Passive Vibration Isolation by Compliant Mechanism Using Topology Optimization

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Compliant mechanisms have been designed for various types of applications to transmit desired forces and motions. In this paper, we explore an application of compliant mechanisms for passive vibration isolation systems. For this, a compliant isolator is used to cancel undesired disturbances, resulting in attenuated output amplitude. A compliant mechanism is equipped with an isolator, while a compliant mechanism also functions as a transmission of force and controls the amount of displacement that is transmitted from it. It can be used as passive vibration isolation. Here, by introducing compliance into the connection, the transmission of applied forces is reduced at some frequencies at the expense of increasing transmission at other frequencies. While transmitted force is the key parameter from the receiver’s perspective, motion at the isolated machine is uninteresting. The force transmissibility is numerically identical to the motion transmissibility. The structural optimization approach is focused on the determination of the topology, shape, and size of the mechanism. The building blocks are used to optimize a structure for force transmission. The flexible building blocks method is used for the optimal design of compliant mechanisms. This approach is used to establish the actuator model of the block and its validation by commercial finite element software. A library of compliant elements is proposed in FlexIn. These blocks are limited in number, and the basis is composed of 36 elements. The force transmitted to the rigid foundation through the isolator is reduced in order to avoid the transmission of vibration to other machines. The preliminary results of FEA from ANSYS demonstrate that compliant mechanism can be effectively used to reduce the amount of force transmitted to the surface.

1. INTRODUCTION

1.1. Compliant Mechanism

Compliant mechanism is the mechanism that relies on its own elastic deformation to transfer or transform motion or force. Common compliant mechanisms function under the application of force at certain location (input) and generate a desired force or deflection at another location (output). Compliant mechanism is designed for passive vibration isolation system (PVIS). In this system, the existing element (i.e., the coil spring isolator) is replaced with the new designed element in order to reduce the amount of force transmitted to the ground or to the foundation of the machine, which tends continuously to damage the base over a longer period of time. This happens because the initial starting machine gives rise to huge amplitude up to the frequency ratio one. This region is identified as the amplification region, and during this, a large amount of vibration force is transmitted. By reducing the force transmit-
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1.2. Topology Optimization

Homogenization based topology optimization is the basis for the design technique proposed in this research.2–4 Topology and size optimization methods are used to design compliant mechanisms, and the design procedure followed is based on the size optimization of the beam-element abstraction derived from the continuum topology solution.5 The topology optimization problem is formulated as a problem of finding the optimal distribution of materials in an extended fixed domain where some structural cost function is maximized.6,7 This work of topology optimization is carried out using ANSYS,8,9 by this, the optimum material distribution is obtained.10 Then the structural optimization11 is done using flexible building blocks12 designed by FlexIn Corporation. These elements are arranged in such a manner that to reduce the amount of force transmitted the trial and approximation method is used. Stability analysis in compliant mechanism design is of utmost importance. From a practical point of view, a CM that is unstable is of no significance. A stable system is defined as a system with a bounded system response. That is, if the system is subjected to a bounded input or disturbance, and the response is bounded in magnitude, the system is said to be stable.

1.3. Vibration Isolation

Vibrations are produced in machines that have unbalanced masses. These vibrations will be transmitted to the foundation upon which the machines are installed (see Fig. 1). This is usually undesirable. To diminish the transmitted forces, machines are usually mounted on springs or dampers as seen in Fig. 2, or on some other vibration isolation material. Vibration isolation reduces the level of vibration transmitted to or from a machine, building, or structure from another source.

1.4. Problem Formulation

Compliant mechanism is the focus of active research because of the stability, robustness, and ease of manufacturing endowed by their unitized construction. In this paper, we explore an application of compliant mechanism for a vibration isolation system with a rigid foundation. The structural optimization approach is focused on the determination of the topology, shape, and size of the mechanism. The building blocks are used to optimize a structure for force transmission.

1.5. Methodology

The displacement amplitude of the coil spring isolator is obtained for varying frequency ratios \( R \) (1.5–5). For the corresponding displacement amplitude, the force transmitted to the rigid foundation is determined. Then by using topology optimization and flexible building blocks, the vibration isolator is designed. The design is subjected to harmonic analysis using ANSYS software. For this design displacement amplitude and for the corresponding amplitude, the force transmission is calculated. For the coil spring isolator and compliant mechanism, isolation efficiency is determined and compared.

2. DESIGN OF COMPLIANT MECHANISM USING TOPOLOGY OPTIMIZATION AND BUILDING BLOCKS FOR VIBRATION ISOLATION

Topological optimization is a form of “shape” optimization sometimes referred to as “layout” optimization. The goal of topological optimization is to minimize/maximize the criteria.
selected (minimize the energy of structural compliance, maximize the fundamental natural frequency, etc.), while satisfying the constraints specified (volume reduction, etc.).

The topology optimization predicts the optimal distribution of the material in the design domain. It is very promising for the systematic design of compliant mechanism because topological design is automated by the given prescribed boundary conditions.

The problem is defined for linear-elastic analysis. Then it is defined for material properties (Young’s modulus, Poisson’s ratio, and possibly the material density). Then the two types of element 2D planes for topological optimizations to generate a finite element model are selected. Load and boundary conditions for a single load case linear structural static analysis are shown in Fig. 3.

Figure 4 and Fig. 5 illustrate the volume constraints for the specific load of 85 kN, and the force transfer path is identified for structural size of 500 mm width and 165 mm height. The optimized path for the transfer of the maximum force is obtained using topology optimization.

2.1. Topology Optimization for Vibration Isolator Using FEA

In this example the boundary condition specified as all the corners of the design domain is fixed, and a point load is applied at the middle of the bottom face. The material property and the design variable and the domain dimension are given below in Table 1.

2.2. Compliant Building Blocks

The optimal design of compliant mechanisms made of an assembly of basic building blocks is chosen in a given library. A library of passive compliant elements is proposed in FlexIn. These blocks are limited in number: the basis is composed of 36 elements as shown in Fig. 6.

2.3. Creation of Building Blocks Library

A library of passive compliant elements made of piezoelectric beams has been implemented in FlexIn. As for passive blocks, a block stiffness matrix is constituted by the assembly of beams stiffness matrices in the global coordinate system. The blocks present some various topologies. Their advantage is that they can furnish multiple coupled degrees of freedom (Dofs), thus generating more complex movements with only one building block.

They are sufficient to constitute a high variety of topologies, and it has been verified that they can describe many existing compliant structures of the literature. Moreover, the block feasibility related to fabrication process constraints can also be taken into account at this stage, which is not the case for classical beam-based optimization approaches. From the library of passive compliant building blocks, the structure formed for the

<table>
<thead>
<tr>
<th>Design domain</th>
<th>500 mm × 305 mm × 165 mm</th>
</tr>
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<tbody>
<tr>
<td>Young’s modulus</td>
<td>200 × 10^9 N/m²</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.29</td>
</tr>
<tr>
<td>Input force</td>
<td>85 kN</td>
</tr>
<tr>
<td>Upper limit of design variable</td>
<td>10 mm²</td>
</tr>
<tr>
<td>Lower limit of design variable</td>
<td>0.1 mm²</td>
</tr>
</tbody>
</table>
2.4. Proposed Approach of Compliant Mechanism in Passive Vibration Isolation

We propose compliant mechanisms as a means to provide efficient and low cost vibration isolation. Due to their monolithic (jointless) construction, compliant transmissions offer many inherent benefits including low cost, zero backlash, ease of manufacture, and scalability. Although leaf springs and cantilever beams employed in previous research are in effect of “compliant mechanisms”, the motion amplification mechanism proposed in this research offers a more effective solution.

The scope of this study is limited to low-frequency isolation because the use of compliant mechanisms in active vibration isolation systems has the greatest advantage in the low frequency range. Since many passive systems are effective and sufficient for high-frequency isolation, the need of active systems for high-frequency isolation is less than that needed for low frequency isolation. We also focus on understanding the effects of the compliant design parameters and attempt to solve problems systematically. The preliminary results of FEA from ANSYS demonstrate that compliant mechanism can be effectively used to reduce the amount of force transmitted to the surface. Figure 8 illustrates how compliant mechanism can be integrated into a vibration isolation system.

To achieve efficient vibration isolation, it is necessary to use a resilient support with sufficient elasticity so that the natural frequency of the isolated machine is substantially lower than the disturbing frequency, $f$, of vibration. The ratio $R$ should be greater than 1.4 and ideally greater than 2 to 3 in order to achieve a significant level of vibration isolation. Damping provides energy dissipation in a vibrating system. It is essential to control the potential high levels of transient vibration and shock, particularly if the system is excited at, or near, its resonant frequency. When it is not possible to prevent or sufficiently lower the transmission of shock and vibration from the source, a resiliently supported foundation block can be used for the passive isolation of sensitive equipment.

2.5. The Existing Coil Spring Isolator (FSL Coil Spring Isolator)

Farrat Isolevel Ltd. (FSL) coil spring isolation systems are used to provide both active and passive vibration isolation with natural frequencies down to 3 Hz in order to isolate the disturbing frequencies down to 6 Hz. Table 2 shows the existing coil spring isolator specification.

In this preliminary study, the existing coil spring isolator is used to reduce the force transmitted from or to the machine. A compliant mechanism is designed to reduce the force transmitted to the foundation by reducing the displacement transmissibility of various frequency ratios. The model shown in

ANSYS
Fig. 9 is designed with the load ranges from 85 kN–28 kN and constant $K$ ranges from 3.4 kN/mm to 1.12 kN/mm; the free height of the structure is 165 mm, and the static deflection due to the self-weight of the load is $\delta_{st} = 25$ mm and 8 mm for the corresponding maximum and minimum load.

The compliant mechanism is assumed to be made of structural steel. The gravity and structural damping are ignored for these preliminary analyses. The motion of output is contributed by displacement controlled input.

### 2.6. Material Selection for Compliant Mechanism

Material for this compliant mechanism is selected based on the Young’s modulus, which includes natural frequency and the area moment of inertia, mass, and also the cross sectional area of a compliant beam. The following equations are used for material selection:

- Natural frequency of compliant mechanism $\omega_n = \sqrt{\frac{k}{m}}$; (4)
- Material constant $k = \frac{192EI}{l^3} = m\omega_n^2$; (5)
- Young’s modulus of the material is $E = \frac{m\omega_n^2l^3}{192I}$; (6)
- Area moment of inertia $I = \frac{bh^3}{12}$; (7)

The size of the designed isolator is $500 \text{ mm} \times 305 \text{ mm} \times 165 \text{ mm}$. From the given maximum load of 85 kN, the maximum mass acting on the isolator is $m = 8500 \text{ kg}$, and the material constant is $k = 3400 \text{ N}$. By varying the dimension of the width and height of the isolator using the area moment of inertia, the thickness of the compliant beams are determined. In this the width of the isolator is $305 \text{ mm}$. Table 3 shows the selection of material using Young’s modulus.

Here the optimum range of dimension is $305 \text{ mm} \times 4 \text{ mm}$ which is having a Young’s modulus of $209 \times 10^9 \text{ N/m}^2$. The required range of $E$ value is around 200 Gpa. Figure 10 and Fig. 11 show the two dimensional and three dimensional respectively for the suggested optimum range of dimensions.

### 3. HARMONIC ANALYSIS

The harmonic response analysis solves the time-dependent equations of motion for linear structures undergoing steady-state vibration. The entire structure has constant or frequency-dependent stiffness, damping, and mass effects. All loads and displacements vary sinusoidally at the same known frequency. The element loads are assumed to be real (in-phase) only.

#### 3.1. Harmonic Response of FSL Coil Spring Isolator

Initially the displacement amplitude shown in Fig. 12 is calculated for various frequency ratios from (1.5–5) for the damping ratio $\zeta = 0.3$ of the coil spring isolator. The forces transmitted in Fig. 13 for the corresponding amplitude and frequency ratios are also calculated.

#### 3.1.1. Displacement Amplitude

The displacement amplitude is calculated by using the static displacement Eq. (8), the frequency ratio, and the damping ratio:

$$\frac{X}{\delta_{st}} = \frac{1}{\sqrt{(1 - R^2)^2 + (2\zeta R)^2}}. \quad (8)$$

Table 3. Selection of material using Young’s modulus.

<table>
<thead>
<tr>
<th>S. No</th>
<th>Dimension, mm</th>
<th>Young’s modulus $E$, N/m$^2$</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>305 $\times$ 3</td>
<td>$278 \times 10^9$</td>
</tr>
<tr>
<td>2</td>
<td>305 $\times$ 4</td>
<td>$209 \times 10^9$</td>
</tr>
<tr>
<td>3</td>
<td>305 $\times$ 5</td>
<td>$107 \times 10^9$</td>
</tr>
<tr>
<td>4</td>
<td>305 $\times$ 6</td>
<td>$60 \times 10^9$</td>
</tr>
<tr>
<td>5</td>
<td>305 $\times$ 7</td>
<td>$35 \times 10^9$</td>
</tr>
</tbody>
</table>
3.1.2. Force Transmitted

The force transmitted for the corresponding displacement amplitude is calculated by using the known material constant and the damping coefficient; it is taken as $\zeta = 0.3$ for the maximum value and the natural frequency of the coil spring isolator;

$$F_T = X \sqrt{\left(k^2 + c^2\omega^2\right)}.$$  \hspace{1cm} (9)

3.2. Harmonic Response of Compliant Isolator

3.2.1. Displacement Amplitude

The displacement amplitude is calculated for compliant mechanism for various frequency ratios ranging from (1.5–5) with the damping ratio $\zeta = 0.3$ by using ANSYS as shown in Fig. 14.

3.2.2. Force Transmitted

The force transmitted (Fig. 15) for the corresponding amplitude and frequency ratios are also calculated by using Eq. (9).

4. RESULTS AND DISCUSSION

4.1. Transmissibility Ratio

The force transmitted by using compliant mechanism is compared with the existing isolator with the constant damping ratio $\zeta = 0.3$ as shown in Fig. 16. The sinusoidal foundation motion at amplitude $x$ and the absolute value of the mass response amplitude $y$ expressed as a ratio $|y/x|$ is the $Tr$. Transmissibility Ratio $Tr = \frac{\text{Force transmitted in kN}}{\text{Disturbing force in kN}}$.

4.2. Isolation Efficiency

Isolation efficiency $\eta$ in percent transmission is related to transmissibility as

$$\eta = 100(1 - Tr)^\%.$$  \hspace{1cm} (10)

Isolation efficiency of the existing isolator and designed compliant mechanism is compared in Fig. 18.

5. CONCLUSION

Compliant mechanisms are proposed to provide cost effective and high performance vibration isolation systems. Their function is to transmit the force for various displacement amplitudes of corresponding frequency ratios. The preliminary results from FEA using ANSYS show that compliant mechanism can provide effective vibration isolation from a sinusoidal
disturbance with known frequency ratios. Both force transmissibility and amplitude transmissibility are discussed, as both of them have same results for validation purpose only.

In this research, we demonstrated, through harmonic analyses, that the disturbance of 0.01 m amplitude, the isolation efficiency of 56% is 1.5 Hz (for the coil spring isolator, the isolation efficiency for the corresponding amplitude and frequency ratio is 30%), and by 98%, it is at 5 Hz for the amplitude of 0.001 m (for the coil spring isolator, the isolation efficiency for the corresponding amplitude and frequency ratio is 93%), using compliant mechanism.

REFERENCES