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Numerical Feasibility Study for a Spaceborne Cooler Dual-function Energy Harvesting System

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Abstract

Spaceborne cryocoolers produce undesirable micro-vibration disturbances during their on-orbit operation, which are a primary source of image-quality degradation for high-resolution observation satellites. Therefore, to comply with the strict mission requirement of high-quality image acquisition, micro-vibration disturbances induced by cooler operation have always been subjected to an isolation objective. However, in this study, we focused on the applicability of energy harvesting technology to generate electrical energy from micro-vibration energy of the cooler and investigated the feasibility of utilizing harvested energy as a power source to operate low-power-consumption devices such as micro-electromechanical system (MEMS) devices. A tuned mass damper (TMD)-type electromagnetic energy harvester combined with a conventional passive vibration isolator was proposed to achieve this objective. The system performs the dual functions of electrical energy generation and micro-vibration isolation. The effectiveness of the strategy was evaluated through numerical simulations.

Key words: Spaceborne Cryocooler, Micro-vibration, Energy Harvesting, Tuned Mass Damper (TMD)

1. Introduction

The quality of high-resolution images obtained from observation satellites can be degraded by undesirable micro-vibrations induced by on-board appendages that have mechanical moving parts, such as reaction-wheel assemblies, control moment gyros, and gimbal-type antennas [1-3]. In addition, the micro-vibration induced by pulse-tube-type cryocooler is also known as one of the undesirable disturbance sources. To meet the objective of obtaining high-quality images from observation satellites, micro-vibration disturbances from the cooler have always been subjected to an isolation objective and lots of technical efforts have been conducted.

For example, passive-type cooler isolators are generally used to achieve the desired isolation performance [4-10] owing to their simplicity and reliability. However, most studies have only dealt with on-orbit micro-vibration isolation, and have not addressed launch environment. To guarantee the structural safety of the cooler supported by a passive isolator and the isolator itself during harsh launch loads, it is essential to consider holding-and-release mechanisms. However, this approach increases the system complexity and lowers its reliability, in addition to increasing the total mass of the system. In order to overcome these drawbacks, Oh et al. [8] proposed a passive cooler isolation system that can be used in both launch and on-orbit micro-vibration environments without requiring an additional holding-and-release mechanism. Its effectiveness was qualified in outer-space. However, because of insufficient damping in the isolator design, the attenuation performance of the isolator in sine vibration tests was poor even when the highest acceleration at the cooler was lower than its design load. Therefore, with the aim of enhancing the vibration attenuation performance in severe launch environments, while effectively isolating the micro-vibration of the cooler, Oh et al. [9] also proposed a passive isolation system employing a pseudo-elastic shape memory alloy mesh washer. This can be used as a smart adaptive isolation system in compliance with a given

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vibration environment. The effectiveness of the isolation system in launch environments was demonstrated through sinusoidal-vibration, random-vibration, and shock tests at qualification levels. Its effectiveness in the micro-vibration environments was also evaluated through micro-vibration measurements [10].

As in the above-listed examples, all efforts were directed toward reducing mechanical vibration disturbances in an on-orbit environment to obtain high-quality satellite images. However, this study focused on the feasibility of positively utilizing the cooler-induced micro-vibrations as a source of renewable power by adopting energy harvesting technology.

Energy harvesting technology has recently received considerable attention in the context of its increasing use for portable, implantable, and ubiquitous sensor nodes. Additionally, low-power-consuming microsystems in daily applications have triggered a need for compact and efficient energy management. Therefore, over the past decade, a great deal of research has been directed toward developing energy harvesters. Micro-electromechanical systems (MEMS)-based energy harvesters are especially widely reported owing to their easy application without limitations of area and environment [11-14]. Bang et al. [11] reported a bulk micro-machined vibration-driven electromagnetic energy harvester for a self-sustainable wireless sensor node application. The validity of the harvester design was verified by numerical and experimental studies. Thomas et al. [12] proposed a micro-power electromagnetic harvester to generate electrical power from the seismic motion of the human body. The purpose of the harvester was to power portable body-worn sensors and personal electric devices. The validity of this harvester design was investigated through practical experiments using human walking motions. Ibrahim et al. [13] proposed a cantilever harvester based on MEMS to power a wireless sensor node. An experiment demonstrated that, by maximizing the electrical-mechanical coupling coefficient, the MEMS-based harvester could produce 0.5 μW at the impedance matching point. Dibin et al. [14] reported a tunable vibration-based electromagnetic micro-generator. This tunable micro-generator has a wide tuning range, which allows it to produce wide-band electrical power over a wide range. They also evaluated the effectiveness of the generator design through theoretical analysis and experimental results.

As exemplified above, many studies have been conducted for on-ground energy harvesting applications. Several types of energy harvesting applications are being implemented on a commercial scale. However, the efforts to facilitate their application in space have been scarce, and only a few basic research studies have been performed. For example, Makihara et al. [15] proposed a semi-active energy-harvesting vibration-suppression system using a piezoelectric actuator, and verified its feasibility through a vibration-suppression test on a large truss structure.

This study extended the energy harvesting technology to a space application, aiming to positively utilize the micro-vibration energy as a renewable power, while still striving to isolate the vibrations to avoid image quality degradation in an on-orbit environment. A spaceborne pulse-tube-type cryocooler was selected as the vibration energy source, which provides several advantages. For instance, the harvester can continuously generate electrical power because the cooler is continuously operated over the entire orbit except during the safe-hold mode when the cooler is powered off. In addition, the singular operating frequency of the cooler is easy to characterize, which is advantageous for designing the energy harvester. Meanwhile, the micro-vibrations induced by on-orbit operation of the cooler are always undesirable for observation satellites because of the aforementioned reasons. Therefore, the level of micro-vibration transmitted to the satellite structures should be managed strictly, even if it is regarded as a potential energy source for the energy harvesting. To implement the dual objectives of scavenging the micro-vibration energy and simultaneously isolating the micro-vibration from the cooler, we propose an electromagnetic energy harvester. To maximize the efficiency of the harvester, it was designed in the form of a general tuned mass damper (TMD) and integrated with a conventional passive vibration isolator [9, 10]. The effectiveness of the proposed system in performing the dual functions of electrical energy harvesting and micro-vibration isolation was demonstrated through a numerical simulation in line with investigating the feasibility of utilizing the harvested energy as a power source for operating low-power-consumption devices such as MEMS devices. In addition, the energy generation capability of the proposed system in a harsh launch vibration environment was numerically investigated.

2. Energy Harvesting System Combined with Conventional Passive Vibration Isolator

To investigate the feasibility of simultaneously harvesting electrical energy and enhancing micro-vibration isolation capability by integrating a TMD electromagnetic energy harvester with a conventional passive vibration isolator, we proposed a numerical simulation model, as shown in Fig. 1. In this study, this model is referred to as a complex system owing to its dual functions of energy harvesting and

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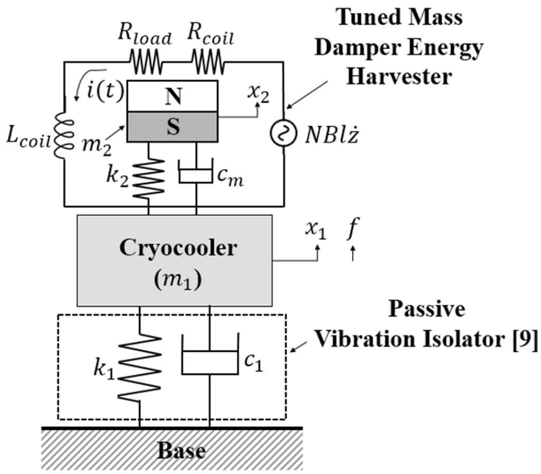

Fig. 1. Numerical simulation model of the complex system

micro-vibration isolation. This complex system comprises a cooler that induces micro-vibrations, a conventional passive vibration isolator [9] to support the cryocooler with low stiffness to isolate the micro-vibration from the cooler, and a TMD-type electromagnetic energy harvester integrated on the cooler. In this model, m_1 indicates the mass of the cooler; the dashpot element c_1 and the spring element k_1 indicate the damping coefficient and stiffness of the conventional vibration isolator, respectively. In the case of the TMD-based energy harvester, m_2 and k_2 indicate the mass of the permanent magnet and the stiffness of the plate spring tuned to the same main excitation frequency of the cooler, respectively. The dashpot element c_m indicates the mechanical damping coefficient of the TMD-based energy harvester. In addition, the TMD-based energy harvester comprises electrical parts, such as L_{coil} for the generation of electrical power from the induction coil, R_{coil} as an internal coil resistance, and load resistance R_{load} for impedance matching. In the figure, i represents the electrical current.

2.1 Equations of Motion

The micro-vibration disturbance load induced by the cooler is given by

$$f = f_0 e^{i\omega t}, \tag{1}$$

where, f_0 and w denote the amplitude and main excitation frequency of the cryocooler, respectively. Once the absolute displacements of the cooler and TMD-based harvester are defined as x_1 and x_2, respectively, the equations of motion for the complex system, as shown in Fig. 1, can be expressed as

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 - c_m\dot{z} - k_2z = f - \kappa i, \tag{2}$$

$$m_2\ddot{z} + c_m\dot{z} + k_2z = \kappa i - m_2\ddot{x}_1, \tag{3}$$

$$\kappa\dot{z} - L_{coil}\frac{di}{dt} - (R_{coil} + R_{load})i = 0, \tag{4}$$

where, z denotes the relative displacement x_2-x_1 between the cooler and the permanent magnet of the TMD-based harvester. Furthermore, the electromechanical coupling coefficient κ can be calculated by

$$\kappa = NBl, \tag{5}$$

where, N is the number of coil turns, B is the average magnetic flux density, and l is the coil length across the magnetic flux.

On account of the small value of the coil inductance L_{coil} and low mechanical vibration frequency, the voltage generated into the internal inductance can be ignored as L_{coil} $(di/dt)\approx 0$. Therefore, the electrical current i can be simplified as following from Eq. (4)

$$i = \frac{\kappa\dot{z}}{R_{load}+R_{coil}}, \tag{6}$$

Equations (3) and (6) can be combined to yield:

$$m_2\ddot{z} + (c_m + \frac{\kappa^2}{R_{load}+R_{coil}})\dot{z} + k_2z = -m_2\ddot{x}_1. \tag{7}$$

The electrical damping coefficient c_e, which is attributable to electrical power generation, can be defined as

$$c_e = \frac{\kappa^2}{R_{load}+R_{coil}}. \tag{8}$$

The total system damping-coefficient can then be defined as the sum of the mechanical and electrical damping-coefficients as

$$c_T = c_m + c_e. \tag{9}$$

From the above Eq. (9), Eq. (7) can be reduced to

$$m_2\ddot{z} + c_T\dot{z} + k_2z = -m_2\ddot{x}_1. \tag{10}$$

Once the absolute displacement of the cooler $x_1=X_1 e^{i\omega t}$, the relative displacement between the cooler and permanent magnet of the TMD harvester $z=Ze^{i\omega t}$, and the total dissipated energy into the dashpot elements of the energy harvester, composed of c_m and c_e, are given, the total average power P_T during each cycle derived from the instantaneous power c_T $(dz/dt)^2$ becomes

$$P_T = \frac{1}{\tau}\int_0^{\tau}(c_T(Z\omega\, e^{i\omega t})\overline{(Z\omega e^{i\omega t})}dt = \frac{c_T|Z|^2\omega^2}{2}, \tag{11}$$

where, τ indicates the period of the cycle 2π/ω. In addition, Eq. (11) can be rewritten by substituting for Z from Eq. (10)

$$P_T = \frac{1}{2}\frac{c_T m_2^2\omega^6 X_1^2}{\{(k_2-m_2\omega^2)^2+(c_T\omega)^2\}}. \tag{12}$$

The total maximum generated power $P_{T_{max}}$ within the TMD-based energy harvester occurs when the eigenfrequency of the harvester equals the excitation frequency of the cooler (i.e., $\omega=\sqrt{k_2/m_2}$) and is given by

$$P_{T_{max}} = \frac{m_2^2\omega^4 X_1^2}{2c_T} \quad (13)$$

In practice, P_T can be expressed as a combination of both the mechanically dissipated energy P_M and the electrically harvested energy P_E. In this manner, the energy transferred to the electrical domain is equal to the absorbed energy into electrical damping coefficient c_e. Therefore, when $P_E=P_T\times(c_e/c_T)$, the maximum power delivered to the electrical domain $P_{E_{max}}$ is given by

$$P_{E_{max}} = P_{T_{max}} \times \frac{c_e}{c_T}, \quad (14)$$

where, c_T and c_e account for $2\xi_T m_2\omega_2$ and $2\xi_e m_2\omega_2$, respectively, and the total system damping ratio ξ_T can be defined as the sum of the mechanical and electrical damping ratios, $\xi_m+\xi_e$. Therefore, Eq. (14) can be rewritten as

$$P_{E_{max}} = \frac{m_2\omega^3 X_1^2 \xi_e}{4(\xi_m+\xi_e)^2}, \quad (15)$$

where, ξ_e can be inferred from Eq. (8)

$$\xi_e = \frac{\kappa^2}{2m_2\omega_2(R_{load}+R_{coil})}. \quad (16)$$

However, not all the energy transduced into the electrical domain will actually be delivered into the load resistance. In the case of electromagnetic transduction, some of the power delivered to the electrical domain is lost within the coil [16-17]. Therefore, the actual power in the R_{load} is a function of the coil and load resistance and is calculated from

$$P_{L_{max}} = \frac{\kappa^2\omega^2 X_1^2}{8(\xi_m+\xi_e)^2(R_{load}+R_{coil})}\left(\frac{R_{load}}{R_{load}+R_{coil}}\right). \quad (17)$$

The maximum power delivered to the load resistance, derived by conversion of the micro-vibration energy of the cooler, can be calculated from Eq. (17).

2.2 Numerical Simulation

In this study, we proposed a complex system combined with a TMD-based electromagnetic energy harvester that performs the dual functions of harvesting electrical energy and isolating the micro-vibration. To analyze the feasibility of the proposed complex system, numerical simulations were performed using the model shown in Fig. 1. Table 1 summarizes the parameters used in the simulation. The values of m_1 and ω were based on the specifications of the cooler; ω_1 and ξ_1 were based on the measured values of the cooler assembly combined with a conventional vibration isolator [9]. The eigenfrequency ω_2 of the TMD-based energy harvester was adjusted to the main excitation frequency of the cooler to maximize the energy harvesting and micro-vibration isolation capabilities based on the general principle of the TMD. The value of the coil resistance R_{coil} was derived from the preliminary design. The mass of the permanent magnet m_2 was selected to have a minimum value within the limits of its effectiveness based on general TMD design specifications [18]. In practice, as the ratio of the TMD mass to the primary mass increases, the damping effect of the TMD on the vibration of the primary structure increases, but the total mass

Table 1. Parameter Values for Simulation

Parameter	Specification
m_1 (kg)	3.8
m_2 (kg)	0.1
ω (rad/s)	36 Hz × 2 π
ω_1 (rad/s)	8.2 Hz × 2 π
ω_2 (rad/s)	36 Hz × 2 π
ξ_1	0.01
ξ_m	0.01
R_{coil} (Ω)	300
κ (V/m/s)	18.97

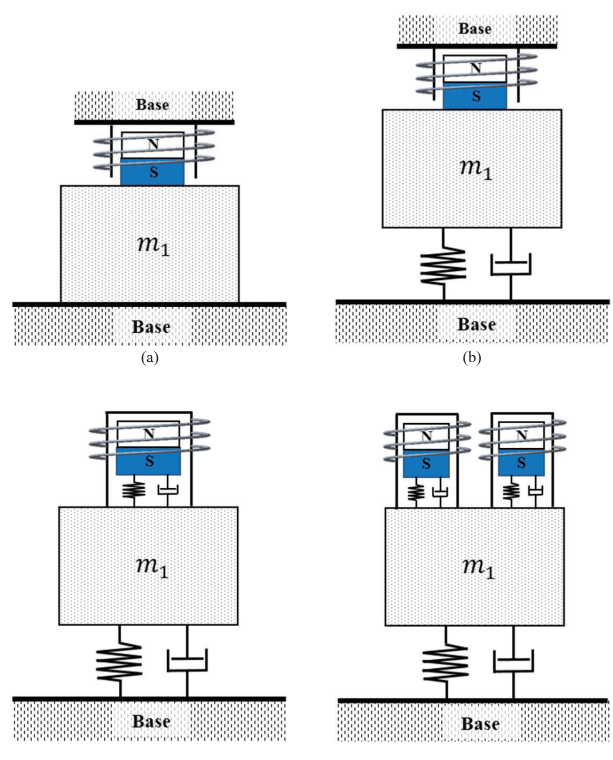

Fig. 2. Simulation cases to compare the performance of electrical power generation and micro-vibration isolation w.r.t the configuration of the complex system

of the system inevitably increases as well. Therefore, there are limitations in determining mass ratio $\mu=m_2/m_1$; the value of μ is generally selected between 0.01 and 0.25.

Figure 2 shows the simulation cases to numerically investigate the feasibility of the dual functions of harvesting electrical energy and isolating micro-vibration. The cases in Fig. 2 (a) and (b), in which the permanent magnet is rigidly fixed on the cooler, are the conditions under with and without the conventional passive vibration isolator, respectively. The cases in Fig. 2 (c) and (d), where the cooler is basically supported by the conventional vibration isolator with low stiffness, are combined with the TMD-based electromagnetic energy harvester proposed in this study; cases (c) and (d) apply one and two harvesters, respectively.

2.2.1 Micro-vibration Energy Harvesting

Figure 3 shows the maximum electrical power $P_{L_{max}}$ harvested from the micro-vibration of the cooler as a function of the load resistance R_{load} for the different analysis cases. In case (a), a permanent magnet is rigidly fixed on the cooler; because the displacement of the permanent magnet is infinitesimally small due to its rigidly fixed condition, the electrical power generated is also very small across the entire range of load resistances. In case (b), the cooler is supported by a vibration isolator; however, on accounts of the increased displacement of the cooler, a maximum generated power of 269.5 μW was observed at an impedance value of 1080 Ω. In case (c), where the TMD-based energy harvester was applied to the cooler assembly, and its eigenfrequency was tuned to the main excitation frequency of the cooler, the maximum generated electrical power was approximately 219.8 μW at an impedance value of 1080 Ω. This value is approximately 370 times higher than that of case (a), in accordance with the condition for maximum generated power in Eq. (17). However, the performance is slightly lower than that of case (b). This is because the energy $P_{T_{max}}$ absorbed into the dashpot elements of the harvester, composed of superposition of c_m and c_e, is distributed into the mechanical and electrical systems, unlike in case (b), where the entire absorbed energy is converted to the electrical domain. Therefore, the difference in the harvested energy between (b) and (c) represents the amount of energy dissipated into the mechanical damping c_m of the harvester. In case (d), two TMD-based harvesters were applied, and the maximum generated electrical power from each harvester was approximately 164 μW; thus, a combined power of 328 μW can be expected from two harvesters. This value is higher than that of case (c) by a factor of 1.49. This value, however, is not corresponded to the factor of 2 even if the number of two harvesters were applied to the cooler. This loss of power is presumed to be related to the decrease of the relative displacement z between the cooler and the permanent magnet of the harvester by increasing the number of mechanical damping elements of the harvester, which dissipate the absorbed energy to heat energy.

To investigate the micro-vibration isolation capability of the proposed complex system, numerical simulations were performed based on the parameters given in Table 1. The electrical damping performance for each case was derived from the optimal value of the load resistance necessary for maximum electrical energy delivered to R_{load} as shown in Fig. 3.

Figure 4 shows the time history of the force transmitted to the base for each case in an on-orbit environment.

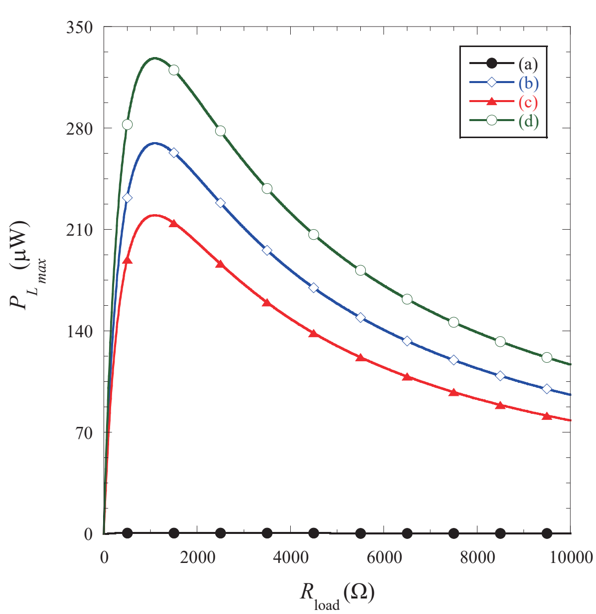

Fig. 3. Harvested power $P_{L_{max}}$ vs. the value of load resistance for each case

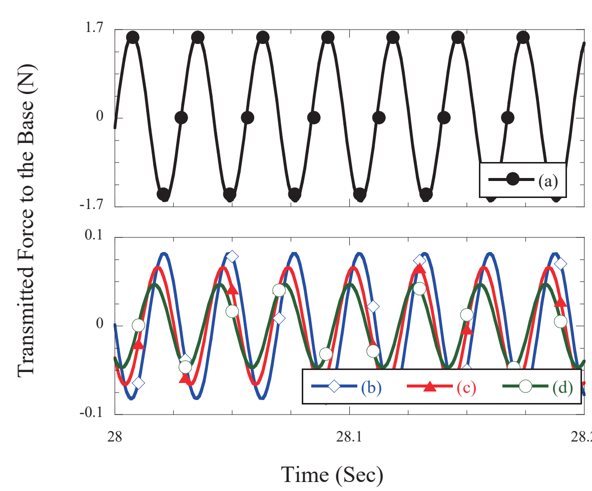

Fig. 4. Time history of the force transmitted to the base for each case

The maximum force of the cooler for case (a) was 1.5 N of sinusoidal amplification, which acts as a primary disturbance source that degrades the pointing stability of an observation satellite. However, the maximum force for case (b) was considerably reduced, to approximately 0.09 N, owing to the sufficient frequency decoupling between the main excitation frequency of the cooler (36 Hz) and the eigenfrequency of the cooler assembly (8.2 Hz). Furthermore, with respect to cases (c) and (d), it is apparent that the additional application of the TMD-based harvester on the cooler assembly enhances the micro-vibration isolation capability compared with case (b). This positive reinforcement of the micro-vibration isolation capability is caused by an occurrence of additional energy dissipation into the mechanical damping of the TMD-based energy harvester. As expected, case (d), which has two TMD-based energy harvesters, exhibits the highest micro-vibration isolation capability among the simulation cases, showing a maximum force level of 0.046 N, effectively reducing the micro-vibration disturbance from the cooler by a factor of 32 relative to case (a).

From the above simulation results, we can conclude that the application of the TMD-based electromagnetic energy harvester on the cooler supported by the conventional passive vibration isolator can achieve the dual objectives of both electrical energy generation and enhancement of micro-vibration isolation capability. In addition, it is apparent that case (d), which has two harvesters, exhibits the greatest capability for both energy harvesting and micro-vibration isolation. However, as a baseline to check the robustness of the complex system, case (c), in which the cooler with a passive vibration isolator is integrated with a single TMD-based energy harvester, was selected in this study to analyze the micro-vibration isolation and energy generation capabilities when the mechanical properties of the complex system such as stiffness k_1 and k_2 are varied according to different values of the mechanical damping ratio ξ_m of the harvester. In this numerical estimation, the electrical damping ratio ξ_e for each value of ξ_m was derived from Eq. 16 by using the value of load resistance R_load, which is determined for the condition at which maximum power $P_{L_{max}}$ occurred.

Figure 5 shows the simulation result when the stiffness k_1 of the complex system is varied with various values of damping ratio ξ_m, while keeping all other parameters unchanged. And the increase in k_1 indicates an increase of the eigenfrequency f_1 of the cooler assembly. Considering the results in terms of energy generation capability, the harvested electrical power in all cases gradually decreased as the eigenfrequency of the cooler assembly approached 20 Hz, then sharply increased with a further increase to 36 Hz, which corresponds to the main excitation frequency of the cooler. These phenomena are related to the total amount of energy absorbed into the dashpot element of the harvester, which heavily depends on the relative displacements between the cooler and the permanent magnet. The micro-vibration energy of the cooler absorbed in the harvester is distributed again into the mechanically dissipated energy and electrically harvested energy. The mechanically dissipated energy is converted to heat, and contributes to the enhancement of the micro-vibration isolation performance. The portion of electrically harvested energy from the total absorbed energy is in the form of useable electricity, except for the energy dissipated in coil resistance; it can be stored into a rechargeable battery or super-capacitor. At frequencies below the cut-off frequency for the excitation frequency of the cooler, which is approximately 25 Hz, the power generated at the lowest value of ξ_m=0.01 is apparently higher than that of the other cases. For instance, it is observed that with the damping ratio ξ_m=0.01, the level of harvested power is approximately 800 times higher than with ξ_m=1. This is due to the considerable relative displacement between the cooler and the permanent magnet of the harvester, whose amplitude increases considerably at the lower value of the damping ratio. At the condition of the frequency coupling around 36 Hz, the relative displacement between the cooler and the permanent magnet of the harvester increases greatly, resulting in a dramatic increase of the maximum electrical power in accordance with the criteria for the maximum electrical power from Eq. (17). However, the energy generation trend is reversed compared to lower frequency ranges, because the total amount of energy absorbed in the TMD-based energy harvester also heavily depends on the total amount

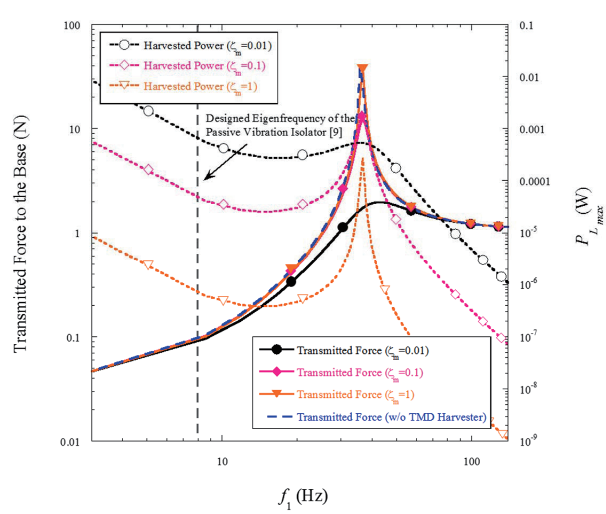

Fig. 5. Transmitted force and harvested power $P_{L_{max}}$ vs. the value of damping ratio ξ_m, for variable stiffness values of k_1

of damping, which is composed of a superposition of the electrical and mechanical systems. When z exhibits a large amplitude due to the eigenfrequency of the cooler assembly being matched with the main excitation frequency of the cooler, the larger the value of the electrical damping, the more the energy generation performance can be achieved by increasing the energy absorbed into the electrical system. In contrast, when the eigenfrequency of the cooler assembly is higher than 50 Hz, the energy generation performance rapidly decreases. This is mainly due to a decrease of the displacement x_1 of the cooler, because the cooler is supported by a larger stiffness value k_1. According to the observations in the simulation results of Fig. 5, from the micro-vibration isolation point of view, all cases show significant micro-vibration isolation capability in the low frequency range by sufficiently separating the eigenfrequency of the cooler assembly from the excitation frequency of the cooler. In addition, the application of the TMD-based energy harvester on the cooler assembly supported by the low-stiffness isolator contributes to the reduction of the micro-vibration levels transferred to the base. When the eigenfrequency of the cooler assembly is calibrated at around 36 Hz, which corresponds to the main excitation frequency of the cooler, the electrical power generated in all cases was dramatically increased by the frequency coupling effect, but the force transmitted to the base was also sharply amplified. This amplification is undesirable, as it would seriously degrade the image quality of the observation satellite. Therefore, to obtain high-performance micro-vibration isolation from the complex system, the eigenfrequency of the cooler assembly should be sufficiently separated from the main excitation frequency of the cooler but at a cost of decreasing the energy generation performance as far as the value of 18 Hz from 36 Hz. This degradation of the energy harvesting performance can be compensated by setting the eigenfrequency of the cooler assembly as low as possible to increase the relative displacement z. Using this approach, a TMD-based energy harvester combined with a conventional vibration isolator, which supports the cooler with a low stiffness at 8.2 Hz, can achieve the dual objectives of electrical power generation and improved micro-vibration isolation capability as envisioned by this study.

Figure 6 shows simulation results in which the stiffness k_2 of the TMD-based energy harvester was varied by a factor of $k_2/k_{2_{\mathrm{nominal}}}$, while keeping all other parameters unchanged. This simulation was performed to investigate the effects on the performance of micro-vibration isolation and energy harvesting when the stiffness of the harvester is varied, which can result from deteriorations and inadvertent changes of the mechanical properties of the harvester from unexpected problems during launch and on-orbit environments. The results of the simulation were then compared for various values of damping ratio ξ_m of the harvester. In the simulation results, when the value of the damping ratio ξ_m increases, varying the stiffness has relatively little effect on the micro-vibration isolation performance. And those conditions with larger values of the damping ratio exhibit relatively higher isolation capability compared with that of the situation without a TMD-based energy harvester, which is plotted with a dotted blue line, across the entire range. Under the condition with a low damping ratio, i.e., ξ_m=0.01, the micro-vibration isolation capability is the highest at a factor of $k_2/k_{2_{\mathrm{nominal}}}$=1, which is the value intended in this study to maximize the effect of the micro-vibration isolation and the energy generation performances. The performance, however, is quite sensitive to stiffness variation. The level of force transmitted to the base increases sharply even for a slight variation of the stiffness k_2, exceeding even the situation without the TMD-based harvester. Nevertheless, the complex system can achieve significant micro-vibration isolation capability on account of the existence of the passive vibration isolator supporting the cooler with low stiffness, while simultaneously harvesting the electrical energy from the micro-vibration energy of the cooler. Furthermore, when the value of the damping ratio is relatively low, i.e., ξ_m=0.01, the power generated is the highest across the entire range compared to cases with higher damping values. Otherwise, it is apparent that the harvested energy across the entire range significantly decreases as the value of ξ_m increases. In terms of physical aspect, this is caused by the sticking phenomenon of the permanent magnet by the relatively larger value of the harvester's damping ratio in line with a much lower mass ratio of the permanent magnet relative

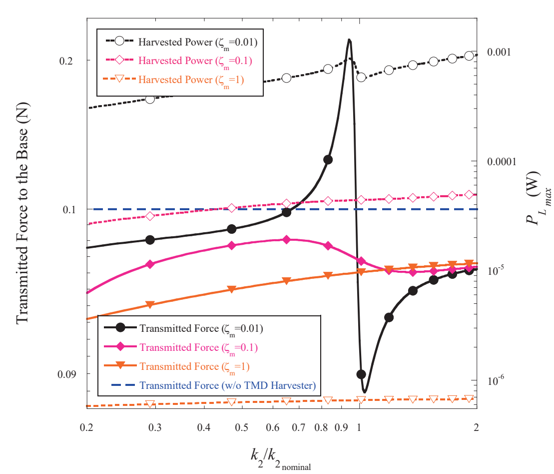

Fig. 6. Effect of stiffness variation of the TMD-based energy harvester k_2 on the performance of electrical energy generation $P_{L_{max}}$ and micro-vibration isolation

to the cooler. In practice, this drawback of the sensitivity of the harvester performance to the damping variation can be lessened by increasing the mass ratio of the permanent magnet. However, this solution also results in increasing the total mass of the system, which is not favorable in satellite systems owing to the strict mass requirement. However, it is no longer of concern for the micro-vibration isolation performance under the assumption that the permanent magnet of the harvester is stuck by a higher damping value, which is the worst unexpected condition in on-orbit environment, since the cooler is supported mainly by the passive vibration isolator with a low stiffness as a baseline.

2.2.2 Launch Vibration Energy Harvesting

Satellites experience several kinds of dynamic environments during flight, such as sinusoidal vibration, random vibration, and pyrotechnic shock [19]. Of special concern are random mechanical vibrations that occur at the maximum dynamic pressure phase by acoustic loads acting on the launch vehicle structures and are transferred to the satellites and the launch vehicle interface adaptor. In general, random vibration has considerable vibration energy and can threaten the structural safety of the payloads and electronic components of the satellite [20]. Practically, launch vibration environment is much more severe than on-orbit micro-vibration environment. However, it also can be regarded as a one of potential energy source for the energy harvesting technology. Therefore, in this study, we investigated the energy generation capability for each case, as shown in Fig. 2, under the assumption that the complex system is subjected to a white noise random base acceleration, which has a flat power spectral density in the frequency domain between 1 to 2000 Hz with 8.5 G_{rms}. For the simulation, the same mechanical properties were applied as for the conventional cooler isolator. In practice, this isolator showed significant launch vibration attenuation performance in the test [9].

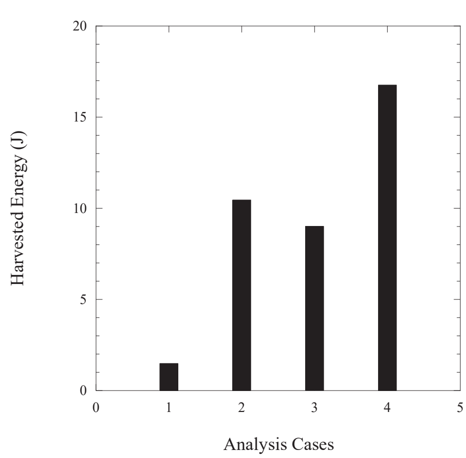

Fig. 7. Harvested energy on the optimal load resistance, R_{load}=1080, under random excitations for each case

Figure 7 shows the output energy on the optimal load resistance R_{load}=1080, which is derived from Fig. 3, according to each case under white noise excitations. In this simulation, the displacement response of the cooler assembly is limited to just ±3 mm owing to the presence of a mechanical limiter in the isolator design [9] to make it stay within the allowable deflection range in a launch environment, and the TMD-based energy harvester is also assumed to be limited in ±3 mm. From the simulation results, case (d) shows the highest energy generation performance. This tendency is the similar to the results shown in Fig. 3 obtained from the micro-vibration environment, because of the same reasons. In this manner, the harvested energy is approximately 11 times higher than that of case (a), in which the cooler is rigidly fixed to the base. Meanwhile, this greatest energy generation capability in case (d) cannot be achieved using the conventional cooler isolator that requires a holding-and-release mechanism as referenced in previous studies [4-7] because their applications need to rigidly fix the cooler, as shown in Fig. 2 (a), during launch to guarantee the structural safety of both the cooler and the isolator itself. In addition, in the event of an unexpected problem in activating the holding-and-release mechanism in on-orbit, the cooler would inevitably maintain its rigidly fixed condition, resulting in considerable performance degradation of both energy harvesting and micro-vibration isolation as shown in Fig. 5. These are the main reasons of using the cooler isolation system [9] that can be used in both launch and on-orbit micro-vibration environments without applying an additional holding-and-release mechanism.

3. Conclusions

This study focused on the applicability of energy harvesting technology to satellite micro-vibration and proposed an electromagnetic energy harvester that converts micro-vibration energy to electrical energy. The harvester design combined a TMD with a conventional passive-vibration isolator to maximize the efficiency of electrical power generation. The combination was also effective in enhancing micro-vibration isolation. Its ability to achieve the dual objectives of electrical energy harvesting and micro-vibration isolation was demonstrated through a numerical simulation. In addition, to evaluate the robustness of the complex system, the effects of varying dynamic characteristics, such as stiffness and damping ratio, on its performance were investigated. The robustness evaluation results demonstrate that varying the

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dynamic characteristics of the harvester has relatively little effect on the micro-vibration isolation performance because of the passive vibration isolator supporting the cooler with low stiffness. Despite the fact that the amount of electrical power generated in an on-orbit environment is infinitesimal, considering that the cooler can be used as a power source at all times, its potential is promising if the electrical energy is used to charge rechargeable batteries or super-capacitors for MEMS-based low-power devices. In addition, the energy generation capability in a launch environment was investigated.

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