DEVELOPMENT OF TUNABLE DAMPER DEVICES FOR VIBRATION REDUCTION IN SATELLITE EQUIPMENT MOUNTED ON HONEYCOMB PANELS

Nicolas SOUIL (1), Florent MASSON (1), Jean-Yves DISSON (2)

(1) METRAVIB Enginnering, 200 Chem. des Ormeaux, 69760 Limonest FRANCE, nicolas.souil@acoem.com

(2) METRAVIB Enginnering, 200 Chem. des Ormeaux, 69760 Limonest FRANCE, florent.masson@acoem.com

ABSTRACT

In the context of satellite equipment mounted on honeycomb panels, where mitigating vibrations from shocks and random environments is crucial, this paper presents a tunable damper prototype based on elastomer material, developed with the French CNES Structural Department. Using experimental and computational works, the study explored stiffness variation through spherical and conical elastomer interfaces, informing the design of a prototype insert that allows better control over stiffness.

Performance testing showed significant stiffness variation, with elastomer pre-strain from 5% to 25%, resulting in a stiffness change by a factor of about 10. This concept promises variable stiffness isolators, enabling versatile material protection to reduce vibrations in satellite equipment.

STATE OF THE ART

Challenges Encountered

The primary goal of this literature review ([1], [2], [3] and [4]) is to identify existing adjustable passive solutions or applicable principles for inserts, based on industrial sources like supplier catalogs and patents. The search encountered two main difficulties. First, many solutions rely on the Tuned Mass Damper principle, which involves a mass-spring-damper system in series, offering high damping over a narrow frequency band. Second, the specific application of inserts, which involves threaded embeddings within another structure, is rare and predominantly found in aerospace industries.

Principles

Despite these challenges, several principles of variable dampers were identified. One principle involves variability through geometric non-linearity, such as in the design of a tunable stiffness composite leg for dynamic locomotion, where stiffness is varied by altering geometric boundary conditions. Another principle is contact activation, as seen in the adjustable stiffness jack spring actuator, which achieves variable stiffness through contact-induced changes in spring length. Impact dampers, like the tunable apparatus for adjusting effective performance of particle impact dampers, offer modular interfaces and variable performance based on the type and number of encapsulated tungsten balls.

Additionally, modular inserts in commercial versions, provide kits with adjustable threadings and stiffness elements.

These inserts offer modularity through kit assembly, resulting in step-wise variability rather than continuous adjustment.

Synthesis

This paragraph summarizes the advantages and disadvantages observed in this study of existing solutions and their application to damped inserts:

Advantages:

- Adjustable device: Single product, versatility, and unit cost.
- Kit/Range: Multiple products, ease of implementation, production/storage.

Disadvantages:

- Adjustable device: Design complexity.
- Kit/Range: Design volume (multiple components).

Non-linearity Technology:

- Contact Activation: Linear mechanical systems, step adjustments, complexity with more than 2 or 3 settings.
- Geometric Variation: Continuous variation, simplicity in scaling variations, implementation demands, and stability risks in use.

Separate Stiffness/Damper Functions:

- More adjustment possibilities, function interaction, and space requirements.

Unified Stiffness/Damper (Damping Material):

- Compactness, fewer adjustment possibilities.

This synthesis highlights the versatility and complexity of various adjustable passive solutions for insert systems, balancing ease of use and implementation challenges.

CONCEPT SELECTION FOR TUNABLE PASSIVE DAMPER

Hypotheses

Based on the study of existing solutions, a concept for a tunable passive damper was determined. This concept is based on the following hypotheses:

- The combined function of stiffness and damping is ensured by an elastomer material, similar to those already qualified for the space sector.
- The insert is adjustable through geometric non-linearity.

Stiffness Variation

The stiffness variation is achieved by altering the contact surface of an elastomer material element. The stiffness of an elastomer element depends on the elastic modulus of the material and its shape factor (S_F [-]), defined by the ratio of loaded surface to free surface (Bulge Area) as described in Eq(1).

$$S_F = \frac{Loaded\ Area}{Bulge\ Area} \qquad (1)$$

The compression modulus E_c [GPa] or apparent stiffness is then defined using this ratio.

$$E_C = 3\,G(1 + {S_F}^2) \qquad (2)$$

with G = shear Modulus [GPa]

The compression modulus or apparent stiffness is then defined by this ratio. To achieve this variation, an interface type of sphere/plane or sphere/cone was studied using ANSYS (Finite Element Method) simulations.
The ANSYS simulations showed that depending on the cone angle, the free or constrained surfaces for a given compression vary, thus differing the access to shape factor variation. The conical interface allows for a more gradual surface variation and more homogeneous stress distribution, making it preferable for the application of adjustable inserts.

Concept Outline

A preliminary concept was developed, consisting of 12 spheres, each 1 cm in diameter, housed in 45° conical cavities. The adjustment is made by tightening a screw with a conical interface that compresses the spheres. The stiffness of the insert was studied based on an elastomer material with a shear modulus of G = 5.5 x 10^6 Pa.
Finite element calculations were performed to estimate the stiffness evolution according to the static compression rate, which is the percentage of sphere compression relative to its free size. The model of Figure 1considers half a sphere in front of a cavity, with the screw interface being symmetrical.

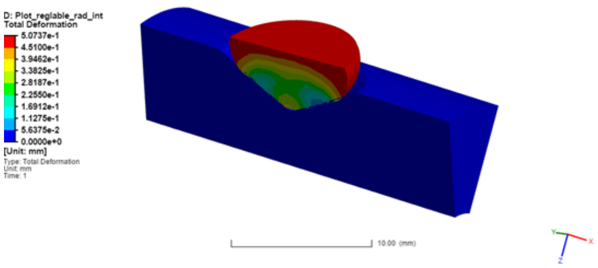


Figure 1: Finite element modeling of sphere/cone interface

The stiffness calculation in the direction of compression or shear clearly showed a variation in stiffness with compression. For compression rates ranging from 1% to 10%, the compression stiffness varied by a factor of 3, and the shear stiffness varied by a factor of more than 10. The axial stiffness of the insert containing 12 spheres can be evaluated to vary from 6 x 10^5 N/m to 6 x 10^6 N/m. This 10-fold stiffness variation validates the principle of variability by compression of an elastomer on a conical interface. This concept can thus meet the expected variations. However, the shape of this interface must be re-evaluated to obtain equal variation in both shear and compression planes, which is characterized experimentally later.

PRELIMINARY TESTS

Methodology

To study the potential of an interface that induces variation in apparent stiffness based on compression, a test campaign was implemented.
For rapid and efficient testing, off-the-shelf materials and geometries were selected for the elastomer spheres, including demonstration elastomers balls and subcontracted SMAC [5] balls with a diameter of 13.5 mm.
Three materials were tested:

- Type A SMACSIL® MB1150-01 (65 ±5 ShA);
- Type B SMACSIL® MB1197-01 (63 ±5 ShA);
- Type C SMACSIL® MB1191-01 (58 ±5 ShA).

These three types of spheres were tested on three types of interfaces in a Dynamic Mechanical Analyzer (DMA) system (Figure 2): spherical interfaces with 2 different diameters, and conical interfaces with 2 different angles. These interfaces were evaluated in two setups to measure stiffness in both tension-compression and shear modes as it can be observed in Figure 3 and 4.

Figure 2: Dynamic Mechanical Analyser

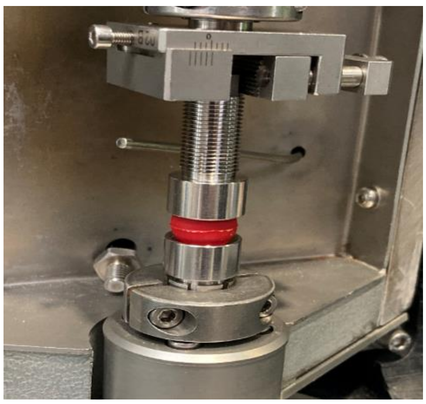


Figure 3: Compression Test Setup

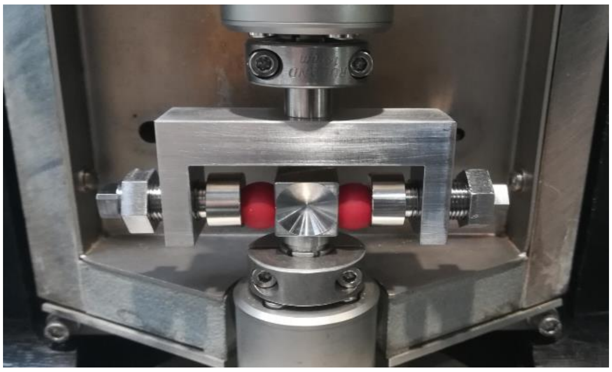


Figure 4: Shear Test Setup

Preliminary Tests

Initial tests were conducted to assess the basic behavior and variations before embarking on more detailed parametric studies.

In Test 1, the variation by compression rate was explored by varying compression from 0 to 20%. Sphere A with a spherical interface was subjected to an excitation of 10 Hz with a 1 µm amplitude. The results demonstrated a significant stiffness variation factor greater than 10, with stiffness values ranging from 1 x 10^5 to 2 x 10^6 N/m for compression rates of 0 to 10%. The damping ratio observed ranged from 0.11 to 0.15 at small dynamic deflection amplitudes.

In Test 2, the variation by material was evaluated by examining how different materials influence stiffness under compression ranging from 0 to 10%. Spheres A, B, and C with a conical interface were excited at 10 Hz with a 1 µm amplitude. The apparent stiffness varied by a factor of 2 depending on the material, with greater differences observed for higher compression levels.

Parametric Tests

Following the preliminary tests, detailed parametric studies were conducted to systematically analyze the effects of different interfaces and materials on stiffness.

In Test 3 (Figure 5), the variation by interface in compression was assessed by using Sphere A with four different interfaces: 25° and 35° cones, and 18 mm and 20 mm spheres. The stiffness ratio between 2% and 10% static compression varied from 2.4 to 3.6, indicating the impact of interface geometry on stiffness. The damping ratio was minimally affected.

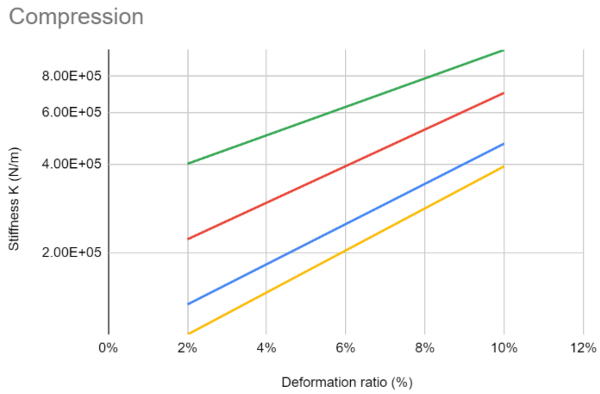


Figure 5: Compression Test Results

In Test 4 (Figure 6), the variation by interface in shear was evaluated using Sphere A with four different interfaces. This test showed that the stiffness ratio between 2% and 10% static compression varied from 2.3 to 6.8, highlighting significant differences based on interface type. The damping ratio remained relatively stable.

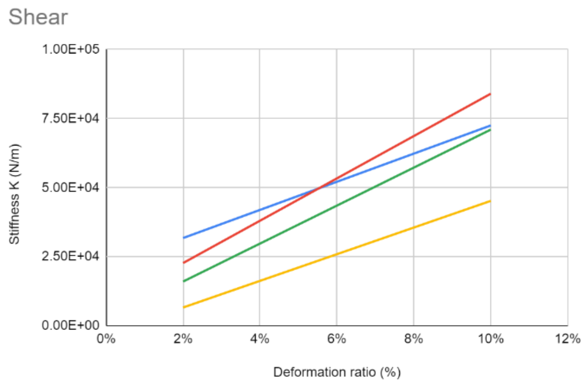


Figure 6: Shear Test Results

In Test 5 (Figure 7), the variation by material in compression was compared using Spheres A, B, and C with a conical interface. The results showed a stiffness ratio of 2 between materials A and C, emphasizing the influence of material properties on stiffness variation.

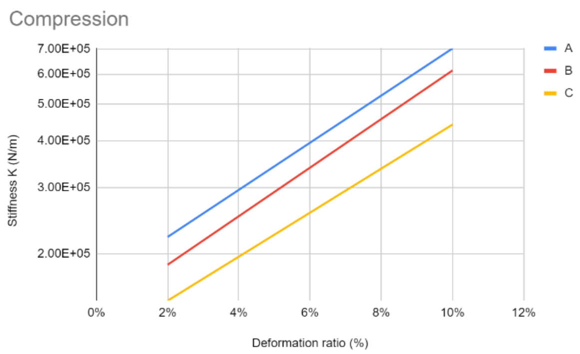


Figure 7: Compression Test Results

Synthesis

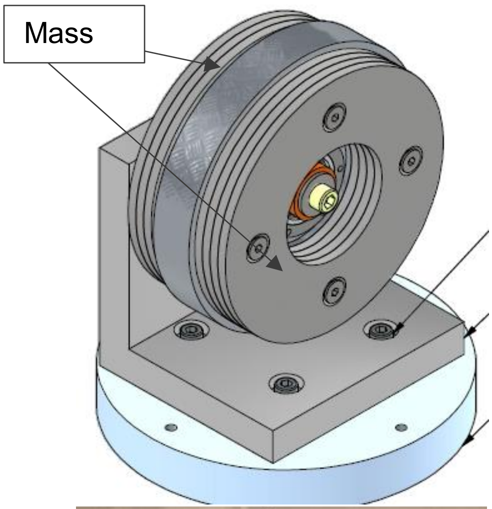
These tests validated that an interface allowed the apparent stiffness to vary by a factor up to 3.1 in compression and 3.7 in shear for 0 to 10% compression ratio, resulting in equal variation according to excitation directions.
At 10% compression, stiffness was observed to be 7.0 x 10^5 N/m in compression and 8.4 x 10^4 N/m in shear.
This factor of 10 difference between compression and shear stiffness had not been identified in the preliminary design phase. The interpretation of this discrepancy is attributed to the difference in implementation and deformation between the calculations and DMA measurements: the shear calculation involved a half-sphere with imposed translation, whereas the DMA tests involved sample rotation/rolling of the sphere, inducing apparent flexibility.

PROOF OF CONCET (POC) DEFINITION

A proof-of-concept insert was developed to validate the experimental concept using proprietary technology. The revised geometry offers higher rigidity and a more compact design compared to the initial pre-design using spherical elastomer parts. It was observed that minimizing available volume results in lower rigidity, while a fully filled volume provides maximum rigidity.

Test Bench Setup

A test bench was implemented to perform sinusoidal sweep tests under varying loads (suspended masses ranging from 2 to 3.5 kg in 0.5 kg increments using washers) in both axial and radial directions. The variable deformation pre-load was applied using a set of washers corresponding to 5% compression increments, varying from 5% to 30%. Each washer imposed a 5% compression increment. This system allowed for the independent constraining of the components from the tightening torque.
Measurement equipment included accelerometers for test mass measurement and input system excitation, an exciter, and an acquisition front end. Sinusoidal excitation tests were performed with a frequency sweep from 0 to 500 Hz and constant displacement input commands (0.1, 0.05 & 0.025 mm) of varying magnitudes, with acceleration limited over the frequency band.
Tests were conducted with suspended masses of 2 kg and 3.5 kg, and compression rates of 5%, 10%, 15%, 20%, and 25% in the axial direction, and 5%, 10%, and 15% in the radial direction as pictured in Figure 8 and Figure 9.



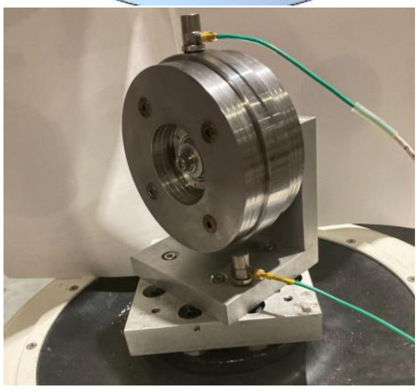
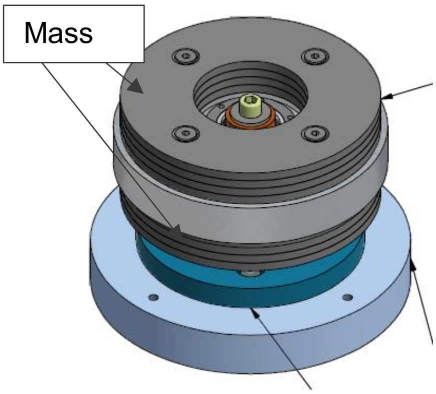


Figure 8: Test Bench Shear Setup



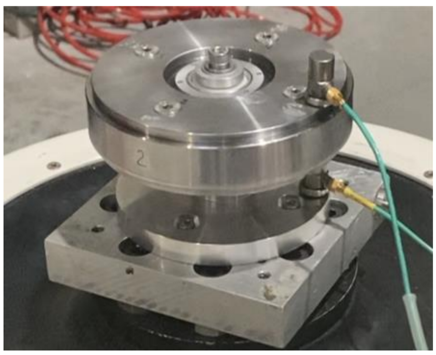


Figure 10: Test Bench Compression Setup

Results

The results were interpreted in two ways, considering a nonlinear behavior observed and mechanical model of test bench : calculation from the maximum frequency f_m and calculation from the cutoff frequency f_c. For each test, dynamic stiffness was estimated from these frequencies as described in Figure 11.

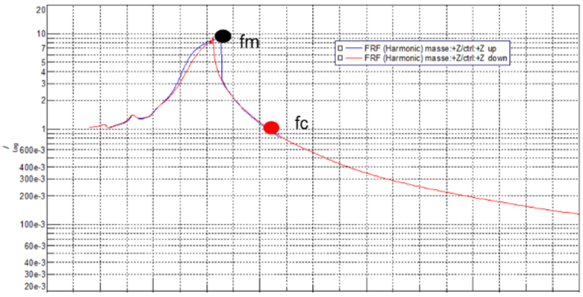


Figure 11: Test Bench Results Interpretation

$$2\pi f_m = \sqrt{\frac{k}{m}} \qquad k = m(2\pi f_m)^2 \qquad (3)$$

$$f_0 = \frac{f_c}{\sqrt{2}} \qquad k = m(2\pi f_0)^2 \qquad (4)$$

An example series of tests with a 2 kg mass in the axial direction showed the evolution of frequencies. The results for a 2 kg mass and specified displacement indicated that the apparent resonance frequency ratio between axial and radial directions varied from 1.1 at 5% compression, to 1.2 at 10%, and 1.3 at 15%.

Tests for 2 kg and 3.5 kg in the axial direction showed in Figure 12 and Figure 13 a factor of 3 variation in stiffness between 5% and 15% compression, and a factor of 11 between 5% and 25% compression.

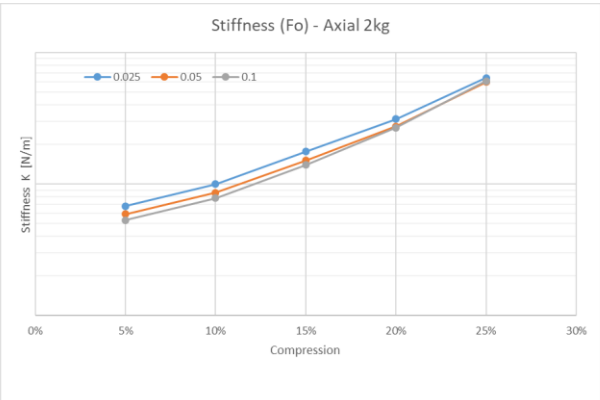


Figure 12: Axial Stiffness (Fo) Results 2 kg

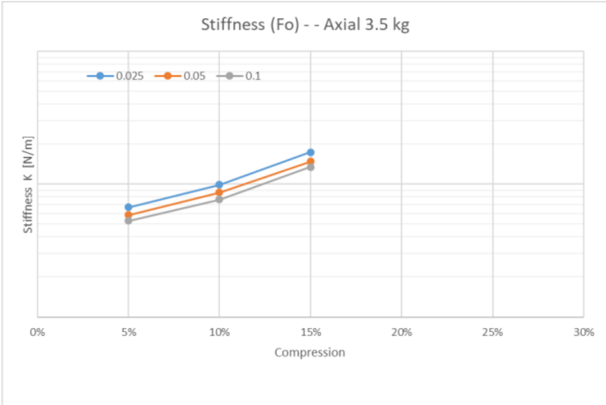


Figure 13: Axial Stiffness (Fo) Results 3.5 kg

Tests for 2 kg and 3.5 kg in the radial direction indicated in Figure 13 an Figure 14 a factor of 2 to 3 variation in stiffness between 5% and 15% compression.

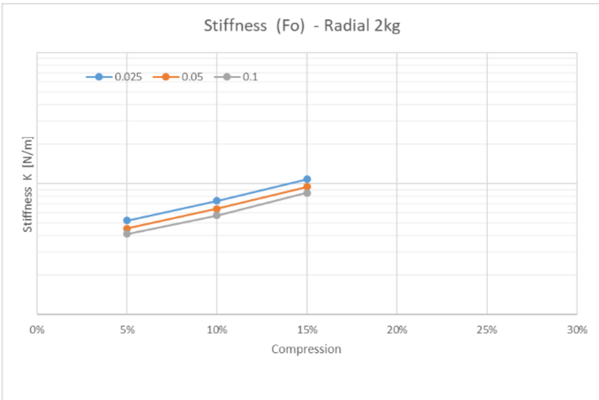


Figure 13: Radial Stiffness (Fo) Results 2 kg

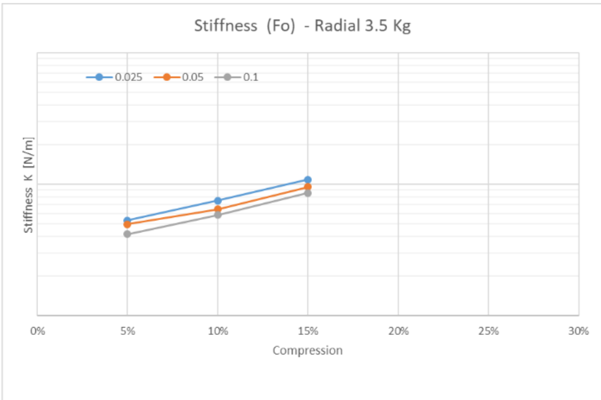


Figure 13: Radial Stiffness (Fo) Results 3.5 kg

The tests validated the concept of adjustable inserts with the following findings:

- Equi-frequency within 30% tolerance, which could be optimized by refining the geometry;
- Variable stiffness with a factor of 3 between 5% and 15% compression, and the potential for a factor greater than 10 with high compression rates;
- Observed stiffness values were significantly lower than the initial design requirements, indicating the need for further optimization.

CONCLUSION AND PERSPECTIVES

Extensive testing on a custom test bench revealed the ability of the adjustable insert to achieve significant stiffness variation. In both axial and radial directions, stiffness ratios were observed to change by a factor of up to 11 under varying compression rates, confirming the effectiveness of the design in adapting to different loading conditions. Frequency response analysis showed that the concept maintained equi-frequency within acceptable tolerances, which can be further optimized with geometric refinements.
The adjustable passive damper concept has proven to be a viable solution for vibration mitigation in satellite equipment. The experimental results validated the approach of using elastomer materials within a compact, adjustable design.

Future work will focus on optimizing the geometry to enhance equi-frequency performance and increase overall stiffness values. Advanced modeling and simulation techniques will be employed to refine the design and predict performance under a broader range of conditions. Additionally, exploring alternative elastomer materials and configurations could yield further improvements in stiffness variability and damping efficiency. The ultimate goal is to develop a highly reliable, adjustable damper that can be seamlessly integrated into various satellite platforms, ensuring superior vibration mitigation and extending the operational lifespan of satellite equipment.

REFERENCES

1. US8789630B2, DESIGN OF A TUNABLE STIFFNESS COMPOSITE LEG FOR DYNAMIC LOCOMOTION, Kevin C. Galloway.
2. US2007/0267791 A1, ADJUSTABLE STIFFNESS JACK SPRING ACTUATOR
3. US 10 , 041 , 558 B1 TUNABLE APPARATUS FOR ADJUSTING EFFECTIVE PERFORMANCE OF PARTICLE IMPACT DAMPER
4. US 8,777,193 B2, VIBRATION ISOLATION FASTENER INSERT
5. SMAC, 66 Impasse Edouard Branly - ZI TOULON EST - BP 119 - TOULON CEDEX 09, https://smac-sas.com/