VIBRATION CHARACTERIZATION AND CONTROL OF MINIATURE STIRLING-CYCLE CRYOCOOLERS FOR SPACE APPLICATION

R.G. Ross, Jr., D.L. Johnson and V. Kotsubo

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California

ABSTRACT

A number of near-term precision space-science instruments have baselined the use of miniature long-life space Stirling-cycle cryocoolers. In support of these instruments, JPL is conducting an extensive cooler characterization test and analysis program focused at developing special sensitive performance measurement techniques and identifying means of improving cooler performance. This paper provides a summary overview of the vibration characteristics of split Stirling cryocoolers of the Oxford type and describes means being developed to achieve vibration levels consistent with the exacting requirements of sensitive infrared spectrometer instruments currently under development for NASA applications. A key emphasis of the paper is on exploring both active and passive means of reducing the residual upper harmonics of the drive frequency that remain with nulled back-to-back compressor and displacer units. Vibration supression results, measured with JPL's unique 6-degree-of-freedom force dynamometer, are presented for the 80K Stirling cooler manufactured by British Aerospace.

INTRODUCTION

The emerging line of second generation miniature Stirling-cycle cryocoolers, which are building on the successful Oxford University ISAMS cooler[1,2], are ideally suited to meeting the cryogenic cooling demands of a growing number of NASA space-science instruments. To satisfy the demanding application requirements, these emerging Stirling cryocoolers are successfully addressing a broad array of complex interface requirements that critically affect successful integration to the sensitive instrument detectors. Low vibration and EMI, and improved cooling performance at lower temperatures (55 to 60K) are particularly important parameters.

In January of 1990, JPL took delivery of one of the first long-life 80K Stirling cryocoolers manufactured by British Aerospace (BAe)[3], and began an extensive characterization activity designed to learn from and build upon the Oxford-heritage in a collaborative program with industry to meet the demands of NASA's near-term space-science instruments. Research has focused on all aspects of cryocooler performance including vibration[4], EMI[5], thermal performance[4,6], and reliability[4].

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This paper focuses on cryocooler vibration characteristics and candidate strategies for vibration control in sensitive space-science instruments. Problems occur when cooler-generated vibration excites elastic deflections and resonances within the instrument structure and components that either adversely affect optical alignment, or generate spurious electrical signals. The latter are generated when electrical current-carrying or capacitively-coupled components undergo relative motions. Instruments such as the NASA Earth Observing System (Eos) Atmospheric Infrared Sounder (AIRS), a precision LWIR spectrometer, require focal-plane mechanical stability on the order of 1μm; similar requirements are proposed for instruments such as cryogenically cooled Ge-detector gamma-ray spectrometers.

VIBRATION CHARACTERISTICS OF STIRLING COOLERS

In characterizing cooler-generated vibration it is useful to speak in terms of the peak vibratory force imparted by the cooler into its supports when rigidly mounted. This force is the reaction force to 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 sinusoidal motion of the Stirling compressor piston and displacer, or natural vibratory resonances of the cooler's elastic structural elements. Although no formally agreed upon requirements exist for acceptable vibratory force levels, a value on the order of 0.2 N (0.05 lbs) has gained acceptance as a reasonable design goal.

To help quantify and understand the force levels generated by present cooler designs, JPL has developed the 6-degree-of-freedom force dynamometer facility shown in Figure 1. This dynamometer has a frequency range from 10 to 500 Hz and a force sensitivity from 0.005 N (0.001 lb) to 445 N (100 lbs) full scale. During operation the forces and moments generated about each of the cooler's axes are available simultaneously for real-time quantitative analysis.

Figure 2 illustrates typical vibration-force spectra measured at JPL[4] for single and back-to-back 800 mW 80 K BAe Stirling compressors with low-distortion drive electronics. Notice that considerable vibratory force is present at upper harmonics of the 40 Hz drive frequency and that these upper harmonics do not readily cancel when two coolers are run phase-locked back-to-back for momentum cancellation.

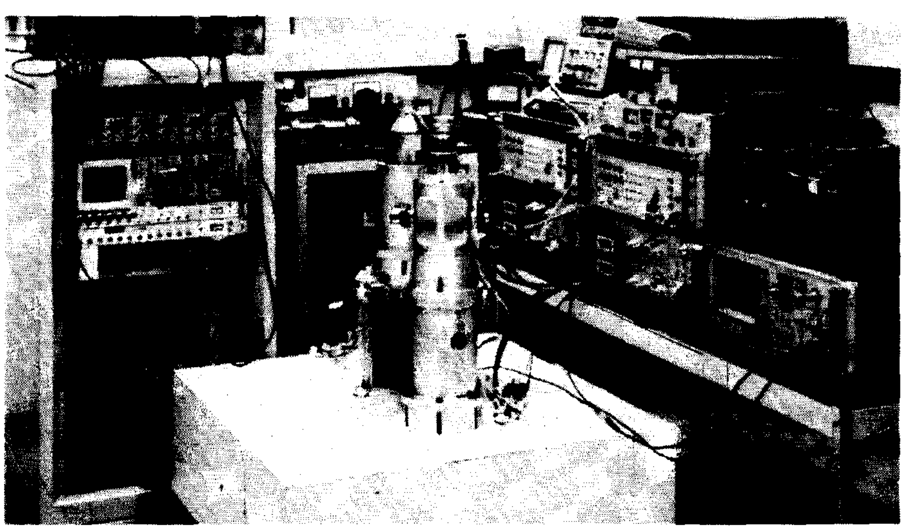

Fig. 1. JPL's force dynamometer facility with BAe coolers undergoing vibration characterization

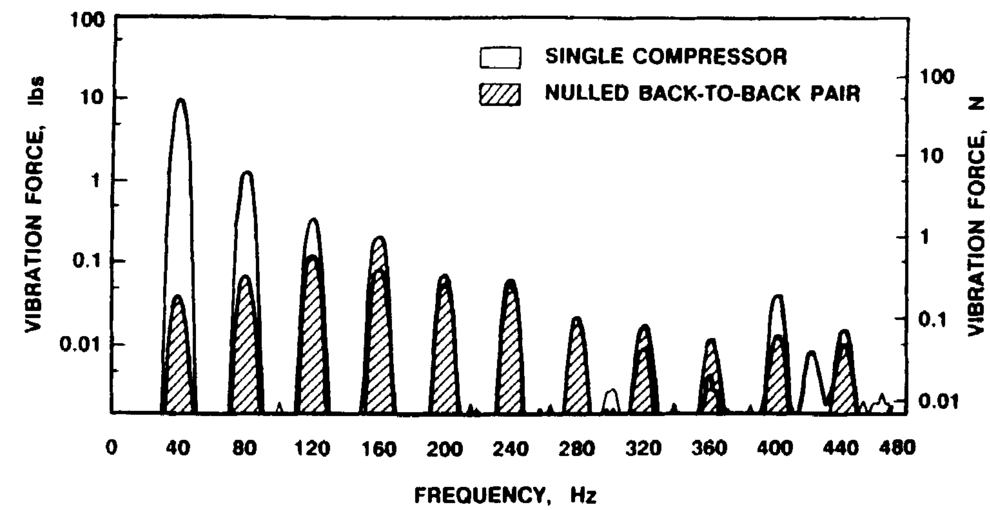

Fig. 2. Vibration force spectra for single and back-to-back BAe 80K compressors

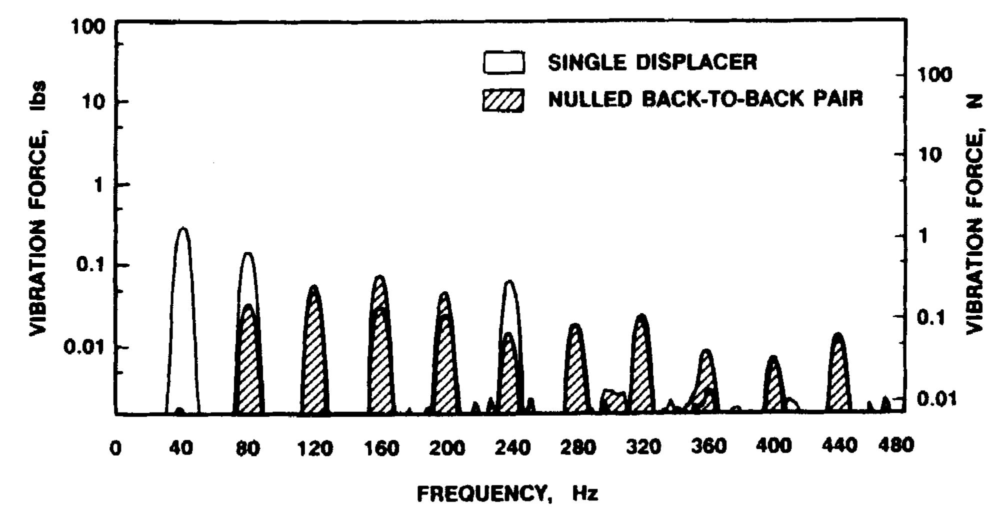

Fig. 3. Vibration force spectra for single and back-to-back BAe 80K displacers

Figure 3 illustrates similar data obtained for single and back-to-back BAe Stirling displacers that mate to the above compressors. Notice that the displacer vibration level is similar to that of the compressors despite its significantly (20X) smaller moving mass. It is hypothesized that this greater relative noise level in the displacer is the result of the displacer being predominantly driven by the gas pressure wave from the compressor, and only secondarily modulated by the displacer drive electronics for accurate phase and stroke control.

Although these vibration levels for back-to-back coolers are near the 0.05 lb goal, the level of vibration cancellation displayed is highly dependent on careful manual nulling with the visibility provided by the force dynamometer and spectrum analyzer. In a flight instrument, means must be found to actively provide this nulling function over the life of the instrument, and hopefully to additionally lower the remaining vibration through incorporation of advanced vibration control techniques.

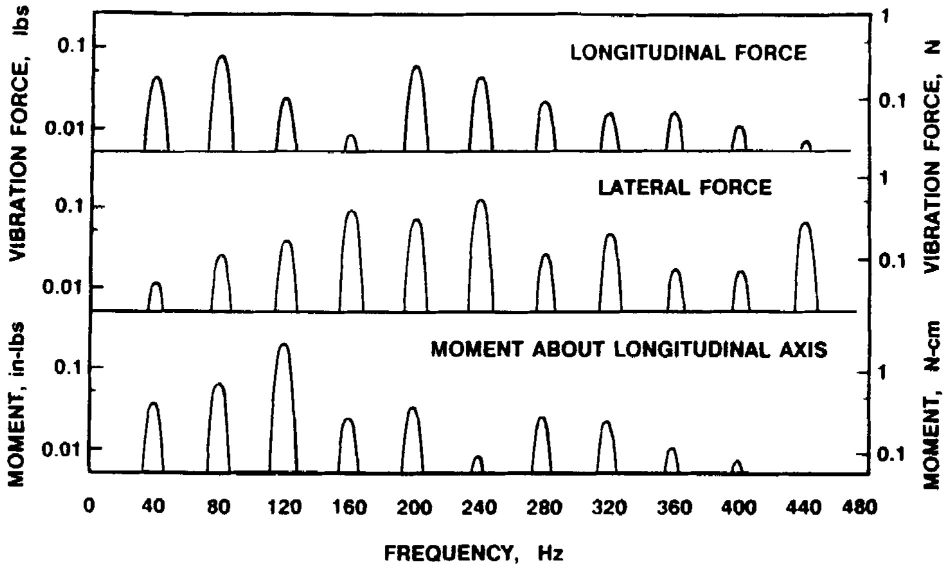

Fig. 4. Low-level residual forces achieved with back-to-back compressors nulled at three harmonics (40, 120, and 160 Hz)

ADVANCED VIBRATION CONTROL TECHNIQUES

In the last five years a variety of powerful adaptive vibration control techniques have been developed, generally based on digital signal processing techniques. These techniques[7] have been addressed to applications as widely varied as automotive noise control and control of vibration of rotating machinery. The general concept is to drive a servo-mechanism in such a manner as to cancel the objectionable vibration as measured by appropriate transducers at one or more important locations.

The back-to-back space cryocooler is ideally suited to these advanced control techniques because it inherently contains the required balancing mechanism (the second cooler) and generates vibration at a limited set of time-invariant frequencies; thus only amplitude and phase control at each important harmonic is required, and adaptive frequency response is not required.

As shown in Fig. 4, the feasibility of selectively nulling individual harmonics has been demonstrated[4] using a technique whereby the fundamental oscillator drive to one of the two back-to-back coolers is modified slightly with the addition of low-level phase-locked signals at multiples of the drive frequency. The amplitude and phase of each of these harmonics is manually adjusted to cancel the residual vibration between the two coolers at the selected frequencies. With the assumption that this nulling condition is stable, one first-order method of vibration control is to permanently incorporate this modified sine-wave into the cooler drive electronics; to deal with operational dependencies, a separate set of nulling parameters can be stored for each important cooler operating state. A recognized limitation of this approach is that, in the post-launch thermal-vacuum space environment or as the cooler ages, the required nulling parameters may change.

One approach to assuring long-term vibration cancellation is to incorporate some form of adaptive selection of the nulling parameters based on continuous or periodic feedback of real time vibration data. Such methods[7] and other digital control techniques under active development elsewhere within the cryocooler community[8] are showing considerable promise in early feasibility demonstrations. The possible limitation of these approaches is the extensive growth in electronics complexity associated with extensive digital signal processing. Selection of the appropriate feedback signal -- compressor piston position, vibration force, or cooler acceleration -- is important, as is making the adaptive control insensitive to disturbances originating from outside the cryocooler (such as from a nearby cooler).

VIBRATION CONTROL THROUGH PASSIVE TECHNIQUES

Although significant advances are being made in reducing cooler vibration through advanced control techniques, passive isolation and damping are also high-leverage and attractive means of minimizing the deleterious effect of cooler vibration. Because of the continuous--as opposed to transient--nature of cooler excitation, the vibration amplitude of instrument elements will increase without limit until the level of excitation energy per cycle matches the energy dissipation due to internal damping. Figure 5 displays this classical response relationship for a single-degree-of-freedom system as a function of the ratio of the drive frequency (cooler vibration harmonic) to the resonant frequency of the responding system (science instrument resonance). For damping ratios between 0.005 and 0.05, typical of aerospace structures, the cooler vibration force is seen to be amplified by factors from 10X to 100X. Unfortunately the level of damping is often lowest for cryogenic structures in vacuum environments; thus, without special consideration these 10X to 100X amplifications are likely to occur in space cryocooler applications.

To quantify the improvement possible with passive techniques, the force amplitude response of a BAe Stirling compressor was carefully measured in the lateral (normal to the piston axis) direction when mounted on a representative rigid aluminum structure with a first mode resonant frequency around 160 Hz. The cross-hatched curve in Fig. 6 displays the order of magnitude vibration reduction obtained when an simple viscoelastic passive damping device was attached to the exterior of the cooler compressor. This relatively high level of improvement, combined with the simplicity and broad-spectrum applicability of passive damping, suggests that incorporation of passive damping techniques should be carefully considered in space-science instruments.

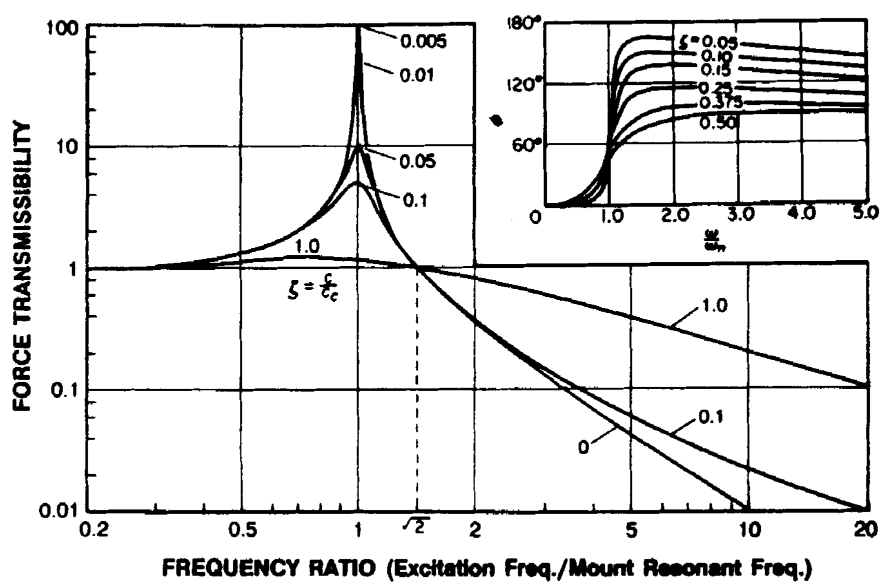

Fig. 5. Amplification of cooler-support interface forces as a function of frequency ratio and damping

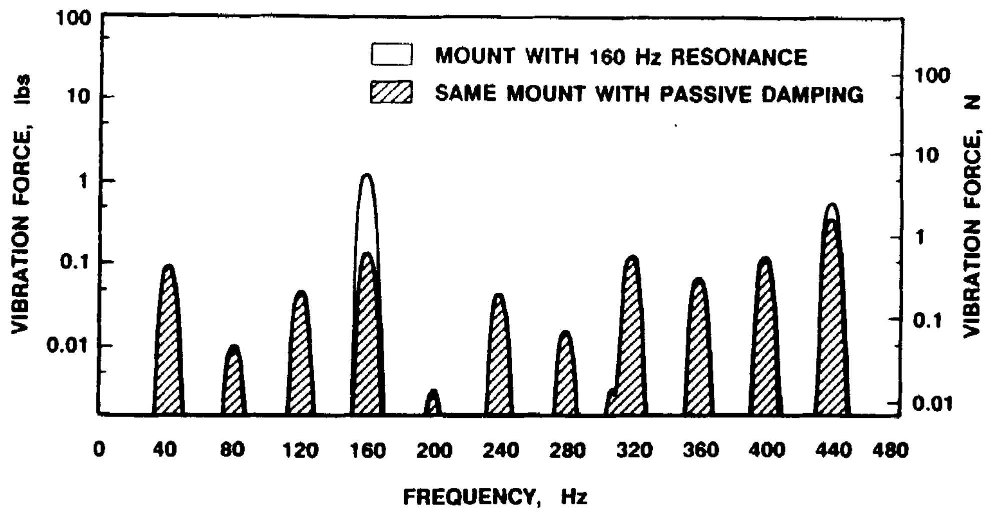

Fig. 6. Effect of damping on lateral force generated by a single BAe compressor mounted on a structure with 160 Hz resonant frequency

As an alternative, or complement, to added damping, Fig. 7 displays the vibration response reduction achieved by optimally positioning the structural resonant frequency midway between cooler drive harmonics, thus minimizing the cooler-structure cross-coupling. The problem is that the close spacing of the cooler vibration harmonics severely constrains the maximum uncoupling that can be achieved to about a 10% frequency offset; as can be noted from Fig. 5, this offset still corresponds to a force amplification factor of around 5x. An additional concern is the stability and predictability of the instrument structural resonant frequencies in the post-launch environment.

A third passive vibration suppression technique is classical vibration isolation achieved by suspending the cooler from the instrument structure with a very compliant mount. This involves operating on the right-hand side of Fig. 5, with the resonant frequency of the cooler/support system well below the cooler excitation frequency. This approach replaces transmitted force between cooler and instrument

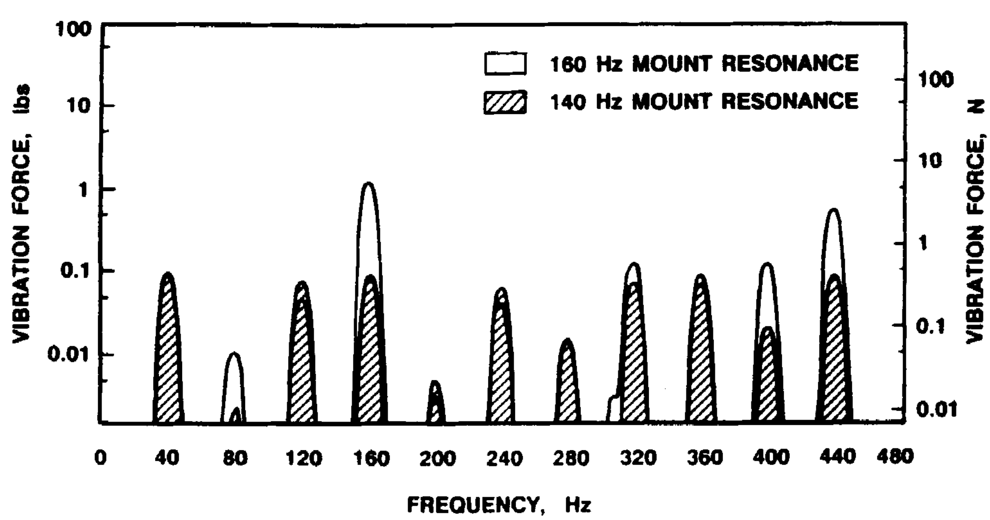

Fig. 7. Effect of offsetting structural resonance to 140 Hz so as not to couple with BAe compressor harmonic at 160 Hz

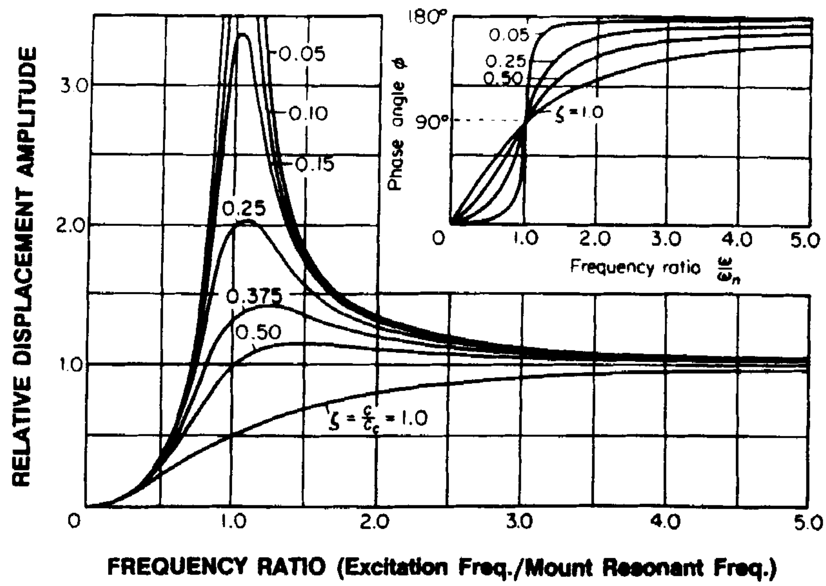

Fig. 8. Displacement amplitude response of soft-mounted cooler as a function of isolation stiffness and damping parameters

with near-constant relative movement between the two as described by Fig. 8. For low damping, the transmitted force with such a system reduces to the spring constant of the suspension system times this relative movement; the force thus falls off with increasing frequency ratio as shown in Fig. 5.

The challenge with passive vibration isolation is to achieve both vibration isolation and launch load survivability without requiring latches during launch. To prevent excessive coupling to launch vehicle resonances, many spacecraft (eg. the NASA Eos platforms) prohibit instrument resonances below 50 Hz; this greatly restricts low-frequency vibration isolation on such systems, or may necessitate complex latching systems. For the displacer, passive isolation is further complicated by the need to minimize motion of the cold finger tip with respect to the remainder of the detector cryostat assembly so as to minimize vibration conducted directly down the cold plumbing to the detector. This is discussed further in the next section.

COLD FINGER VIBRATION ISSUES

Because the cooler cold finger must be attached to the sensitive instrument detector by a high conductivity thermal link, transmission of vibration from the cold finger to the detector is also an important issue that must be carefully addressed. In a representative cryostat assembly, such as shown in Fig. 9, resonant frequencies are likely to occur near the strong harmonics of the cooler drive frequency because of the substantial mass of the thermal conductors in combination with the minimum-gage structural supports required to achieve high levels of thermal isolation. This likelihood of low-frequency resonances increases the probability of a highly resonant response to cold-finger vibration input.

For both vibration isolation as well as accommodation of differential expansion motions upon cooldown, flexible foil or wire thermal links -- shown in Fig. 9 -- are normally used to decouple the cold finger motion from the thermal link. However, even with this good isolation, minimum cold-finger vibration remains important.

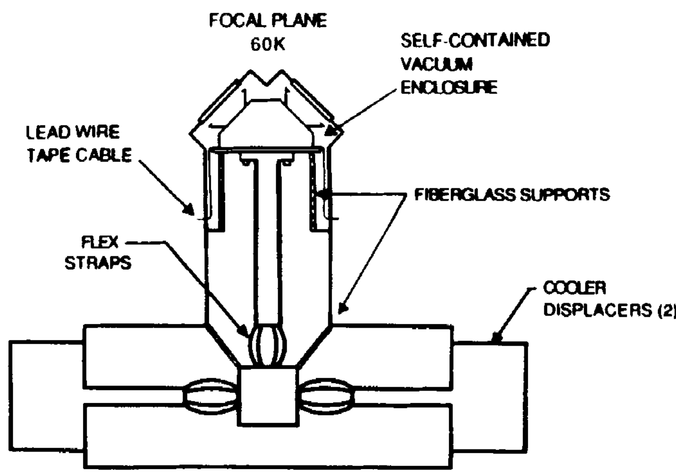

Fig. 9. Example cryostat assembly integrating cooler cold finger to infrared detector

To assess the expected level of cold-tip vibration with a rigid displacer mount, micro-accelerometers were attached to the tip of the BAe 80K cooler cold finger. Displacement spectra, computed from the measured acceleration spectra, are presented in Fig. 10 for both the lateral and longitudinal (finger-axis) directions. The larger 2.5 μm motion in the finger-axis direction at 40 Hz agrees well with the computed dilation of the cold finger in response to the 40 Hz fluctuating pressure of the Stirling cycle.

SUMMARY

Meeting the performance goals of near-term space-science instruments places demanding requirements on long-life space Stirling-cycle coolers. One of the most challenging requirements is achieving acceptably low levels of vibration of the instrument detector and sensitive optical and electronic elements. Advanced development efforts have begun to address these challenges with a broad spectrum of both passive and advanced active control techniques. Major reductions in cooler generated vibration are expected as these concepts mature into flight hardware for the emerging space-science instruments such as the Eos AIRS.

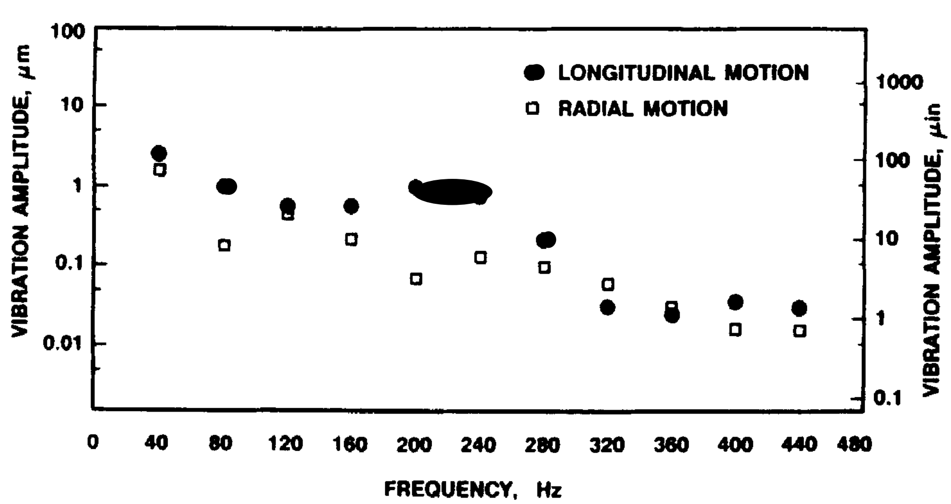

Fig. 10. Measured vibration amplitude at BAe cold finger tip with rigidly mounted displacer

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