

# Design of Aural Undetectable Cryogenically Cooled Infrared Imagers

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## ABSTRACT

For the sake of weight and compactness, the typical enclosure of a portable cryogenically cooled infrared imager is made in the form of a light metal thin-walled shell, serving as an optical bench, accommodating a telescope, an optical train and an Infrared Detector Dewar Cooler Assembly (IDDCA). Such IDDCAs normally rely on a miniature rotary Stirling cryogenic cooler, which is known as a powerful source of wideband vibration export that excites the inherently lightly damped structural resonances in the imager enclosure resulting in unacceptable structure-borne vibration, microphonics, and acoustic noise.

Resulting from this are spoiled imagery and increased range for aural detectability of forward observers who must remain undetected, potentially for long periods of time. As a matter of fact, aural nondetectability distance becomes one of the crucial figures of merit (along with overall weight, battery life, imagery quality, etc.) characterizing the performance of a modern portable infrared imager.

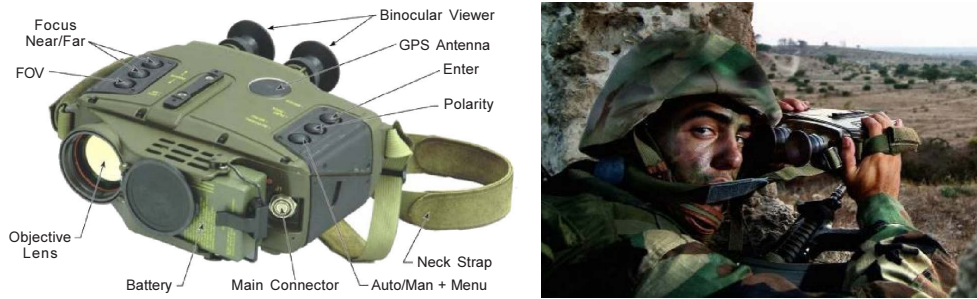
In a novel approach, we have mounted the IDDCA within the enclosure using a special silencing pad, the mechanical properties of which are chosen so as to not produce excessive line of sight jitter and effectively attenuate vibration transmission over the typical high frequency range containing the relevant structural resonances of the enclosure.

The residual noise radiation from the imager enclosure is then attenuated practically to a background level by reshaping the radiation modes with a purpose of diminishing the overall volume velocity. This is achieved by finding the “critical point” and affixing there an optimally sized correction mass.

The authors report on a successful attempt to develop a cooled imager that is inaudible at greater than 10 meters (even during the cool down mode) per MIL-STD-1774D (Level II).

## INTRODUCTION

Novel tactics for carrying out military and antiterrorist operations call for the development and deployment of a new generation of warfare technology. In particular, portable infrared (IR) imagers (cooled or uncooled) converting the thermal battlefield into a dynamic visual imagery enhance the day/night situational awareness and targeting ability of the leaders of combat infantry and Special Forces. Advances in uncooled thermal imaging technology offer aural stealth along with low power consumption; however, such imagers normally have a limited operational range and are not really adequate for tracking and observing fast moving targets.



**Figure 1.** 3-5 $\mu$ m FPA Long Range Thermal Imager "LRTI" (courtesy Elbit Systems of America)

The superior performance of a cooled IR imager is achieved by reducing the temperature of the focal plane array (FPA) down to cryogenic temperatures typically using a closed-cycle mechanical (Stirling) cryogenic cooler. It is widely accepted that the "best technology for the true IR heat detection is the cooled detector" [1].

Figure 1 shows the typical cryogenically cooled hand-held, 3-5 $\mu$ m FPA Long Range Thermal Imager "LRTI" (Elbit Systems of America). Such an imager comprises an optical train and Integrated Dewar Detector Cooler Assembly (IDDC) relying on the Ricor Type K560 cryogenic cooler, which is, for the sake of weight and compactness, mounted directly upon the thin-walled imager enclosure made of aluminium.

Miniature, rotary driven close cycle Stirling cryogenic coolers, which find wide use in such imagers, comprise two major reciprocating components: a compression piston and displacer-regenerator [2]. Their 90° out of phase reciprocation, as required for producing the useful cooling effect, relies on two crank-slider arrangements actuated by the rotary DC brushless drive. The reciprocating motion of the compression piston provides the required pressure pulses and the volumetric reciprocal change of a working agent (pure helium, typically) in the expansion space of an expander. The displacer-regenerator, reciprocating inside a cold finger, shuttles the working agent back and forth from the cold side to the warm side of the cooler. During the expansion stage of the thermodynamic cycle, heat is absorbed from the cold finger tip, to which the IR FPA is thermally and mechanically interfaced, and during the compression stage heat is rejected to the environment from the hot cold finger base.

Figure 2 shows the typical Ricor integral rotary driven microcryogenic cooler model K561, which was developed primarily for use in the portable IR imagers. In spite of apparent simplicity, this is a sophisticated piece of machinery, comprising numerous movable components, the mechanical motion of which produces periodic microcollisions resulting in a radiation of wideband vibration and noise. In particular, it is important to mention periodic metal-to-metal slap occurring between the piston and cylinder occurring twice per crankshaft revolution and producing wideband vibration export comprising numerous harmonics of the driving frequency [3].

For the sake of weight and compactness, such a cooler is normally rigidly mounted upon the light-metal thin-walled imager enclosure having finite stiffness and showing numerous undamped resonances over the entire frequency range. The above vibration export is, therefore, easily transformed into a resonant structural vibration resulting in a loud acoustic noise, the spectrum of which comprises specific resonant peaks correlating closely with the frequencies of the said structural resonances. From experience, even silent cryogenic coolers may produce unacceptable microphonics and aural structure-born noise when mounted upon a wrongly designed and resonating enclosure of an IR imager.

The "noisy" thermal imager may be remotely detected using inexpensive listening sniper detection device relying upon a high-sensitivity unidirectional microphone or audibly spotted when used in a close proximity to the opponent force, thus compromising the personnel's safety and mission's success. Resulting from this is that aural stealth along with imagery quality, compactness, low power consumption and long lifespan becomes a crucial figure of merit characterizing the modern IR imager [1]. Development of acoustically quiet portable IR imagers makes a wider class of missions possible.



**Figure 2.** Integral rotary microcryogenic cooler Ricor model K561

Suppressing the structural resonances is feasible (to a particular extent only) by using heavy and stiff optical benches or enclosures showing no resonant phenomena over the range of audible frequencies; this approach can not be considered when the weight and size are of concern. A resonant behavior of the imager enclosure may be also controlled by applying free or constrained layer damping treatment, where damping occurs as a result of the cyclic tensile or shear deformation of the damping layer [4-6]. The performance of these approaches, though, is not sufficient since the above deformations are relatively small and not all the types of structural vibration necessarily cause the desired sort of deformation in the damping layer. Moreover, this approach is not always practical because of the complicated geometry, the added weight, outgassing and compromised heat management.

Essential suppression of the structural resonances is also possible using the principle of a wideband dynamic absorber [4-7]. In this approach, auxiliary optimally sized mass/masses are affixed through optimally compliant and damped mounts in the appropriate locations upon the imager enclosure. This, however, requires intolerable increase of the mass and is not always viable within a densely packaged imager.

The authors [4] attempted to enable effective use of an imager enclosure as a vibration isolated and damped acoustic envelope. This objective has been achieved by making use of the vibration mounting of a cryogenic cooler with the purpose of wideband dynamic damping of the enclosure structural resonances over the typical mid-frequency range and vibration isolation over the typical high-frequency range. The authors developed and fabricated the silencing pad, where the top and bottom interface plates are bonded using the silicon molded layer essentially avoiding metal-to-metal contact. Special technologies were developed for the final accurate machining of the pad, thus assuring precision mounting of the IDDCA package upon the imager enclosure and reliable hysteresis-free alignment of the optical axis. The vibration protective pad was optimally stiffened and damped for achieving the best acoustical performance without developing excessive cooler-induced line of sight jitter. The authors of [8] reported on the successful effort of designing the inaudible at greater than 10 meters cryogenically cooled infrared imager complying with the most stringent requirements for 10-meter aural nondetectability MIL-STD-1774D (Level II) [9] when operating in the steady-state mode.

In this paper, the authors disclose the outcomes of their effort towards further improving the aural stealth performance delivered by the portable infrared imager in the open-loop mode thus meeting the most stringent MIL-STD-1774D (Level II) 10 meters aural nondetectability requirement. For this purpose, the authors make a practical use of the so-called “weak radiators” concept [10]. In this approach, the essential suppression of the noise radiation by the vibrating planar structure may be achieved not only by suppressing the vibration magnitudes, but by an appropriate redistribution of

the radiating surface velocities or reshaping the “radiation modes”. Ideally, cancelling the volume velocity effectively eliminates the noise radiation – the surface becomes a weak radiator. In theory, this redistribution may be achieved by optimally placing a plurality of the optimally sized correction masses.

It is important to note that the suppression of wideband noise produced by the structure, showing numerous lightly damped structural resonances over the wide frequency range, using such an approach is not really practical. The numerous radiation modes need to be treated simultaneously by optimally placing a large number of said correction masses; this task not always can be accomplished because of the weight limitations and since not all the optimal locations are available for correction masses placement. In our particular case of the vibration mounted cooler, most of the high frequency content has already been attenuated by the vibration protective pad. There normally remain one or two low frequency active radiation modes that need to be treated, and this may be conveniently achieved using a single correction mass.

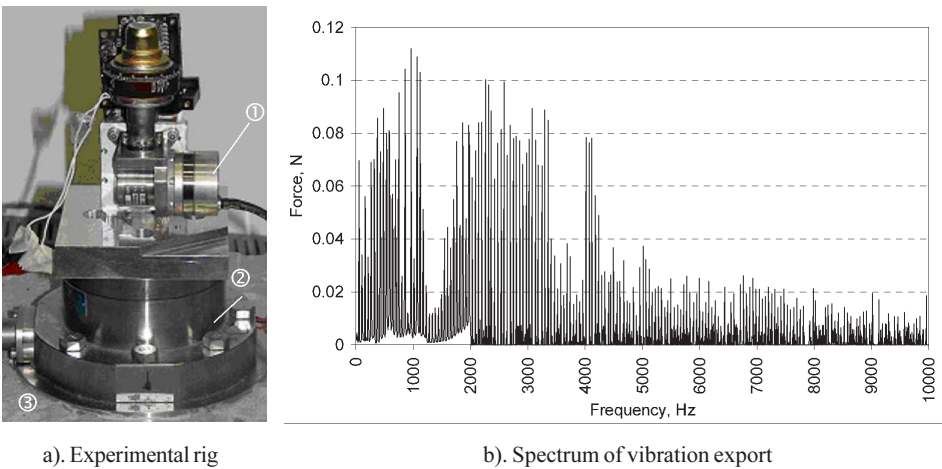
**VIBRATION EXPORT BY CRYOGENIC COOLER**

Figure 3a shows the test rig, where the IDDCA relying on the Ricor model K561 integral cryogenic cooler ① is mounted upon the 3-axis Kistler Type 9272A piezoelectric dynamometer ② performing simultaneous measurement of the three mutually perpendicular components of the interface force, further – vibration export. The dynamometer is mounted upon the heavy and pneumatically vibration isolated concrete block ③. The measuring set-up consists of the Bruel & Kjaer Type 2635 Charge Amplifier and Data Physics ACE Signal Analyzer (not shown).

The graph in Figure 3b shows the typical spectrum of vibration export produced by the above IDDCA in the direction normal to its mounting surface. From Fig. 3b, the internal vibroimpact process, which is inherent in such coolers, produces an essentially wideband multi-tonal spectrum comprised of the numerous harmonics of the driving frequency over the wide frequency range up to 10 kHz.

**ANALYTICAL MODEL OF THE CRYOGENIC ENGINE VIBRATION MOUNTED UPON FLEXIBLE ENCLOSURE**

To facilitate the use of the system enclosure as a vibration isolated and damped acoustic envelope, we use a compliant vibration mount—silencing pad—for isolating the source of the vibration export and for dynamically damping the structural resonances utilizing the inertia of the relatively heavy cryogenic engine. Figure 4 shows the simplified dynamic model of the imager enclosure with the vibration mounted cryogenic engine [8]. The dynamic properties of the above enclosure with



**Figure 3.** Measurement of vibration export

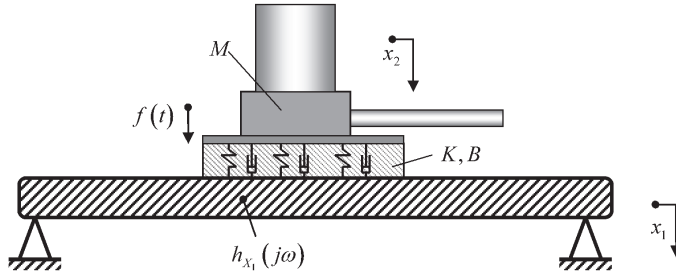


Figure 4. Dynamic model

dismantled cryocooler are given through the complex frequency response function (point receptance),  $h_{x_1}(j\omega)$ , where  $\omega$  is the circular frequency and  $j = \sqrt{-1}$ . This property may be directly measured using a standard hammer test or evaluated using FEA modeling. Further, the cooler mass is  $M$ ; visco-elastic properties of the vibration mount are  $B$  and  $K$ , respectively. In the model below, the vibration export (excitation force),  $f(t)$ , is applied to the cooler body. The absolute deflection of the enclosure and the cooler from the position of static equilibrium are  $x_1(t)$  and  $x_2(t)$ , respectively.

### OPTIMAL DESIGN OF THE VIBRATION MOUNT

It is important to notice that the driving frequency may vary in accordance with the heat loading applied to the cooler and the frequency content of the above vibration export may differ from cooler to cooler. Hence, ensuring the most efficient vibration and noise control requires minimizing the heights of the resonant peaks in the magnitude spectrum of the enclosure mobility given by (3) by optimally choosing the visco-elastic properties  $B$  and  $K$  of the vibration mount.

Using Fourier formalism

$$x_1(t) \leftrightarrow X_1(j\omega), \quad x_2(t) \leftrightarrow X_2(j\omega) \quad \text{and} \quad f(t) \leftrightarrow F(j\omega) \quad (1)$$

the equations of motion in the frequency domain for the system in Figure 4 may be considered in the following form [8]:

$$\begin{aligned} X_1(j\omega) &= h_{x_1}(j\omega)(K + j\omega B)[X_2(j\omega) - X_1(j\omega)], \\ -M\omega^2 X_2(j\omega) + (K + j\omega B)[X_2(j\omega) - X_1(j\omega)] &= F(j\omega). \end{aligned} \quad (2)$$

Solving (2) for  $X_1(j\omega)$ , we obtain the point receptance and mobility of the enclosure with vibration mounted cooler:

$$\begin{aligned} H_{x_1}(j\omega) &= \frac{X_1(j\omega)}{F(j\omega)} = \frac{h_{x_1}(j\omega)(K + j\omega B)}{[1 + h_{x_1}(j\omega)(K + j\omega B)][K - M\omega^2 + j\omega B] - h_{x_1}(j\omega)(K + j\omega B)^2}, \\ H_{v_1}(j\omega) &= \frac{j\omega X_1(j\omega)}{F(j\omega)} = \frac{j\omega h_{x_1}(j\omega)(K + j\omega B)}{[1 + h_{x_1}(j\omega)(K + j\omega B)][K - M\omega^2 + j\omega B] - h_{x_1}(j\omega)(K + j\omega B)^2}. \end{aligned} \quad (3)$$

Using (3), we can calculate the complex velocity spectrum of the enclosure with vibration mounted cooler, this is

$$V_1(j\omega) = \frac{j\omega h_{x_1}(j\omega)(K + j\omega B)F(j\omega)}{[1 + h_{x_1}(j\omega)(K + j\omega B)][K - M\omega^2 + j\omega B] - h_{x_1}(j\omega)(K + j\omega B)^2}. \quad (4)$$

Obviously, by choosing the low frequency and undamped vibration mount (small  $B$  and  $K$ ) we can effectively isolate the vibration transmission from the cooler to the imager enclosure and, thus, to solve the problem of structure-borne noise generation. However, excessively soft and undamped vibration isolation may lead to an intolerable cooler-induced line of sight jitter, settling times and deflections due to the g-forces along with environmental vibration and shocks. On the other hand, over-stiffening or over-damping the vibration mount will eventually diminish the effect of vibration

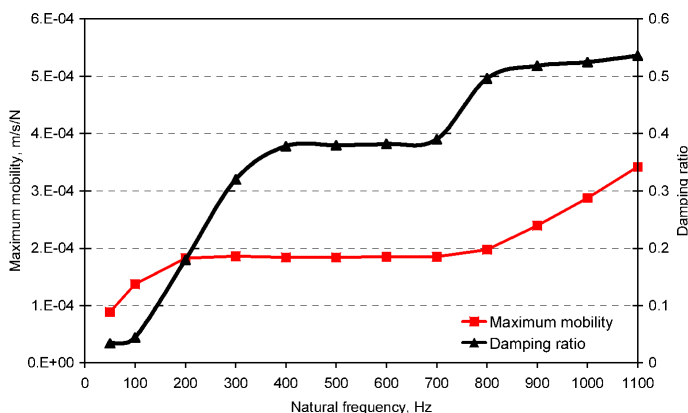


Figure 5. Optimized properties of vibration protective pad

isolation. Obviously, there exists some optimal combination of the pad stiffness and damping providing for the best vibration control and sufficient optical stability.

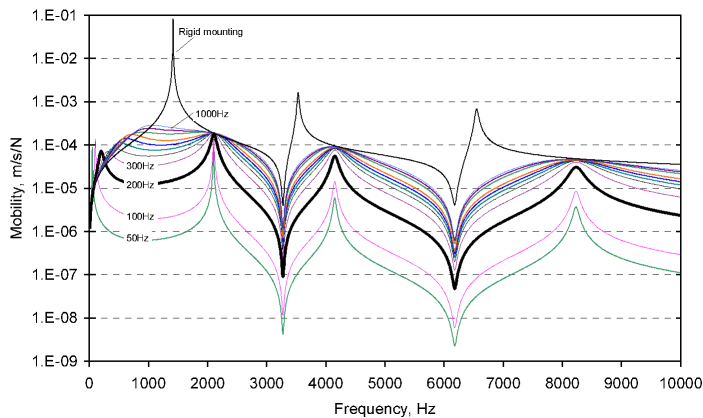
To estimate the range of optimal parameters, for each value of the natural frequency of the mounted cooler  $1/2\pi\sqrt{K/M}$  in the range 50 – 1100 Hz, using the numeric procedure relying on (3), we found the optimal damping ratio  $B/2\sqrt{MK}$ , that minimizes the height of the resonant peaks in the spectrum of mobility over the entire frequency range. In the course of these calculations the experimentally obtained complex frequency response function  $h_{x_i}(j\omega)$  has been used.

Figure 5 shows the dependency of the optimal damping ratio and the mobility maximum on the value of the mounted cooler's natural frequency. From Figure 5, a low frequency vibration mount needs to be lightly damped, thus providing for the best control over the resonant phenomena in the enclosure simply by isolating the "receiver" from the source of vibration export. Increasing the natural frequency of the above mount leads to deterioration of the attainable performance; such behavior is typical for the regular vibration isolator. Further, increasing the resonant frequency of the vibration mount from approximately 200 Hz up to 700 Hz, we observe an odd "saturation" in the performance curve. It is important to notice that in the range of 400-700 Hz the optimal amount of damping ratio remains practically constant, namely 40%. In this frequency range, the visco-elastically supported lumped mass of the cryocooler behaves similar to a wideband dynamic absorber towards the originally undamped enclosure.

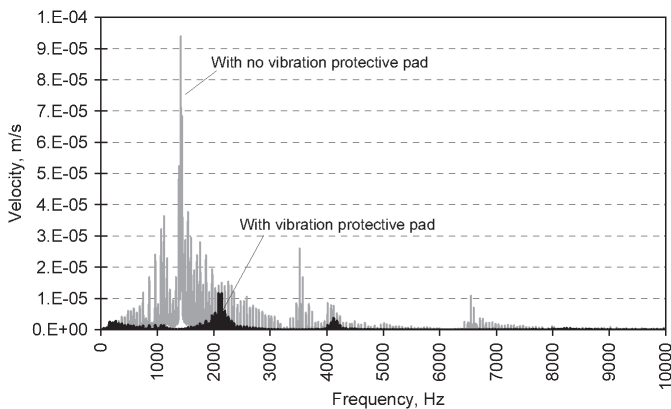
And finally, by increasing the resonant frequency of the vibration isolator in excess of 700 Hz we observe sharp deterioration of the attainable performance. Such behavior is again typical for regular vibration isolator. From experience, a vibration mount for the cryocooler that provides a resonant frequency of 200 Hz and a damping ratio of 20% is quite adequate for maintaining the cooler-induced line of sight jitter and settling times within specified limits. Further, it is important to notice that such a damping ratio is quite achievable without making use of exotic technologies.

Using (3), we calculated the family of optimized mobility spectra for the enclosure with a compliantly-mounted cryogenic engine at different natural frequencies. This is portrayed in Figure 6, where in the vicinity of the first resonance, the shape of the mobility curve is typical of a dynamically damped system, while in the high frequency range, the effect of vibration isolation is more dominant. In Figure 6, the optimized mobility curve, which corresponds to a vibration mount with a resonant frequency of 200 Hz and a damping ratio of 20% is plotted using the bold line. Figure 7 compares the theoretically calculated velocity spectra both with (black curve) and without (grey curve) the vibration protective pad. From Figure 7, it is seen that the enclosure vibration is essentially attenuated over the entire frequency range. The residual vibration is mainly observed at 2000 Hz. For this calculation, we used expression (4) for the case of the rigid cooler mounting and for the case of the above optimized compliant cooler mounting; the experimental spectrum shown in Figure 3b was used for the excitation force  $F(j\omega)$ .





**Figure 6.** Optimized mobilities of the imager enclosure with vibration mounted IDDCA



**Figure 7.** Comparison of the surface velocities

#### EXPERIMENTAL AND AURAL NONDETECTABILITY TESTING PER MIL-STD 1474D

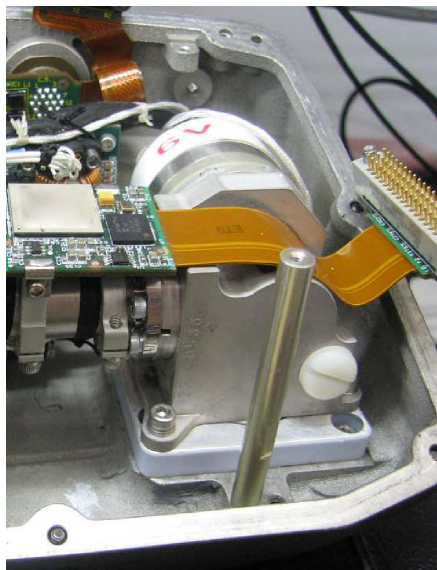
Based on the preceding analysis and experience, a vibration protective pad for the Ricor model K560 integral cryogenic cooler was manufactured. Figure 8 portrays the vibration protective pad comprising the bottom plate for mounting upon the enclosure and top plate for mounting the cooler housing. The plates are bonded one to another using a military-grade clear RTV. By varying the pad geometry and RTV grade, we obtained a pad having a natural frequency and damping ratio of approximately 200 Hz and 20%, respectively [8].

From experimentation, the above pad behaves as a linear vibration mount and survives harsh vibrational conditions like random vibration of magnitude 10 g rms (uniform PSD over the range 10-500 Hz) and half-sine 40 g at 18 ms shock per MIL-STD 810F [11]. Varying the ambient temperature over the range -20°C to 120°C appears to have no visible influence upon the pad dynamic properties. Special technologies were developed for the final accurate machining of the pad, thus assuring precision mounting of the IDDCA package upon the imager enclosure and reliable alignment of the optical axis.

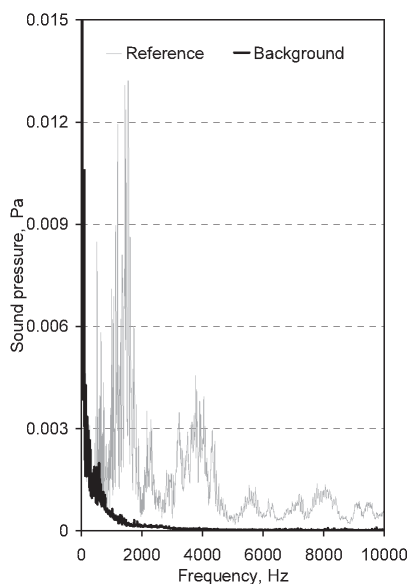
Figure 9 shows the IDDCA relying upon the Ricor model K561 cryogenic cooler mounted upon the bottom cover of the typical infrared imager using the above mentioned vibration protective pad. During the design stage, the imager enclosure was acoustically optimized. This involved adding extra mass to the enclosure in the region of the cooler mounting, using low-profile stiffening ribs and smoothing the internal and external shapes, as portrayed in Figure 9. Before proceeding to the acoustical testing, the optical testing has proved that cooler induced line of sight jitter is within the tolerance.



**Figure 8.** Vibration protective pad.



**Figure 9.** Portable IR imager with vibration mounted IDDCA.



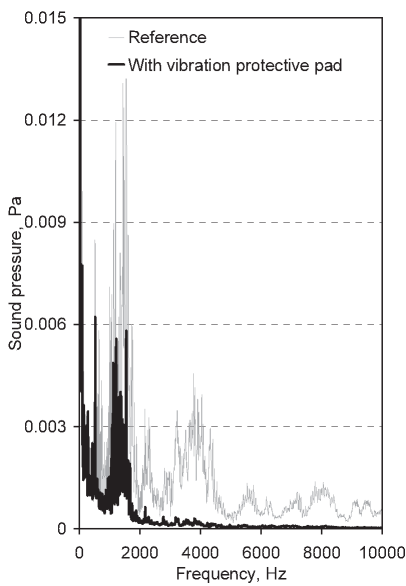
**Figure 10.** Spectrum of sound pressure - reference.

In Figure 10, the black bold curve shows the spectrum of the background sound pressure (SP) and the reference light gray curve shows the spectrum of the near field (measuring distance 5cm) sound pressure produced by the cryogenic cooler when driven in the cool-down mode and mounted inside the IR imager with no vibration protective pad. In Figure 10, the well pronounced peaks at 500, 1200, 1500, 2200, 3200, 3800 Hz correspond to the structural resonances.

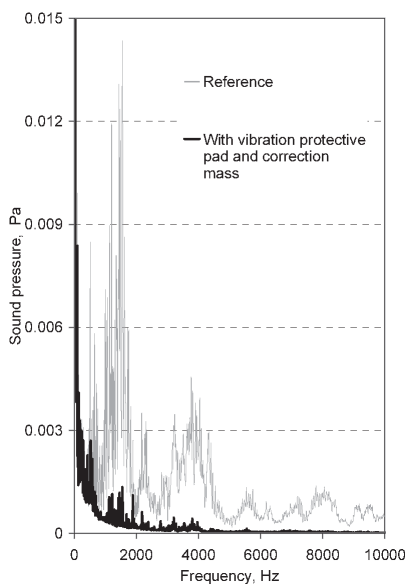
Using only the above optimized vibration protective pad yields essential vibration and noise attenuation. Figure 11 compares the SP spectra produced by the imager in cool-down mode with no vibration protective pad (reference) and the imager with vibration protective pad. From Figure 11, essential noise attenuation was attained over the entire frequency range, where starting from 1800 Hz the noise was practically attenuated down to the background level. However, the residual noise radiation at 500 Hz and 1000-1700 Hz frequency band remained excessively high to meet the 10 meter requirement and showed the potential of compromising the imager acoustic stealth during the cool down phase of operation.

Next, the authors have made a practical use of the concept of “weak radiators” [10]. As was explained in the Introduction, an appropriate redistribution of the radiating surface velocities or reshaping the remaining “radiation modes” towards cancelling the overall volume velocity was conveniently achieved by optimally placing a single optimally sized point mass. This became pos-





**Figure 11.** Spectrum of sound pressure – performance of vibration protective pad.



**Figure 12.** Spectrum of sound pressure – performance of vibration isolation and mode reshaping

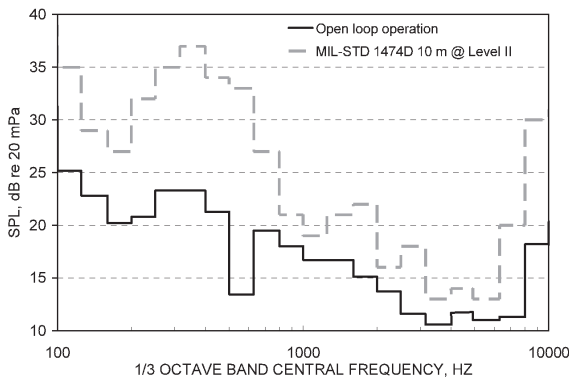
sible primarily because the silent pad has already attenuated most of the high frequency vibration content, and because there remained only two essentially unsuppressed active radiation modes for reshaping.

From experiment, we found the optimal placement and optimal weight (26 gr) for the correction mass. Figure 12 shows the outcome of this effort — the noise produced by the imager during the open-loop cool-down phase was practically attenuated down to the background level over the entire frequency range (compare this with Figure 10).

The purpose of further testing was to ensure compliance of the imager operating in the cool-down mode with 10 meters aural nondetectability (Level II) MIL-STD 1474D requirements [9]. The testing was carried out in a hemi-anechoic chamber, as portrayed in Figure 13, where the IR imager is mounted 1.2 meters above the floor using a tripod, and a microphone is distanced 2 meters from the imager. Self-explanatory Figure 14 compares the above MIL-STD requirement (grey dashed curve) with the outcome of the actual testing (bold black curve).



**Figure 13.** Aural nondetectability testing in hemi-anechoic chamber.



**Figure 14.** Actual sound pressure level measurement vs MIL-STD 1474D requirement.

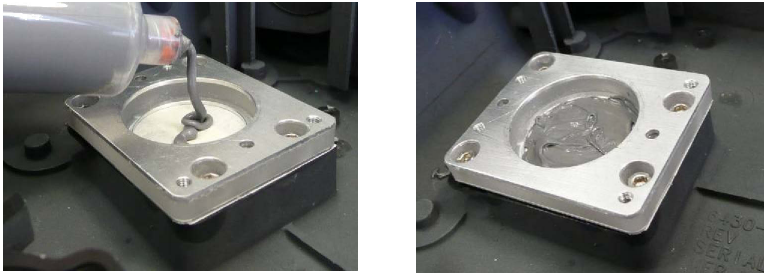


Figure 15. Application the heat conductive  $\lambda$ -gel

## HEAT MANAGEMENT

Since the cryogenic cooler operates in a closed space with no good thermal contact with the imager enclosure and with no means for the heat sinking through forced convection, there appears a problem of rejecting the heat radiated by the cooler (e.g. 5 W at 23°C ambient) to the environment. As is known, the vibration isolator relying on silicon rubber is a poor heat conductor. Increasing its heat conductivity by adding heat conductive filler may compromise the adhesion performance and tensile strength. The problem was successfully solved [8] by applying a very soft paste-type DP series silicone  $\lambda$ -gel, made by Geltec (see <http://www.geltec.co.jp>) that shows outstanding thermal conductivity up to 6.5 W/mK, low outgasing, and persistence in mechanical and thermal properties over a wide range of temperatures. The above DP-100  $\lambda$ -gel was applied in such a manner as to fill the gap of approximately 1 mm between the cooler housing and the pad's bottom plate, as portrayed in Figure 15. In doing so, the entire amount of gel was effectively encapsulated inside the pad's cavity, thus preventing it from getting into the sensitive areas of the imager interior and staining the optics. Experimentation has shown that the temperature drop between the cooler and enclosure skin under standard ambient conditions (23°C, 8 hours of steady-state operation) is less than 1 degree, which is considered as highly satisfactory.

## CONCLUSIONS

The required aural stealth of a sophisticated, cryogenically cooled portable IR imager has been achieved by mounting the cryogenic engine within the imager enclosure using a vibration protective pad and redistributing the radiation modes using a single, optimally-sized and placed corrective mass. The 10-meter aural non-detectability distance per MIL-STD 1474D (Level II) in the cool-down mode has been achieved without increasing the mass and dimensions of the IR imager at the same power consumed by the cooler and the same temperature of the cryogenic engine skin.

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