Experimental Demonstration of Cryogenic Cooler Disturbance Attenuation Using the Non-Linear Passive Isolation System

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This study demonstrates the isolation performance of a non-linear passive isolator to enhance the pointing performance through isolating disturbances induced by a spaceborne cryogenic cooler. In this paper, a non-linear passive isolation system with varying stiffness under launching and under on-orbit conditions to attenuate the micro-vibration of the cooler is proposed. The isolation system provides low stiffness for small excitation of the cooler during on-orbit operation and high stiffness when the excitation range is relatively larger under the launch environment. The performance of the passive isolation system is investigated through both a launch environment test and a micro-vibration measurement test of the cooler combined with non-linear passive isolators.

Key Words: Pulse Tube-Type Cryocooler, Compressor, Micro-Vibration, Isolator

1. Introduction

Pulse tube-type cryocoolers are widely used to cool down the focal plane of an imaging sensor to low cryogenic temperatures due to their advantages of simplicity, low cost and reliability.1–4) The major advantage of cryocoolers in comparison with Stirling-type coolers is the omission of any moving part in the cold head, which results in a significantly increased mean time to failure (MTTF). Besides these advantages, a secondary benefit is the low vibration output, which is essential for highly sensitive detectors. The measured vibration level of a pulse-tube cooler is reduced by a factor of four compared with that of a Stirling-type cooler.5)

For satellites, where high pointing stability is required, disturbances generated by the compressor of a cryogenic cooler are critical to the success of the mission.6) Cryogenic coolers provide a cryogenic temperature for sensitive infrared, gamma-ray, and X-ray detectors, but they also produce undesirable micro-vibration during operation, which may seriously affect the optical observation performance. Even though the micro-vibrations from the compressor are very small, they are a major source of pointing performance degradation. Future space missions, such as highly accurate land observation satellites and celestial observation, sometimes require control and management of micro-vibration. The amplitudes of acceptable micro-vibration are also becoming much lower. The micro-vibration isolation for a compressor of a cryogenic cooler is an important technology and mounting an isolator to attenuate the micro-vibration transmitted to the satellite structure is one way to avoid degradation of image quality caused by the blurring of images.

Several types of vibration isolation systems for such applications have been proposed and investigated.7–10) For example, Veprik et al.7) proposed a vibration protection system for an airborne infrared application which relies on a stiff and heavily damped vibration isolator in combination with an undamped tuned vibration absorber for use in the design of the linear split Stirling cryocooler. Flint et al.8) proposed active vibration control systems to reduce cryocooler disturbances. The system achieved reductions of more than two orders of magnitude and showed considerable promise for reducing the effects of cryocooler disturbance loads. It can also be extended to the fly-wheel isolation. Kim et al.9) proposed a passive tuned vibration absorber (TVA) to effectively suppress the monotone vibrations in the cooler system. To overcome the isolation performance degradation resulting from the detuned and mistuned TVA, they10) also proposed the use of magneto-rheological elastomer (MRE) for vibration mitigation of a cryogenic cooler. Veprik et al.11) introduced an ultra-low vibration linear split Stirling cryogenic cooler achieved by optimizing the counterbalance mass and springs rates so as to produce full cancellation of the fundamental component of the vibration export without affecting the overall cooling performance. The micro-vibration output of the compressor can also be attenuated by proper thermal control because an asymmetric temperature gradient on the compressor is a potential source of an increase in micro-vibration. Oh et al.12) proposed a heat pipe implementation method, which is very effective to guarantee a symmetric temperature gradient on the compressor even though a heat pipe might have failed.

Since passive isolation approaches do not supply energy to the system, passive systems are always stable, and the passive methods are easier to implement in actual systems than the complicated active methods because they do not need controllers, sensors or filters. In this study, to attenuate
the micro-vibration of the compressor, a non-linear passive isolator with varying stiffness under launching and under on-orbit conditions is proposed and investigated. The characteristic variation of the isolator depends on the excitation range of the isolator. The isolator provides high stiffness during the launch to protect the compressor from the launch loads and low stiffness for on-orbit isolation of the compressor disturbances when the excitation is much smaller than the launch environment. To demonstrate the effectiveness of the isolation system, we performed launch environment tests and a micro-vibration measurement test of the compressor combined with the isolator. The test results indicate that the non-linear passive isolation system guarantees the structural safety of the compressor under the launch environment and effectively attenuates the micro-vibration of the compressor under on-orbit conditions as intended in the designs.

2. General Description of a Spaceborne Cryocooler Assembly

2.1. Compressor of a cryogenic cooler

Figure 1 illustrates a compressor assembly of a cryogenic cooler to cool down the focal plane of the IR imager to low cryogenic temperatures. The compressor assembly is mainly composed of a compressor, housing for the compressor, a non-linear passive isolator and a transfer line. The transfer line with helium gas is connected to the cold finger of the IR imager to cool down the focal plane. The micro-vibration of the compressor is attenuated by the isolator with non-linear characteristics under launching and an on-orbit environment. The thermal interface for the heat pipe installation is on the upper side of the compressor. The compressor weighs approximately 4.0 kg including the case, is approximately 200 mm in length and 80 mm in width, and generates heat of 35 W over the entire orbit at end of life (EoL) to cool down the focal plane of the imager to 80 K except for the safe-hold mode when the compressor is turned off. The allowable operating and non-operating temperatures of the compressor are from −10 to 45°C and from −30 to 65°C, respectively. To maintain the desired cooling performance and comply with the allocated mission lifetime, the thermal control target temperature of the compressor is less than 35°C. The operation frequency of the compressor is 36 Hz and it generates a maximum disturbance force of 1.46 and 0.13 N in the longitudinal and lateral directions, respectively.

2.2. Thermal design configuration of compressor assembly

Figure 2 shows a 3-D view of the thermal design configuration of a compressor assembly.9) The thermal control subsystem for the compressor assembly is mainly composed of a heat pipe, a radiator and a heater. The compressor assembly is mounted on an optical bench located on the inside of the spacecraft (S/C), and the radiator is installed on the S/C panel that is oriented towards deep space. To unload heat from the cooler to the radiator, a heat pipe is used. Heat pipes with a diameter of 9 mm are rigidly clamped by bolting from the upper side of the compressor. Heater control is implemented to keep the compressor within the non-operating allowable temperature threshold range under the safe-hold condition.

The heat pipe for thermal control of the compressor should be designed to guarantee the heat-transportation capability and isolator performance in an on-orbit environment. To attenuate the micro-vibration of the compressor with axial force of 1.46 N at an operating frequency of 36 Hz, a non-linear characteristic isolator with varying stiffness under launching and on-orbit conditions is mounted on the compressor as shown in Fig. 2. The isolator provides high stiffness during the launch to protect the compressor and transfer line from the launch loads and low stiffness for on-orbit isolation of the micro-vibration. For effective performance in on-orbit isolation, the position of the isolator should be kept within a low-stiffness range of ±0.5 mm from the nominal position. Therefore, transmission of a larger force to the compressor through thermal deformation of the heat pipe is not allowed under an on-orbit environment. In addition, if the axial stiffness of the heat pipe is too high, the level of the transmitted micro-vibration force through the heat pipe is increased. As a result, the isolator performance is also degraded because the compressor that is supported by an isolator with low stiffness is constrained by the stiffened heat pipe. Therefore, a bent-shape heat pipe with diameter of 9 mm and relatively low stiffness is incor-
porated in the heat pipe design as shown in Fig. 2 to ensure
the isolator’s performance under on-orbit conditions. The
heat pipe routing is also designed horizontally to the ground
to avoid performance degradation of the heat pipe due to the
1 G gravity effects during the ground test.

The heat pipe should also be designed to guarantee ther-
mal control performance of the compressor. The thermal
performance of the 9 mm heat pipe has been demonstrated
by on-orbit thermal analysis using thermal parameters
obtained from the thermal balance test performed in the
space simulated vacuum environment.12)

3. Non-Linear Passive Isolator

3.1. Design drivers for non-linear passive isolator

The starting point of the isolator design is to meet the line
of sight (LOS) stability requirement of the whole telescope.
The calculated LOS stability is less than $5 \times 10^{-8}$ rad,
although the details obtained from the jitter analysis are
not shown here. The transmitted force level of the compres-
sor through the isolator in the longitudinal direction of
the compressor shall be less than $0.73 N_{rms}$ to meet the
above LOS requirement on-orbit configuration because the
micro-vibration in this direction of the compressor is the
main source of degradation of LOS stability.

The isolator should also be designed to guarantee the
structural safety of the transfer line and the compressor itself
under the launch environment. The allowable deflection of
the transfer line is within $\pm 3$ mm under random and sine
vibration loads.

3.2. Configuration and characteristics of non-linear
passive isolator

Figure 3 shows the external configuration of the non-
linear passive isolator assembly to isolate the micro-vibra-
tion of the compressor. The isolator is composed of a shaft
for the mechanical connection between the compressor and
isolator assembly, visco-elastic rubber elements with non-
linear characteristics under launch and on-orbit environ-
ments, a support bracket with visco-elastic rubber elements,
and a mechanical limiter to protect and maintain the transfer
line of the compressor to within the allowable deflection
range in lateral direction movement of the compressor under
a launch environment. The axis of the visco-elastic rubber
elements is arranged so that they are in the plane of the
center of gravity of the compressor. This minimizes the
additional movement of the fixation point of the transfer line
resulting from rotations of the compressor. The axes are also
arranged normal to each other and 45 degrees relative to the
length axis of the compressor. Through this arrangement,
each support bracket can be loaded using lateral forces
and an additional launch lock device to guarantee structural
safety under a launch environment is not necessary.

Figure 4 shows a visco-elastic rubber element with char-
acteristics of low and high stiffness according to the excita-
tion range of the compressor. This part makes a major
contribution to the isolation performance of the isolator.
Each rubber element inside the four support brackets pro-
vides a range of $\pm 0.5$ mm in all directions with low stiffness,
and a much higher stiffness and loading capability when
exceeding this low-stiffness range. This offers a very low
stiffness for small excitation during in-orbit operation of
the compressor and, simultaneously, the capability to take
over high loads during launch.

To measure the characteristics of the non-linear passive
isolator in the lateral direction ($x$-axis) of the compressor
assembly, a constant rate extension and contraction test of
the compressor assembly is performed. Figure 5 shows the
test set-up configuration for the constant rate extension/contraction test of the compressor assembly. The support
brackets are fixed to the cross-head, and the compressor is
connected to the load cell. In the test, the load is applied
to the compressor assembly with the isolator by raising
and lowering the cross-head, and the load ($p$) and displace-
ment \((d)\) of the isolator are measured. Figure 6 presents an example of the \(d-p\) relation in the lateral direction of the compressor assembly obtained from the constant rate (0.5 mm/min) extension/contraction test. The test results indicate that the isolator assembly shows non-linear characteristics. The stiffness of the isolator increases as the displacement increases. The low-stiffness value of the isolator at low-displacement range of \(\pm 0.5\) mm is 28 N/mm and the high-stiffness value is 98 N/mm when exceeding the low-stiffness range.

To measure the characteristics of the isolator in the longitudinal direction (\(y\)-axis) of the compressor, a low-level sine sweep test of the isolator assembly is performed to search for a natural frequency of the compressor supported by the isolation system. The micro-vibration level of the compressor on this axis is much higher than that of the lateral axis. It is also critical to pointing performance degradation. Therefore, the main isolation target of the compressor is the \(y\)-axis. Figure 7 illustrates the test set-up configuration of the low-level sine sweep tests. A sinusoidal exciting force with amplitude of 1 N is applied to the compressor assembly using a vibration shaker, and the acceleration of the isolator is measured using an accelerometer. To demonstrate the 0 g condition in the orbit condition and keep the nominal position of the low stiffness range of the isolation system, the compressor is hung by rubber string as shown in Fig. 7.

Figure 8 shows the frequency response spectrum of the compressor assembly obtained from the low-level sine sweep test on the longitudinal axis. The result indicates that the 1st eigenfrequency of the compressor with the isolation system under the 0 g condition is approximately 18.2 Hz, which is the main frequency in longitudinal axis movement of the compressor assembly. The 2nd eigenfrequency of 21 Hz is observed in the torsional mode of the compressor assembly.

On the basis of these test results and the mechanism of the isolation system, we propose an equivalent model of the compressor isolation system as shown in Fig. 9. The spring elements \(k_h\) and \(k_t\) indicate the stiffness of the heat pipe and transfer line in alignment with the \(y\)-axis of the compressor, respectively. The isolator model is composed of dashpot elements \((c_H \) and \(c_L)\) and spring elements \((k_H \) and \(k_L)\). The isolator is basically a passive system but the characteristic variation of the isolator depends on the excitation range of the isolator. The isolator provides very low stiffness \((k_L)\) and damping \((c_L)\) for small excitation during on-orbit operation of the compressor within a range of \(\pm 0.5\) mm in all directions. The isolator also provides a much higher stiffness \((k_H)\) and damping \((c_H)\) when the excitation range is relatively large under a launch environment, and this makes it possible to take over the high load capability. Table 1 summarizes the measured and estimated stiffness values of the isolation system and the subscripts \(x\), \(y\) and \(z\) indicate the axis of the compressor defined in Fig. 2. In Table 1, the stiffness values of the isolation system are summarized because the on-orbit isolation concept is to put the eigenfrequency of the compressor assembly derived from the compressor mass (4.0 kg), \(k_{by}\), \(k_{by}\) and \(k_{by}\) on the frequency region lower than the compressor operation frequency of

![Fig. 6. Isolator characteristics \((d-p)\) in the lateral direction of the compressor assembly obtained from the extension/contraction test.](image)

![Fig. 7. Test set-up configuration for the low-level sine sweep test of the isolation system.](image)

![Fig. 8. Frequency response spectrum of the compressor assembly obtained from the low-level sine sweep.](image)

![Fig. 9. Equivalent model of the compressor assembly with the isolation system.](image)
36 Hz and as lower as possible. The stiffness value of $k_{Ly}$ in Table 1 is estimated from the 1st eigenfrequency of the compressor assembly under the 0 g condition, as shown in Fig. 8. The expected decreasing rate through the isolation system is 1.18 when the measured damping value of $c_{Ly}$ is 7.6 Ns/mm.

4. Non-Linear Passive Isolator Performance Test

4.1. Launch environment test

To confirm the effectiveness of the non-linear passive isolator design for the launch environment, sine and random vibration tests are performed. Figure 10 shows the sine vibration test specifications for $z$-axis excitation of the compressor and the output accelerations measured at the CoG of the compressor. The $z$-axis is the most critical axis. The highest acceleration at the CoG of the compressor under a sine qualification load in $z$-direction excitation is 30 g at 51 Hz, and the amplification factor is 3.3 compared to the 9 g input. Figure 11 shows the random qualification test results in $z$-axis excitation of the compressor. The maximum accelerations occur below 30 Hz, and acceleration at higher frequencies is strongly damped out as expected due to the effectiveness of the visco-elastic rubber elements of the isolation system. The maximum acceleration measured at the CoG of the compressor is 5.95 grms and 30 g for the random and sine vibration tests, respectively. These are smaller than the compressor design load of 45 g. The movement of the transfer line is also limited to 2.1 mm, which are sufficient margins to the allowable stress in the transfer line. The test results indicate that the non-linear passive isolation system is effective to guarantee the structural safety of the compressor under a launch environment.

4.2. Isolation performance test set-up configuration

To confirm the effectiveness of the non-linear passive isolator design for compressor disturbance reduction, an isolation performance test on the KISTLER table is performed under a simulated on-orbit 0 g condition. Figure 12 shows the isolation system performance measurement test set-up configuration on the KISTLER table. In the test, a whole imaging sensor assembly composed of an imaging sensor, buffer volume and compressor with an isolation system are mounted on the KISTLER table, and the compressor is hung by a gravity compensation 0 g device to demonstrate on-orbit condition, so that the isolator elements are centered and provide the nominal low stiffness in all directions. The mass of the compressor is compensated by the counter mass which is equivalent to the mass of the compressor assembly, and this is representative of the in-orbit configuration. The transfer line is considered in the test which affects the natural frequency of the isolation system, but the heat pipe was not applied in the test because the stiffness of a heat pipe is negligibly small compared with those of the transfer line and isolation system in the $y$-direction. The compressor is

| Item               | Stiffness [N/mm] |
|--------------------|------------------|
| Isolation system   |                  |
| $k_{Hx,z}$         | 98.0             |
| $k_{Lx,z}$         | 28.0             |
| $k_{Ly}$           | 48.7             |
| Transfer line      |                  |
| $k_{ty}$           | 9.8              |
| Heat pipe          |                  |
| $k_{hy}$           | 0.98             |

Fig. 10. Results for the sine vibration test of the compressor assembly in $z$-axis excitation.

Fig. 11. Results for the random vibration test of the compressor assembly in $z$-axis excitation.

Fig. 12. Test set-up configuration for the isolation performance test on the KISTLER table.
operated by a cooler drive electronic device with an operation frequency of 36 Hz. Liquid cooling is applied to control the temperature of the warm-end part which is directly connected to the focal plane of the imaging sensor, and the heat on the warm-end is generated during imaging sensor operation when the focal plane is cooled down to the target temperature of 80 K. To keep the compressor within the allowable operating temperature range during operation, air cooling is applied.

In the test, the following three cases of measurements are performed under different constraint conditions of the isolator.

1) Rigid condition
To identify the compressor disturbance level itself, measurement is performed under rigid connection of the compressor by exchanging the isolator assembly to the rigid brackets. In addition, the disturbance level of the compressor with various warm-end temperatures (15, 25 and 35°C) is measured to identify the relationship between the warm-end temperature and the disturbance level of the compressor.

2) 0 g condition
To investigate the effectiveness of the isolator design and confirm the design requirement compliance, measurement is performed with a gravity compensation device as shown in Fig. 12. The rubber elements in the isolator are free, and the 0 g device is applied to the compressor, so that the low-stiffness visco-elastic rubber elements are centered and provide the in-orbit configuration.

3) 1 g condition
To qualitatively check the behavior of the isolator when the compressor is shifted out of the low-stiffness range for whatever reasons, such as misalignment of a heat pipe or unexpected thermal distortion from the heat pipes, the test is performed under the 1 g condition. The isolator elements are free, but because no gravity compensation device is mounted on the compressor, the rubber elements in the isolator are compressed so that they bridge the low-stiffness range and provide higher stiffness in the gravity direction.

4.3. Isolation performance test results summary
Figure 13 shows an example of the test results obtained from the rigid condition of the compressor when the input power to the compressor is 60 W, and the operation frequency of the compressor is 36 Hz. In the test, the warm-end temperature is set to 35°C, and the measurement is started when the focal plane temperature of the imaging sensor is stabilized to 80 K. The maximum disturbance level of the compressor is 1.46 N in the longitudinal direction of the y-axis at 36 Hz, and this is the main disturbance source to degrade the LOS stability. The disturbance level of the lateral direction of the x- and z-axes is very small compared with that of the longitudinal direction of the y-axis, and these are not critical to the LOS stability.

Figure 14 shows the relationship between the warm-end temperature of the imaging sensor and the disturbance level of the compressor obtained from the rigid condition of the compressor. The input powers to the compressor at various warm-end temperatures are also plotted in the figure. The warm-end temperatures of 15, 25, and 35°C correspond to the predicted temperatures at the coldest, nominal and hottest temperatures on-orbit. They are obtained from the on-orbit thermal analysis of the imaging sensor assembly. The test results indicate that the disturbance level slightly increases as the warm-end temperature increases, but the differences in the disturbance level are not particularly high. The results also show that the input power to the compressor increases as the warm-end temperature increases.

Figure 15 shows the test results obtained from the 0 g condition of the compressor supported by the isolation system. The maximum peak force value at 36 Hz in the y-axis is 1.27 N, which is a much lower value than the rigid condition as shown in Fig. 13. The disturbance level of the compressor is reduced by a factor of 1.23 compared with that of the rigidly fixed compressor. This is almost the same value as the predicted value of 1.18 estimated from Fig. 8 and Table 1. The disturbances in the high-frequency region are also greatly damped out. The disturbances in x- and z-axes
are also dramatically damped out in the high-frequency region, although the peak values at 36 Hz are slightly higher or almost the same as those obtained under the rigid condition. These facts indicate that the passive isolation system proposed in this study is effective for attenuating the compressor disturbances which seriously degrade the LOS stability.

Figure 16 compares the time profiles of the compressor disturbances with and without the isolation system. The figure indicates that the disturbance level of the compressor without the isolation system is higher than that of the compressor with the isolation system because the higher frequency modes are excited. On the other hand, the time profiles obtained from the isolation system show that the isolator effectively attenuates the disturbances with higher frequency modes.

Table 2 summarizes the test results for the peak force level (N) at 36 Hz and the root mean square value ($N_{\text{rms}}$) of the compressor disturbance force obtained from the time profiles. The RMS value of the compressor with the isolation system is 0.8 $N_{\text{rms}}$ and the decreasing rate with respect to the value under rigid conditions is 2.2. This decreasing rate is much higher than the value of 1.23 obtained from the peak force level at the operating frequency of 36 Hz as shown in Figs. 13 and 15. The RMS value is slightly higher than the target value of 0.73 $N_{\text{rms}}$ to meet the LOS stability requirement. However, the current design of the isolation system seems to be applicable because the expected disturbance level of the compressor for the real application is lower than that of the compressor tested in this study.

Figure 17 shows the test results obtained from the 1 g condition without the 0 g device under the assumption that the compressor is shifted out of the low-stiffness range for some reason, such as misalignment of a compressor heat pipe or thermal deformation of a heat pipe. The rubber elements of low stiffness are in contact with the high-stiffness rubbers, and they are compressed by the 1 g effect in gravity direction. The peak value at 36 Hz in the $y$-direction is increased to approximately 3.4 N. In addition, much higher disturbance levels are observed in the directions other than the rigid condition of the compressor because the disturbance level is amplified due to the frequency coupling of the isolation system under the 1 g condition with the compressor excitation frequency of 36 Hz. The eigenfrequency of the isolation system under the 1 g condition measured from the modal test is 34 Hz. This indicates that there is a possibility to increase the disturbance level if a heat pipe is not well aligned to maintain the low-stiffness range of the isolator.
5. Conclusion

In this study, a non-linear passive isolator for attenuation of compressor disturbance is proposed to enhance LOS stability, and the effectiveness of the design is investigated by a launch environment test and a performance test on the KISTLER table. The passive isolator provides high stiffness during the launch to protect the compressor from launch loads and a low stiffness for on-orbit isolation of the compressor disturbances. The launch environment test results demonstrate that the isolation system is effective to guarantee the structural safety of the compressor under a launch environment. The isolation performance test results indicate that the isolator proposed in this study is effective in reducing the disturbance level of the compressor to enhance the LOS stability. The test results also indicate that the heat pipe installation to maintain the low-stiffness range of the isolator is an important parameter to maintain isolation performance.

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