Jitter Suppression Techniques for Mechanical Cryocooler-Induced Disturbances

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Closed-cycle mechanical cryogenic refrigerators, or cryocoolers, are an enabling technology for next generation infrared (IR) sensors. Passive cryoradiators and stored cryogen systems have been used successfully in the past, but the increased cooling requirements for emerging systems cannot practically be met with these passive techniques. Modern systems are employing much larger focal plane arrays that dissipate more energy and have higher parasitic thermal loads than in the past. Additional "on chip" FPA data processing capability, such as time delay and integration (TDI) and analog-to-digital conversion (ADC), is further driving up the heat loads. While loads are going up, temperatures are going down. The desire to operate at long wave infrared (LWIR) wavelengths (>9 microns) for a broader range of remote sensing missions is driving the need for 35-40 K refrigeration, significantly colder than past systems that operated at shorter wavelengths. Unfortunately, the use of a mechanical rather than passive cryocooler introduces an additional jitter source that must be properly mitigated. Techniques include the use of inherently low vibration cryocoolers, closed-loop active vibration cancellation servo systems, damping struts, soft mounts, or a combination of these techniques. Implementation of these techniques within a proper system engineering context is presented.

1 INTRODUCTION

Mechanical cryogenic refrigerators (cryocoolers) have largely supplanted passive cryogenic refrigerators, such as stored cryogen systems and radiators, for space-borne infrared sensor cooling because of the mass and packaging advantages provided. This is particularly true for cryogenic instruments requiring less than 120 K where the mass advantage is particularly profound. Unfortunately, the use of a cryocooler introduces undesirable operational characteristics. For example, the active cryocooler obviously requires input power, which the passive systems do not. The optimum system approach typically comes down to the decision between accepting the cryocooler input power in return for the aforementioned mass/packaging advantages.

Another undesirable operational characteristic of the cryocooler is exported vibration, primarily arising from its moving mechanisms. These vibrations, if not properly managed, can lead to image blur. "Management" of the exported cryocooler vibrations can be accomplished in a number of ways. One approach is to select a cryocooler technology that is inherently quiet relative to competing technologies. For example, turbo Brayton cryocoolers tend to be quieter than the linear Stirling and pulse tube cryocoolers that dominate the space infrared sensor market.[1] Inherently quiet solid state cryocooler technology being developed jointly by Raytheon and Johnson Research and Development offers another promising option.[2] Another approach is to complement the use of a Stirling or pulse tube cryocooler with complex drive electronics that provide active vibration cancellation capability .[3] Still another approach is to mount the cryocooler to the supporting structure using a mechanically "soft" mount. In almost all cases, the cryocooler cryogenic interface is attached to the instrument, typically a focal plane array, with a flexible thermal strap to dampen the transmitted vibrations.

This paper addresses the challenge of managing exported cryocooler vibration primarily from the system (sensor level) perspective. Various integration techniques and technologies are explored. The strengths and weaknesses of the competing integration methods and the compatibility of different cooler architectures to those integration methods are

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discussed. The premise that the optimum cryocooler integration approach for each sensor design must be determined from within a proper system engineering context is supported.

2 FUNDAMENTAL VIBRATION MODELING AND ANALYSIS[4]

As illustrated in Figure 1, cooler vibration for most cooler systems is represented by a series of discrete sinusoidal force inputs consisting of a fundamental frequency and its harmonics, each with its individual amplitude. Solutions and analysis must consider both force levels and frequencies. Because the excitations consist of a fundamental frequency and its harmonics, it is often possible to significantly reduce the system level impact of cooler vibration by either adjusting the cooler frequency a small amount away from any system natural frequencies or conversely carefully designing the system structural resonance to be well away from the cooler drive frequency. At its most basic the cooler can be modeled in the system as three translational and three rotational series of sinusoidal forcing functions. Modeling can be performed on an axis by axis basis (better) or as a vector sum input basis. More sophisticated models may take into account a higher level of detail such as the individual force inputs at mount attachment points for pallet based systems or vibrational phase offsets for simultaneous multi-cooler operation.

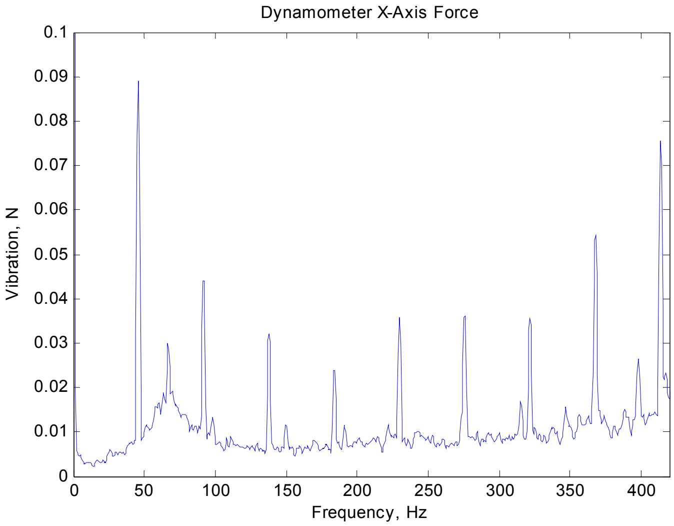


Figure 1. Typical Cooler Vibration Profile

2.1 Force disturbance approximation

The vibration source related to cryocoolers is largely due to imbalances in the moving parts of the assembly and internal gas flow. The net acceleration of the cooler assembly must then be reacted to by the attachments of the cooler to the support structure. This imbalance type of disturbance differs from other disturbances (such as propulsion jets etc) in that as the stiffness of the mount is reduced, the net attachment force is also reduced. In the limit, if the stiffness is reduced to zero, there is no transmitted force at all. Therefore it is more useful to think of the cooler vibration problem as motion isolation versus force isolation. However, if the moving mass is small compared to the overall cooler mass, the approximation of the disturbance as a force is valid and simplifies the analysis. Also since much cooler component vibration data is obtained from dynamometer force measurements it is more convenient to use the force model.

2.2 Exported vibration suppression techniques

2.2.1 System architectural approaches

It may be possible to determine that isolation is not required at all. Architectural approaches involving drive frequency tuning and duty cycling might provide enough vibration suppression to eliminate the need for additional isolation. If these methods prove inadequate then mechanical isolation will be required.

2.2.2 Cooler based vibration cancellation

Most space cryocoolers have feedback-based vibration cancellation with the exception being continuous flow coolers (i.e. coolers utilizing the Turbo Brayton Cycle), which have such inherently low vibration that it is unnecessary. Vibration cancellation uses the cooler drive electronics to reduce the emitted disturbance by balancing the drive forces. Load cell sensors at the mounting points or cooler mounted accelerometers are used to provide feedback, and sophisticated algorithms modify cooler drive parameters resulting in a significantly reduction in exported vibration. If this vibration cancellation is not sufficient for the system performance, it becomes necessary to proceed to more elaborate cooler mount designs incorporating vibration isolation.

2.3 Vibration isolation method is dependent upon mount

Any additional vibration suppression is highly dependent on the selected mounting configuration. Coolers can be mounted and integrated into a sensor in a variety of ways, fitting into three broad categories:

1. Cooler pallet or module mounting, in which the cooler package is located proximal to the sensor but on a structurally isolated pallet.
2. Remote or alternate mount location, in which the cooler or parts of the cooler is mounted some distance away from the sensor on a separate structure
3. Simple hard mount to the sensor

The design challenge is to mount the cooler with the sensor and surrounding structure, while optimizing the performance of the cooler. Locating cooling power as close to the sensor as possible will minimize parasitics and thermal strap losses, allowing for a higher temperature set point. Maintaining straightforward access for heat reject is necessary to allow for reduced hot side temperatures and efficient structural isolation along the heat reject path.[5]

Each of the mounting systems has advantages and limitations and affords varying degrees and methods of vibration isolation. The methods are described below and the most logical jitter suppression and isolation techniques to employ with that mount method are described. It is of course possible to use other combinations of mount method and jitter suppression, but they would less than optimal.

2.4 Cooler mounted on mallet approaches

One common cooler integration scheme is to mount all cooler modules on a single structure or pallet which is separate from the sensor structure, but with the cooler itself being proximal to the sensor with cold tip often penetrating directly into the sensor. This method allows for the greatest number of options in terms of jitter suppression while maintaining a minimal cold link delta T.

For any isolation to work it must be designed and analyzed on an axis by axis basis over all six degrees of freedom for the cooler mount. That means that the structure comprised of the cooler and all its mounting framework is treated as a rigid body with three translational degrees of freedom and three rotational degrees of freedom. An ideal isolation mount will then have the CG and moments of inertial centered on the mount in such a way that there is no cross coupling of vibrational modes; each mount axis can then be treated independently. Cross coupling can be modeled and taken into account but will prevent the design from being optimized as one axis will be interacting with the other.

Each mount axis is then designed to have the isolation required on that axis by reducing the system stiffness and/or adding damping until exported vibration levels meet requirements for that excitation axis within the system. Additionally, to bring rotational and translational frequencies into the same range the suspension mounts should be attached at the radius of gyration of the suspended mass.

For proper jitter isolation and suppression the ideal cooler mount has the cooler system structurally isolated entirely from both the sensor and the surrounding structure with the exception of strut mount. Exported vibration is isolated and/or damped at the strut. The extremely low amplitudes of vibration being considered often make typical damping and stiffness reduction methods ineffective due to stiction effects. Damping ratio Q values are amplitude dependent, and approach very high values (low damping) for low amplitude vibrations. [6]

The three cooler pallet mount isolation methods mentioned above all depend on the ability to mount the cooler on an exact constraint, (kinematic) 6 DOF mount with a prescribed set of stiffness and damping characteristics for all 6 DOF. Primary load path is via a hexapod strut set, often with provisions for damping and alignment adjustments. A simple approximation of the cooler pallet is a lumped spring-mass-damper system. Two counteracting requirements drive the stiffness requirements. Low system stiffness allows for low exported vibration under on-orbit vibrational operation, though launch survival and dynamic envelope requirements often dictate a system minimum stiffness. Large deflections under launch vibration in some instances can be controlled by soft impact snubbers.[6]

2.4.1 Heat rejection without introducing a structural path for vibration

Developing a flexible heat rejection path presents some particularly difficult challenges. The very small forces and deflections coupled with high heat loads require extreme attention to details of the design. Mechanical isolation of cryocooler vibration depends on the magnitude of the heat rejection. Techniques that are typically used for each regime are summarized as follows: These are to be considered approximate only.

5 to 20 W: Conductive flexible thermal straps
20 to 100 W: Flexible heat pipes (FHP)
100 to 1000 W: Loop Heat Pipes (LHP)

There are certain flexibility issues, however, that should be understood prior to use of conductive flexible thermal straps and/or flexible heat pipes. Thermal straps are typically manufactured from multiple layers of thin metal foils or braided wires. Test results for stiffness have shown significant deviations from linearity. The layers or wires can experience a considerable amount of binding and friction build up then slip, referred to as “stiction.” It is not certain that all thermal straps perform in this manner. However, based on these results, it is recommended that tests should be performed prior to their first use to verify stiffness characteristics.

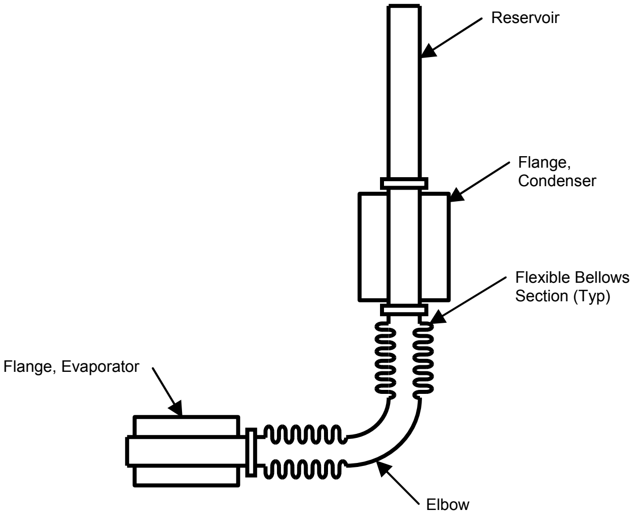


Figure 2. Flex heat pipes arranged for 6 DOF isolation

Flexible heat pipes typically rely on corrugated metal bellows for flexibility. These metal bellows are rigid in twist or torsion. Flexibility in all degrees of freedom can be achieved by introducing a 90° bend into the corrugated section. In addition, the metal bellows require a braided wire sleeve for pressure containment of the working fluid inside. As a result there are actually two different stiffness regimes. One is for small displacements, and the other for large displacements. Large displacements are representative of using a flexible heat pipe to connect two flanges that are misaligned or require adjustment, for example, whereas small displacements would be representative of on-orbit disturbances. It is felt that the two slopes are a consequence of the static and dynamic coefficients of friction associated with the braided wire sleeves, similar to the thermal strap characteristics discussed above. Unfortunately, the stiffness is significantly higher for small displacements than for large displacements. Again, tests should be performed to verify the stiffness characteristics of flexible heat pipes prior to first use.

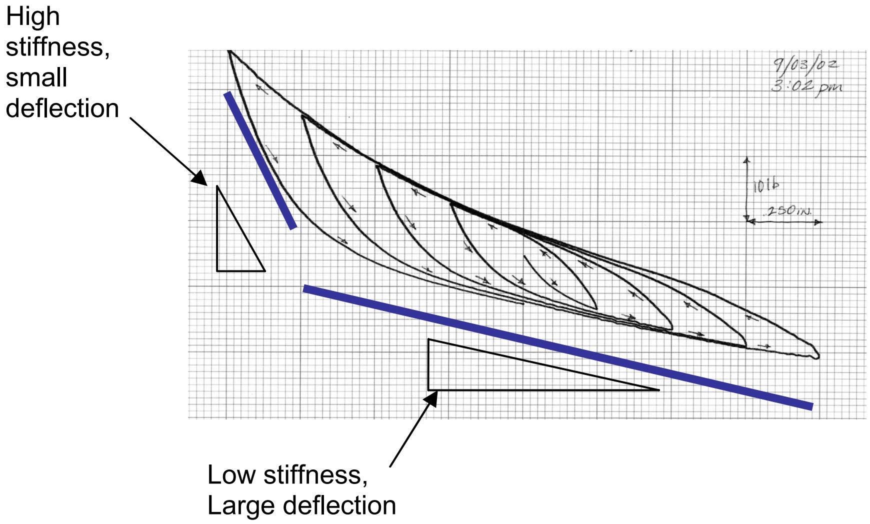


Figure 3. Typical load displacement behavior of flex straps and pipes

Unlike conductive thermal straps or flexible heat pipes, LHPs have small diameter transport tubes (0.094 to 0.156-inch OD) that connect the evaporator with the condenser. As a result, the stiffness of these devices is linear and obeys Hooke's law. For this reason, it is recommended that LHPs should be considered even for lower power regimes when on-orbit flexibility is a driver. Mini-LHPs are that are currently state-of-the-art technology should be considered for these applications.

2.4.2 Isolation approaches for pallet mounted cooler

Real cooler mounts deviate significantly from these ideal conditions. Geometrical and packaging constraints often prevent location of mounts in a balanced and non-cross coupled fashion. Structural requirements for mounting hard points, deviation from rigid body idealization, installation and thermal considerations all drive design and performance trades. Non-structural attachments to the cooler pallet often prevent proper cooler vibration isolation by providing significant load paths via heat reject heat pipes and vacuum or contamination closeouts. The challenge for good isolation often becomes to reduce stiffness of these attachments.

Detailed below are simplifications and descriptions of the various isolation approaches employed once a cooler system can be considered as a simplified lumped mass model, (cooler on pallet), on spring damper suspension.

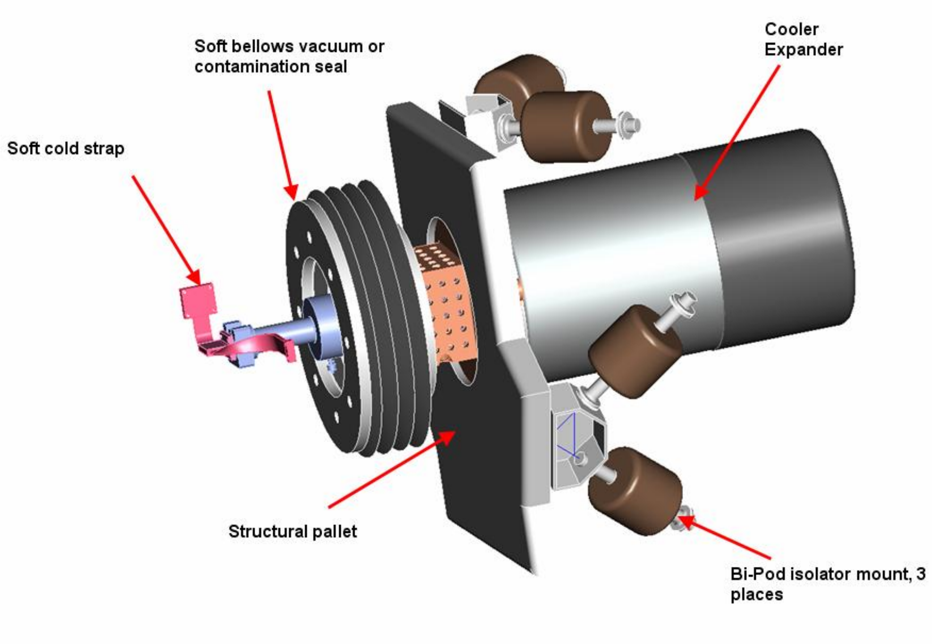


Figure 4. Cooler mounted on hexapod structural pallet

2.4.2.1 Passive Two Parameter Isolation

The simplistic approach to mount the cooler on isolation springs requires that the natural frequency of the suspension be low enough to provide adequate isolation. This produces large static 1 G displacements which can cause clearance problems during environmental loading. The other issue in using a simple spring is that the damping at resonance is uncontrolled leading to large motion during random vibration loading. Adding a damper to the spring system alleviates this concern but reduces the vibration isolation at higher frequencies due to the damper isolation characteristics.

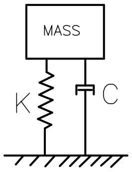


Figure 5. Conventional 2 parameter spring-damper system

2.4.2.2 Passive Three Parameter Isolation

A three parameter isolation approach adds another spring in series with the damper element.

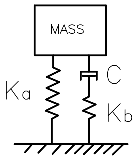


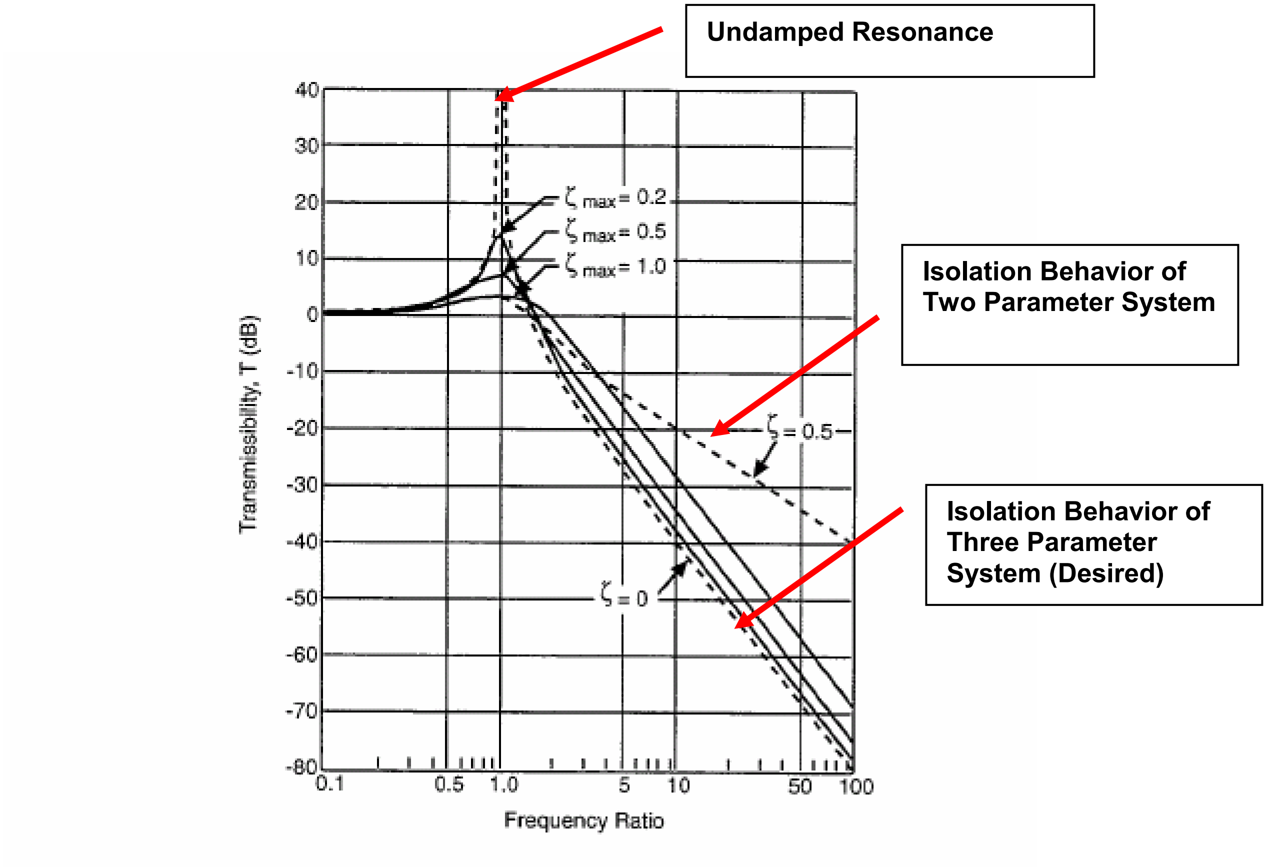
Figure 6. Modified 3 parameter spring-damper system

The addition of the series spring with damper allows the damper to be bypassed at high frequencies and is the key to how the system works.

Figure 7 describes the relative transmissibility of various isolation schemes.[7] An un-damped 2 parameter conventional spring-mass-damper exhibits an undesirably large resonance peak which can be excited by random vibration disturbances. Increasing the damping reduces the resonance, but at a cost of limiting the high frequency roll-off to **$1/\omega$** instead of the un-damped **$1/\omega^2$** thus limiting high frequency isolation. Adopting a 3 parameter damped strut can produce the **$1/\omega^2$** roll-off and still retain damping at resonance but at the cost of considerable design complexity.

2.4.2.3 Active Isolation[8]

Active struts can also provide extremely high performance isolation. The implementation of active isolation involves using a control system to remove the force transmitted by sensing the load and actively adding energy to the system to compensate. This can take the form of piezo-electric drives or voice coil magnet designs. The advantage of these systems is the potential for even greater isolation compared to passive systems but at significantly higher cost and complexity and significant impact on system reliability. The added electronics and power required would suggest that active isolation for cooler isolation would not be recommended.



Undamped Resonance

Isolation Behavior of Two Parameter System

Isolation Behavior of Three Parameter System (Desired)

Figure 7. Isolation comparison

All of these approaches have been used successfully. Sometimes simply being able to tune the cooler drive frequency is enough to solve the system vibration problem (Figure 8). Other applications may require fairly complex three parameter isolation. There is no single optimum approach.

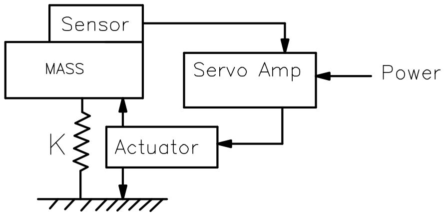


Figure 8. Active vibration canceling system

2.5 Alternate mounting location

2.5.1 Compressor off telescope

One method of reducing the impact of cooler vibration on sensor performance is to remotely locate the primary source of cooler vibration on a separate structure from the sensor. Two cooler technologies, Joule-Thompson (JT) and Turbo Brayton, are best suited to this approach and a third, split Stirling, can to a limited extent be adapted to this approach.

Turbo Brayton. The Turbo Brayton cooler is a continuous flow device and as such the fluid lines can be extended in length with relatively small impact on cooler performance. The lines are cold and thus incur parasitic losses with increase with length. While the TB cooler is already inherently low in vibration, this makes it possible to move the compressor off-sensor.

JT. JT coolers are particularly well-suited to remote compressor location. High pressure ambient lines can be extended in length between the high pressure compressor and cooling orifice with little or no impact to cooler performance either from the thermodynamic effects or parasitics. The cold JT expansion end effectively produces no vibration and can easily incorporate a cold reservoir.

2.5.2 Cooler off telescope cold transport loop

In larger systems it may be possible to utilize a cryogenic transport loop to convey cooling from the cooler, which is mounted remotely on a separate structure, to the sensor being cooled. A cooling loop will have significant parasitic heat loads and can be complex and difficult to integrate into a system, however they allow for excellent vibration isolation. Cooling loops have been successfully deployed in space and can offer great benefits with regards to the overall system package options.

2.5.3 Direct hard mount thermal storage and cooler duty cycling

The simplest cooler mounting system involves a direct hard mount to the sensor. The vibration isolation method of choice is not so much isolation as it is elimination. The method employed in this case is the integration of some substantial thermal mass via bulk material or phase change Thermal Storage Units (TSU). This type of arrangement is useful when the sensor has a limited duty cycle and the cooler can be turned off during imaging sequences. Heat loads are absorbed via the TSU and cooler vibration is eliminated. The cooling system and power supply will of course have to be of larger capacity.

3 SUMMARY

The inclusion of mechanical cryocoolers in present and future space-based systems is necessary in order to take advantage of larger and more capable detectors that are becoming available. A variety of methods are available for mitigating the effects of the disturbances emitted by these cryocoolers, ranging from the use of simple soft mounts to fundamental architectural solutions involving separate cryocooler mounting structures, thermal storage units, cryogenic transport loops, and the use of complex but inherently low-vibration cryocoolers. The choice of optimal solution is based on a large number of factors including the mission type, sampling rate, and characteristics of the structures employed in the sensor. As the sensitivity of the sensor system to cryocooler disturbance increases, so to does the complexity and cost of the mitigation method that must be used. As a result, the interplay between the sensor system as a whole and the cryocooler subsystem must be well thought-out and should be accounted for as early as possible during the design phase.

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