Low Vibration Microminiature Split Stirling Cryogenic Cooler for Infrared Aerospace Applications

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ABSTRACT

Spaceborne infrared instrumentation is known to be inherently susceptible to cryocooler induced vibration, the attenuation of which usually relies on actively multi-tonal momentum cancellation. In this approach, a typical single-piston expander is actively counterbalanced by a motorized counterbalancer, and the typical dual-piston compressor is counterbalanced by synchronizing the motion of the opposing moving pistons working in the "master-slave" mode. The feedback signals are usually provided by external vibration sensors (load cells or accelerometers).

Although compliant with the most stringent space requirements, such a conservative vibration control approach can result in using outdated, oversized, overweight and overpriced cryogenic coolers for some applications. Such a "space heritage" practice becomes increasingly unacceptable for space agencies now operating within tough monetary and time constraints.

The authors are advocating a purely passive approach to vibration control, relying on the combined principle of tuned dynamic absorber and low frequency vibration isolator. This concept has the potential to outperform systems of active vibration cancellation with respect to overall system effectiveness and warrants particularly strong consideration for cost-sensitive missions.

The initial target of the development is "Operationally Responsive" space programs, which may well be satisfied by this approach.

INTRODUCTION

Vibration export is one of the key parameters characterizing cryogenic coolers targeted for use on military, commercial and scientific space payloads featuring inherently vibration sensitive spaceborne infrared instruments.

The best available space cryocoolers rely primarily on the historical "Oxford" heritage technology [1-3], where the closed cycle Stirling refrigeration is achieved by the cyclic compression and expansion of the gaseous working fluid via the approximately sinusoidal reciprocation of compressor and expander pistons. Resulting from the unbalanced motion of the above mechanical components is the vibration export, the instantaneous magnitude of which is the product of moving mass and instantaneous acceleration [4]. Normally, the major portion of this vibration export occurs at the driving frequency. The high frequency content (multiples of the driving frequency) may be attributed to a deviation of the acceleration of moving components from the ideal sinusoidal waveform due to nonlinearities in pneumatic and mechanical springs, mechanical friction in clearance seals, high-frequency contamination of driving voltage and current, nonuniform current-to-force transformation in the linear actuator, oversaturation of actuator armature material, etc.

The best practice, therefore, relies on a concept of actively assisted momentum cancellation [5,6]. In this approach, the compressor typically comprises two back-to-back pistons, one of which — "Master" — is driven by a tonal voltage of a constant frequency, the magnitude of which varies automatically so as to maintain the desired temperature of the infrared detector. The second piston — "Slave" — is actuated in a "counterbalancer mode" by a composite voltage comprising the fundamental frequency and its multiples. The magnitudes and phases of each tonal component in the "Slave" voltage are automatically tuned so as to attenuate the residual vibration export over the wide frequency range covering typically up to 10 harmonics of the driving frequency [5]. The typical feedforward active control system is composed of a conditioning board and dedicated controller. The feedback signals (residual vibration) are usually delivered by vibration sensors (accelerometers or load cells).

The principles of vibration export cancellation originated from the expander are similar to the above. The counterbalancing force is produced by a motorized linear counterbalancer driven actively by a separate controller having similar operational principle and the architecture.

Although excellent vibration cancellation is usually achievable with this approach [7], the known drawbacks are the added complexity and, therefore, reduced reliability of control electronics. In general, the control electronics are less reliable than the thermo-mechanical units and, therefore, are the "weakest link" in the overall reliability chain. Furthermore, such electronics typically rely on expensive radiation hardened components. This explains the high incurred cost, which surprisingly, often comes to half of the entire cooler price. Added weight and size can also be issues, especially for the typical mini, micro and nano-satellite applications which are of particular interest to the authors.

Further drawbacks of this approach may be attributed to the need for using vibration sensors to monitor the residual vibration and produce the feedback signals. It appears that overall performance depends on the proper placement of such sensors along with the dynamic properties of support structure. The technology limiting factors include the presence of unpredictable structural resonant amplifications along with the impossibility of separating dynamic responses originated from the expander and compressor.

In addition, the data acquisition hardware needs to operate with full dynamic range and maximum sensitivity, as is needed to accommodate the high and low vibration levels. This limits the **A/D** conversion accuracy and complicates control at low vibration levels when the system approaches the almost counterbalanced condition.

It is important to note also that there are additional penalties, primarily associated with using an auxiliary motorized counterbalancer and dual-piston compressor which is considered to be the standard approach for addressing the vibration export issues, especially when combined with active vibration cancellation. In particular, similar single-piston rivals show essentially better electromechanical performance, higher reliability, smaller size, lower weight and lower manufacturing cost, resulting, of course, in lower recurring price.

Ricor recently reported on the successful development and fielding of the novel model K527 microminiature long-life tactical split Stirling linear cryogenic cooler [8 – 12] for use in a wide range of portable hand held and gyro-stabilized infrared imagers. Technical comparison [10] with coolers of the same cooling power at 80K@23°C indicates that this cooler is the smallest, lightest and the most efficient in the range.

Because of the tight constraints imposed primarily on weight, price and cooling performance, the design of this cooler abandons the above-explained complexities (dual-piston compressor approach, flexural bearings, contactless clearance seals, etc) in favor of mechanical simplicity. The compressor design relies on a "moving magnet" resonant single-piston concept featuring a very light piston assembly and contact seals.

As to the expander design, Ricor accepted the regular "tactical" approach relying on the pneumatic actuation of the resonant mass-spring displacer-regenerator and contact clearance seals. Special emphasis was again given to making the moving assembly lighter.

The feasibility of this approach was proven recently in the course of a highly accelerated life test (including temperature extremes) [13] where a similar cooler driven by a "moving coil" actuator at 60% of its full power lasted in excess of 45,000 hours. The authors believe that removing the driving coil from the cooler interior, thus eliminating the contamination, flying leads and feedthroughs, will greatly improve cooler reliability. Along these lines, the stable and relatively low reject temperature, zero gravity, zero vibration and shock environment typical for a spacecraft vehicle will eliminate most of the risk factors; thus, cooler life might be further greatly extended.

As a part of the effort towards adaptation of leading edge microminiature tactical cryogenic coolers to space applications, in this paper, the authors are disclosing their complex approach to the control of vibration export produced by this cooler.

To start with, the off-axis vibration export produced by the compressor was minimized by design down to the most stringent requirement of less than 0.1N rms over the typical frequency range 0-2000Hz. This was attributed primarily to a very accurate alignment of the lightened moving assembly driven by a symmetric "moving magnet" actuator. The actuator has been designed to produce extremely small side forces and a very uniform current-to-force transformation ratio, being practically independent of the magnet ring position within the working range.

Similar design principles were adopted for the expander portion of a cryogenic cooler. In particular, accurate dynamic design allowed for achieving temperature-independent resonant operation, thus eliminating the need for auxiliary electrodynamic actuation of the expander for corrective magnitude/phase control. The simplest pneumatic actuation using a stepped driving plunger, as used in tactical cryocoolers, appears to be quite adequate. Further design improvements involved accurate alignment of the stepped bushing-plunger seals, use of a flexural link between the displacer-regenerator assembly and driving plunger, use of machined springs producing zero side forces and a special zero-contamination purge/fill procedure. This resulted in essential removal of design constraints, which, in combination with the low-weight moving components, allowed for compliance with the above stringent requirement for off-axis vibration export.

The residual on-axis vibration export produced by the compressor and expander was also reduced as a result of weight reduction and smoothing the reciprocation of the moving components; yet it still did not meet typical space requirements.

In this paper, the authors advocate a purely passive concept of suppressing the above vibration export by making use of a combined concept of tuned dynamic absorber and optimized low frequency vibration mount, as explained in [8-12, 14].

In this approach, both compressor and expander units with in-line mounted tuned dynamic absorbers are compliantly supported from the optical bench or other host structure using optimal single-degree-of-freedom vibration mounts.

The purpose of the tuned vibration absorber is to suppress vibration export at the driving frequency. Along these lines, the vibration isolator delivers the attenuation of vibratory force transmission over the entire high frequency range and dynamically decouples the compressor and expander units from the rest of the support structure. This decoupling is essential for increasing the suppression ratio at the driving frequency.

DYNAMIC MODEL AND FREQUENCY RESPONSE FUNCTIONS

In Figure 1, the two-degrees of freedom model represents the vibration mounted compressor equipped with tuned dynamic absorber, where the stationary support structure (base) is assumed to be rigid and stationary. This assumption is valid when (i) the support structure is heavy enough and (ii) its resonant frequency is significantly higher than the first resonant frequency of the vibration mount and (iii) no resonant high frequency excitation in the support structure takes place. In particular, M_1 and M_2 are the masses of the compressor housing (including clamping) and tuned dynamic absorber, respectively. The combination represents the stiffness and damping of a compliant compressor mounting to the stationary base; combination represents stiffness and damping of the dynamic absorber spring. The time function represents arbitrary force stimulus applied to the compressor. Resulting from the action of this force are dynamic deflections counted from appropriate positions of static equilibrium.

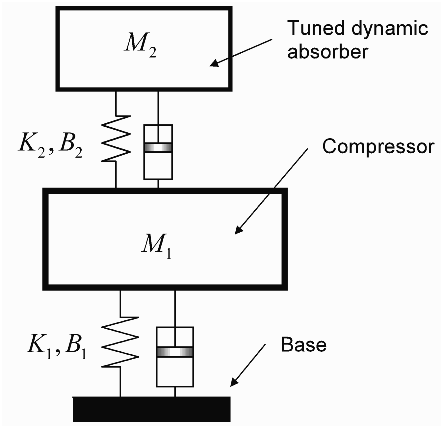

Figure 1. Dynamic model.

In particular, M_1 and M_2 are the masses of the compressor housing (including clamping) and tuned dynamic absorber, respectively. The combination K_1, B_1 represents the stiffness and damping of compliant compressor mounting to the stationary base; combination K_2, B_2 represents stiffness and damping of the dynamic absorber spring. The time function $r(t)$ represents arbitrary force stimulus applied to the compressor. Resulting from the action of this force are dynamic deflections x_1, x_2 counted from appropriate positions of static equilibrium.

The differential equations of motion take the form

$$\begin{aligned} &M_1 \frac{d^2 x_1}{d^2 t} + K_1 x_1 + B_1 \frac{dx_1}{dt} + K_2 (x_1 - x_2) + B_2 \left(\frac{dx_1}{dt} - \frac{dx_2}{dt} \right) = r(t) \\ &M_2 \frac{d^2 x_2}{d^2 t} + K_2 (x_2 - x_1) + B_2 \left(\frac{dx_2}{dt} - \frac{dx_1}{dt} \right) = 0 \end{aligned} \tag{1}$$

The force transmitted to the base (vibration export) is

$$f(t) = K_1 x_1 + B_1 \frac{dx_1}{dt} \tag{2}$$

Using complex Fourier transform, $G(j\omega) = \int_{-\infty}^{\infty} g(t)\, e^{-j\omega t} dt$, where ω is angular frequency and $j = \sqrt{-1}$ is complex unity, we make a transition from time into a frequency domain

$$x_1(t) \Leftrightarrow X_1(j\omega);\; x_2(t) \Leftrightarrow X_2(j\omega);\; r(t) \Leftrightarrow R(j\omega);\; f(t) \Leftrightarrow F(j\omega)$$

and substitute into (1). In doing so, we obtain the set of linear equations

$$\begin{aligned} &-\omega^2 M_1 X_1 + K_1 X_1 + j\omega B_1 X_1 + K_2 (X_1 - X_2) + j\omega B_2 (X_1 - X_2) = R(j\omega) \\ &-\omega^2 M_2 X_2 + K_2 (X_2 - X_1) + j\omega B_2 (X_2 - X_1) = 0 \end{aligned} \tag{3}$$

The solution to (3) is trivial using, for example, the Cramer's rule. In particular, we may find the complex frequency response function (receptance), relating the dynamic response of the compressor and the above stimulus applied to the compressor housing, this is:

$$\begin{aligned} &H_{11}(j\omega) = \frac{X_1(j\omega)}{R(j\omega)} = \\ &= \frac{\left(-M_2\omega^2 + K_2 + j\omega B_2\right)}{\left[-M_1\omega^2 + K_1 + K_2 + j\omega(B_1 + B_2)\right]\left[-M_2\omega^2 + K_2 + j\omega(B_1 + B_2)\right] - \left(K_2 + j\omega B_2\right)^2} \end{aligned} \tag{4}$$

Using (2) and (4), we find the absolute transmissibility − complex frequency response function relating the force transmitted to the base and the above stimulus applied to the compressor housing, this is:

$$T(j\omega)=\frac{F(j\omega)}{R(j\omega)}=\frac{X_1(j\omega)(K_1+j\omega B_1)}{R(j\omega)}=$$
$$=\frac{(K_1+j\omega B_1)(-M_2\omega^2+K_2+j\omega B_2)}{\left[-M_1\omega^2+K_1+K_2+j\omega(B_1+B_2)\right]\left[-M_2\omega^2+K_2+j\omega(B_1+B_2)\right]-(K_2+j\omega B_2)^2} \tag{5}$$

In (4) and (5), setting $\omega_{dr}=\sqrt{K_2/M_2}$ and $B_2\to 0$, minimizes the magnitude of the above frequency response functions at a given driving frequency ω_{dr}, i.e., the fundamental component of the force transmitted to the base may be nullified independently of other system parameters, e.g. $T(j\omega_{dr}),\ H_{11}(j\omega_{dr})\to 0$.

The above explains the operational principle of the tuned dynamic absorber, which needs to be "tuned" such as to have a resonant frequency exactly equal to the driving frequency.

It will be very instructive now to study the influence of the properties of vibration mount on the attainable performance. This study will be based on the typical numerical example, where $M_1=0.33kg,\ M_2=0.1kg$, representing the actual values of the technology demonstrator, as explained below. The dynamic absorber will be tuned to the driving frequency, 75Hz, therefore $K_2=M_2\omega^2=0.1\times(2\pi\times 75)^2=22207\ (N/m)$. The damping ratios for this example will be $\zeta_1=5\%,\ \zeta_2=0.2\%$ for the vibration mount and dynamic absorber, respectively. These values were evaluated experimentally (see below) for the all-metal flexural links.

Figure 2 shows the dependencies of a module of absolute transmissibility $|T(j\omega)|$ on the frequency for different resonant frequencies of vibration mount $\frac{1}{2\pi}\sqrt{\frac{K_1}{M_1}}$ ranging from 20 to 100Hz.

In Figure 2a, at the chosen (driving) frequency of 75 Hz all the graphs show favorable deep antiresonant notches, the widths and depths of which strongly depend on the resonant frequency of the vibration mount. Namely, the softer the vibration mount is, the better performance may be achieved. Figure 2b shows the dependence of the suppression ratio at the driving frequency on the resonant frequency of the vibration mount, indicating that very impressive suppression ratios are achievable if a low frequency vibration mount is chosen.

The rms magnitude of vibration export produced by the compressor and expander may be calculated using the expression $\Phi=M\Delta\omega_{dr}^2/\sqrt{2}$, where M and Δ are the mass of the moving component and half stroke, respectively. Calculations show that maximum magnitudes of vibration export produced by the compressor and expander might be as high as 23 N rms and 3 N rms.

From the above, a suppression ratio of 115 will be needed to achieve the typical requirement of compressor vibration export 0.2 N rms; therefore, the resonant frequency of vibration mount 60Hz will be adequate. This case is represented by the 60-Hz curve in Figure 2a and is marked in Figure 2b. Using such a vibration mount will reduce the vibration export from the expander to 0.026 N rms.

It is important to note that the axial deflection of the vibration mounted expander under such conditions will be also quite small.

From calculation using Eq. (4), the maximum possible magnitude is 0.4µm rms, which is quite acceptable for the most demanding electro-optic instrumentation normally having essentially larger focus depth. It is also important to note the presence of the high frequency resonance, which is known to be one of the drawbacks of tuned dynamic absorbers normally limiting their use in appli-

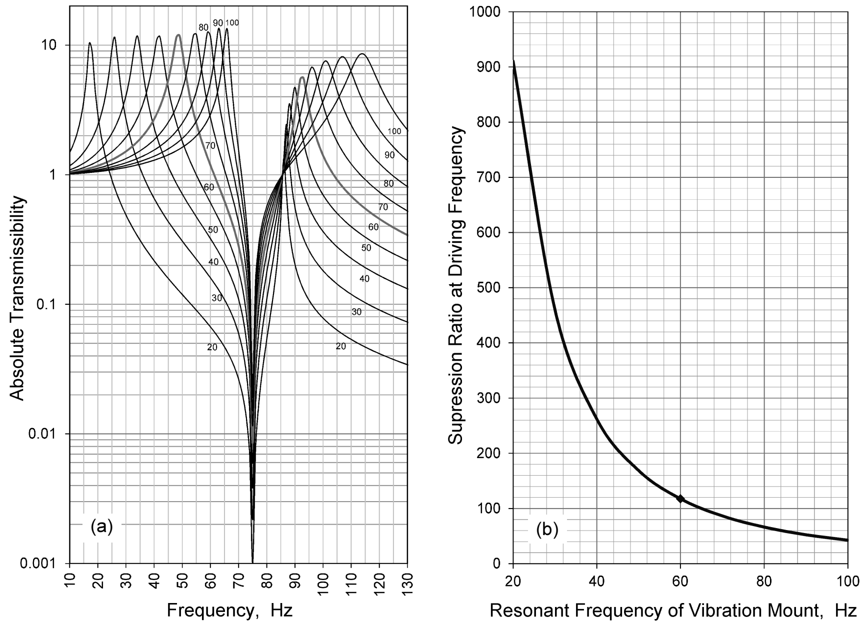

Figure 2. Frequency response functions (a) and suppression ratio (b) at different resonant frequencies of vibration mount.

cations exposed to external vibration and shock. This issue, however, is not relevant for space applications, which are literally free of g-forces, gravity, shocks and vibration. Nevertheless, special attention needs to be paid to the protection of such an arrangement during the mission launch stage. This may be achieved by using viscoelastic snubbers or controllable locks. The proper solution depends mostly on system design; this discussion is beyond the scope of this paper.

VIBRATION PROTECTIVE ARRANGEMENT AND ATTAINABLE PERFORMANCE

Figure 3 shows the full scale technology demonstrator. In Figure 3a, the thermo-mechanical unit is comprised of a single-piston compressor ① and an expander unit ②, which are interconnected using a compliant transfer line ③. Both the compressor and expander units are clamped inside the holders ④, ⑤ which are, in turn, supported from the base ⑥ using four flat flexurals ⑦ fabricated by photo etching of fully hardened stainless spring steel. The thermal straps ⑧ are made of multi-strand copper wire and connect thermally the compressor and expander holders to the base. Two identical tuned dynamic absorbers ⑨ are in-line mounted upon the compressor and expander units.

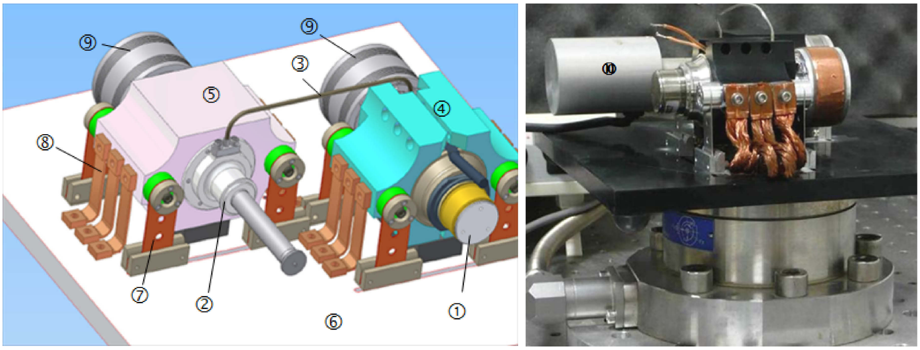

Figure 3. Vibration protective arrangement.

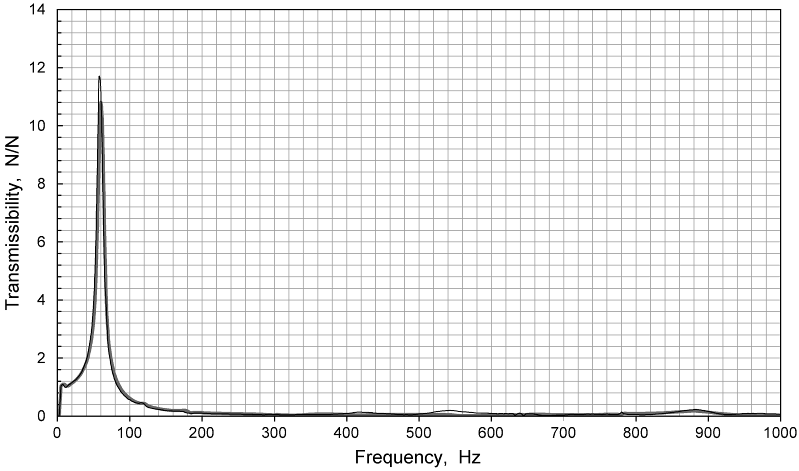

Figure 4. Frequency response functions of vibration mounts.

The above shown design provides for a true SDOF vibration isolation of compressor and expander units having resonant frequencies in all directions (with the exception of axial) in excess of 300 Hz, as needed for the system integrity and mechanical stability of the cold finger. Figure 3b shows the picture of the above described technology demonstrator, where the cold finger is placed inside the evacuated envelope of the simulation dewar ⑩; the temperature diode and heat load resistor are mounted upon the cold fingertip allowing for heat load mimicking, temperature monitoring and, therefore, closed loop temperature control operation. The cooler induced vibration export characterization has been performed using the experimental setup shown in Figure 3b, where the above technology demonstrator is mounted on a four-component Kistler Type 9272 dynamometer, being clamped upon a heavy vibration-isolated Newport table to provide isolation from ground induced vibration. From testing, both vibration isolators were characterized by the resonant frequency 60 Hz and damping ratio of about 5%, this assessment relies on the frequency response functions shown in Figure 4. These values were adequate for testing purposes. It is important to note that for environmental conditions typical of an aerospace vehicle (zero gravity, g-forces and vibration), even a softer vibration isolator is applicable. In a practical implementation, the tuned dynamic absorber comprises a heavy inertial ring, made of tungsten for compactness, which is clamped between two sets of flat flexural bearings (fabricated by photo-etching from fully hardened SST304 0.5mm sheet), separated by spacers to eliminate fretting and damping effects associated with dry friction between adjacent springs, as shown in Figure 5. From practice, such a design allows for obtaining the desired very low damping ratio, typically 0.2%, as needed for sufficient vibration suppression [14] and a typically small deviation ±0.3Hz of the resonant frequency from the nominal value.

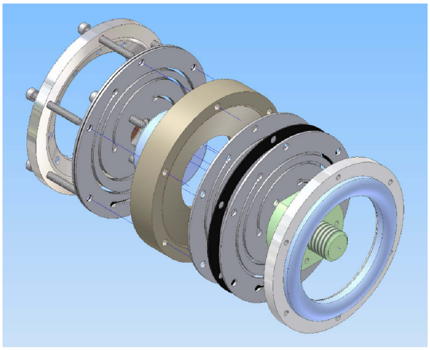

Figure 5. Tuned dynamic absorber - exploded view.

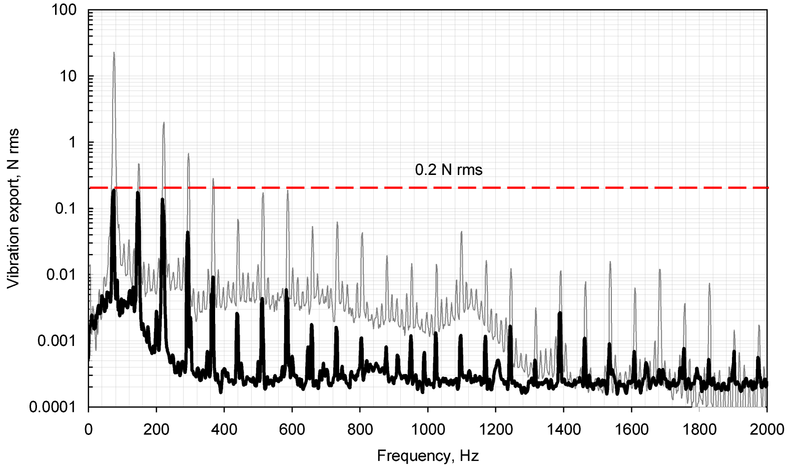

Figure 6a. Measured vibration export in on-axis direction (black); gray is with rigid mount.

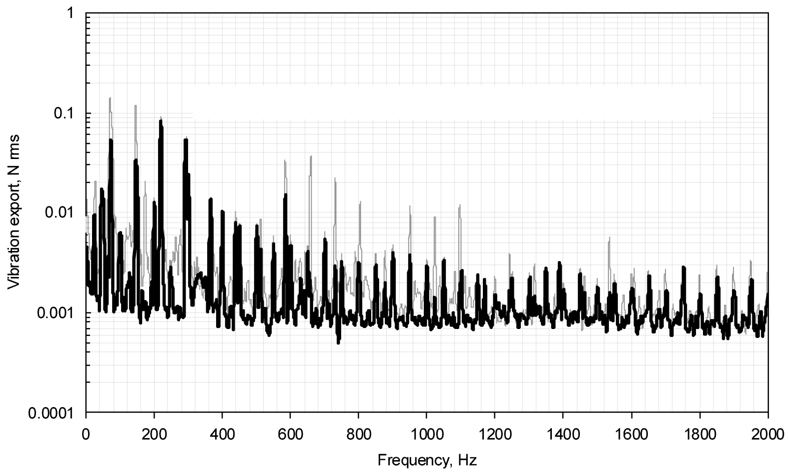

Figure 6b. Measured vibration export in horizontal off-axis direction (black); gray is with rigid mount.

Figures 6a, 6b and 6c show the typical spectra of vibration export produced by the entire cooler (compressor and cold head) running in a temperature control mode 500 mW@80 K@23°C (solid black curves), where, for reference, the spectra representing the case of rigid mounting (light gray curves) are superimposed.

To start with, in Figure 6a, the reference vibration export produced by the cooler in the on-axis direction comprises a dominant fundamental component approaching 23 N rms along with multiple higher order harmonics, four of which are higher than the typical requirement 0.2 N rms represented by the dashed line. As to the off-axis vibration export in the horizontal (b) and vertical (c) directions, even in the reference case of rigid mounting, the spectral components of vibration export are significantly lower than the typical requirement. The relatively low level of higher order harmonics has been achieved by the above described mechanical design ensuring the smooth mechanical motion of lightened mechanical components.

The direct benefit of this is that only suppression of on-axis vibration export is needed, and this may be achieved by combining a SDOF vibration isolator and a tuned dynamic absorber. In particular, in Figure 6a, the vibration export at the driving frequency is approximately 115-fold attenuated down below the required 0.2 N rms, and the higher order components are also attenuated well below this requirement.

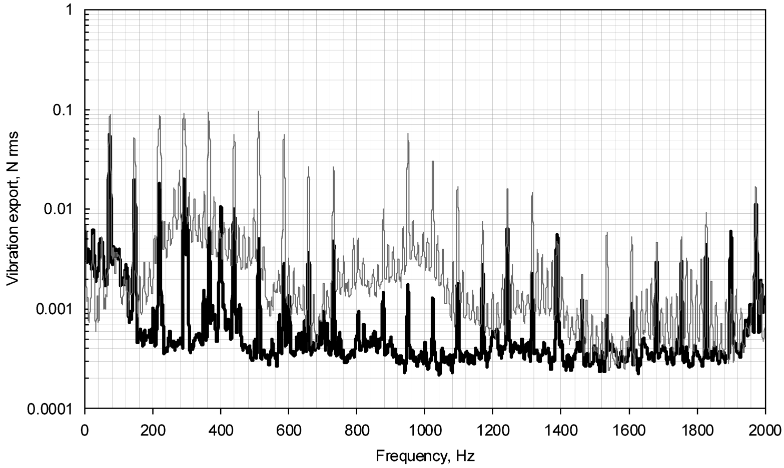

Figure 6c. Measured vibration export in vertical off-axis direction (black); gray is with rigid mount.

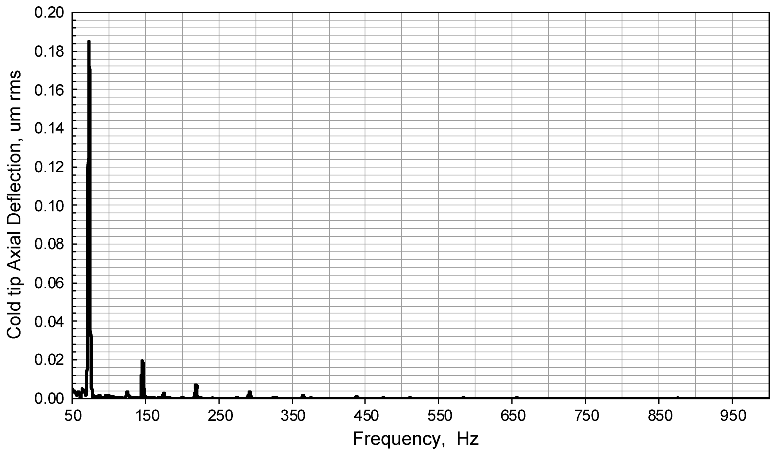

Figure 7. Axial deflection of the cold fingertip.

It was also quite important to evaluate the dynamic responses of the vibration mounted expander unit. The reason was the danger of an unacceptable line of site jitter resulting from excessive cold finger tip motion in cases where the focal plane array is mounted directly, or when transmission of large dynamic forces occurs through insufficiently-compliant conductive braids in the case when the Focal Plane Array is mounted upon a separate pedestal. From experiment, the highest spectral component is 0.18 μm rms and corresponds to the driving frequency of 75 Hz (see Figure 7); the off-axis and angular motion was practically undetectable. In light of the large depth of focus, such a minor cold fingertip motion is negligible. A similar situation was observed for the compressor unit, meaning that transfer line reliability is not compromised at all.

CONCLUSIONS

The demonstrated approach to attenuation of the vibration export produced by the components of a tactical split Stirling cryogenic cooler is quite adequate for many aerospace applications, especially for budget-constrained missions relying on mini and micro satellites. The favorable combination of a tuned dynamic absorber and low frequency vibration isolation seems to be ideally suited for relatively mild aerospace environmental conditions characterized by low reject temperatures and which are practically free of gravity, shocks and vibration.

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