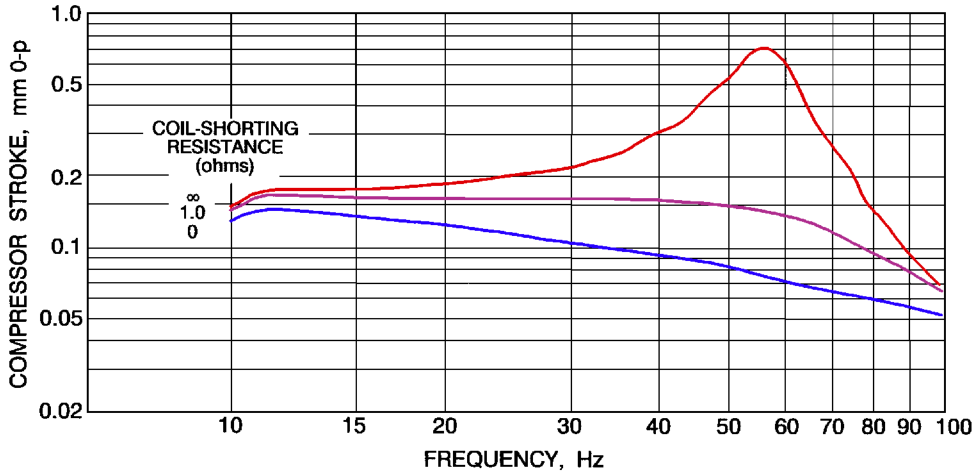

Figure 13. Compressor stroke during 3-g sine sweep at 2 octaves/min versus coil shorting resistance (BAe 50-80K cooler).

SUMMARY AND CONCLUSIONS

The vibration generated by mechanical cryocoolers has long been identified as a critical issue associated with their integration into precision space instruments. The key sensitivity is the extent to which the cooler's vibration harmonics excite spacecraft resonances and prevent on-board sensors from achieving their operational goals with respect to resolution and pointing accuracy. To reduce the cryocooler's vibration signature to acceptable levels, a variety of active vibration suppression technologies have been developed and implemented over the past 15 years. At this point, nearly all space cryocoolers have active vibration suppression systems built into their drive electronics that reduce the peak unbalanced forces to less than 1% of their original levels. Typical systems of today individually control the vibration in each of the cryocoolers lowest drive harmonics, with some controlling as many as 16 harmonics.

A second vibration issue associated with cryocoolers is surviving launch vibration. Here the same pistons and coldfingers that generate vibration during operation are often the most critical elements in terms of surviving high input acceleration levels. Since electrical power is generally not available during launch, passive vibration suppression technologies have been developed. Common vibration damping techniques include electrodynamic braking via shorted motor coils and the use of particle dampers on sensitive cryogenic elements. Data have been presented on the typical performance of these systems, including means of predicting the damping associated with electrodynamic breaking of the cryocooler's linear-motor driven elements.

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