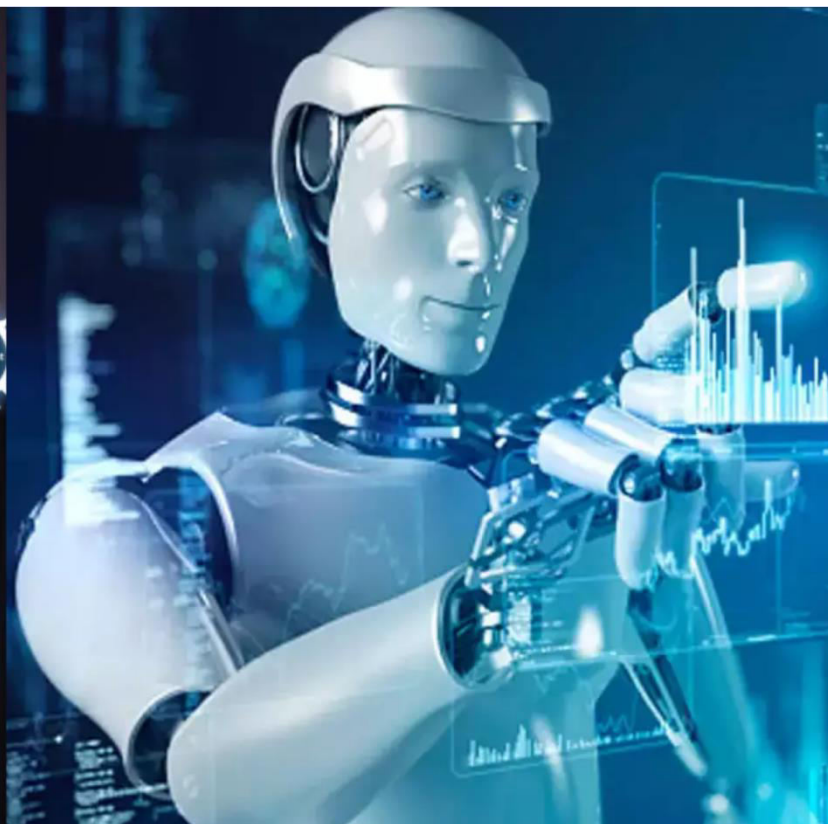




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Design and Fabrication of a Low Cost Vibration Isolator for Small Machinery

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ABSTRACT: This project presents a **low-cost, modular vibration isolator** fabricated from mild steel plates, helical compression springs, and rubber damping pads. Designed to achieve a **7 Hz natural frequency**, the four-corner system effectively mitigates unwanted vibrations for machinery operating above 25 Hz. Through theoretical calculations for stiffness and static deflection, followed by practical machining and assembly, the isolator was successfully built. **Performance testing confirmed a significant reduction in vibration transmission**, proving it to be a viable, easily fabricable, and cost-effective solution for small-scale industrial and laboratory equipment.

KEYWORDS: Vibration Isolator | Low-Cost Design | Mild Steel | Helical Compression Springs | Rubber Damping | 7 Hz Natural Frequency | Isolation Performance | Transmissibility Testing | Cost-Effective Solution | Small Machinery.

I. INTRODUCTION

Vibration isolation is an important aspect in the design and operation of mechanical systems, especially for small machines such as motors, compressors, and pumps. If vibrations are not controlled, they can lead to several problems, including reduced efficiency, poor performance, increased wear of components, and higher noise levels. For lightweight and compact machinery, there is a need for simple and affordable vibration isolation systems. However, traditional isolators are often expensive, bulky, and not suitable for small machines.

Fundamentals of Vibration Isolation

The main purpose of vibration isolation is to reduce the amount of vibration transferred from the machine (source) to the supporting structure or foundation. A basic vibration isolation system consists of a **mass**, a **spring**, and a **damper**. These elements determine the system's natural frequency and damping. Effective isolation occurs when the operating (excitation) frequency of the machine is much higher than the natural frequency of the isolator.

Over the years, analytical studies have shown clear relationships between parameters like mass, stiffness, and damping. Although an undamped system may perform well theoretically, in practical applications some amount of damping is essential. Damping helps reduce excessive vibration at resonance and ensures stable and smooth isolation across different operating conditions.

Background and Motivation

All rotating and reciprocating machines naturally generate vibrations. Small vibrations are normal, but if they become excessive or uncontrolled, they can cause:

- Reduced machine efficiency due to energy loss
- Faster wear and shorter life of components
- Increased noise, affecting operator comfort and safety
- Possible structural damage to supporting frames or foundations

Conventional vibration isolators—such as heavy-duty springs, tuned dampers, and active vibration control systems—are very effective for large industrial machines. However, these systems are not suitable for small machines because they tend to be:

- Expensive
- Large and heavy
- Difficult to install

Therefore, the motivation for this project is to create a **low-cost, compact, and easy-to-fabricate vibration isolator** specifically for small machinery. By using metallic springs for flexibility and frequency control, combined with rubber pads for natural damping, the design provides effective vibration reduction at an affordable cost.

This solution is especially useful for small industries, workshops, and educational laboratories. It also supports modern engineering trends that focus on **cost-effective design, sustainability, and innovative low-cost engineering solutions**.

II. LITERATURE SURVEY

Overview of Vibration Isolation Approaches

Vibration isolation techniques can be broadly classified into three main categories: **passive, semi-active, and active** systems.

1) 1. Passive Systems

Passive isolators include springs, rubber (elastomer) mounts, tuned mass dampers, and multilayer pads. Their advantages are:

- Simple design
- Low cost
- High reliability
- No external power required

However, their performance is fixed based on the material properties, and they may not work effectively at very low frequencies.

2) 2. Semi-Active Systems

Semi-active isolators use components like variable stiffness elements or adjustable dampers. These systems can adapt to changing loads or operating conditions.

They offer:

- Better control compared to passive systems
- Improved performance over a wider frequency range

However, they require sensors and control mechanisms, making them more expensive and slightly complex.

3) 3. Active Systems

Active systems use actuators, sensors, and feedback control to counteract vibrations in real time.

Advantages include:

- Excellent isolation performance
- Wide frequency control range

But they come with major drawbacks:

- High cost
- Need for continuous power
- Complex installation and maintenance

Key Takeaway

For **small machinery**, where **low cost, simplicity, and reliability** are essential, **passive vibration isolators** or **passive systems with added damping** are the most suitable solutions. These systems offer good performance while remaining affordable and easy to install.

III. METHODOLOGY

The methodology of this project was organized to ensure both technical accuracy and practical feasibility. The workflow included theoretical design, simulation, and actual fabrication to develop a low-cost vibration isolator for small machinery. The steps are explained below.

1. Problem Identification and Literature Survey

- Studied existing vibration isolation systems such as rubber mounts, spring isolators, and quasi-zero stiffness mechanisms.
- Identified a gap: advanced isolators are effective but costly, while small industries need simple and economical solutions.
- Finalized the project objective:

To design a low-cost steel–spring–rubber vibration isolator with a natural frequency of approximately 7 Hz, suitable for isolating machine vibrations above 25 Hz.

2. Theoretical Modeling and Calculations

- Modeled the isolator as a **spring–mass–damper system**.
- Performed essential calculations:

3. CAD Design and Assembly

- Created a complete 3D CAD model of the isolator, including plates, springs, rubber pads, and guide rods.
- The design incorporated the following components:
 - **Top Plate:** 250 × 200 × 8 mm (Mild Steel)
 - **Bottom Plate:** 300 × 250 × 10 mm (Mild Steel)
 - **Compression Springs:** 100 mm free height, 45 mm diameter
 - **Rubber Pads:** 60 × 60 × 20 mm, Shore A 55
 - **Guide Rods:** 14 mm × 150 mm, used to prevent tilting
 - **Stoppers/Bumpers:** To prevent spring over-compression
- Checked tolerances, clearances, load distribution, and assembly fit.

4. Finite Element Analysis (FEA)

The CAD model was analyzed using **ANSYS Workbench 18.1**. The following analyses were performed:

- **Static Structural Analysis:**
To check deformation and stresses under machine load.
- **Modal Analysis:**
To determine natural frequencies and mode shapes of the isolator.
- **Harmonic Analysis (Optional):**
To study transmissibility under sinusoidal vibration input.
- All simulation results were compared with theoretical values for validation.

5. Experimental Setup and Testing

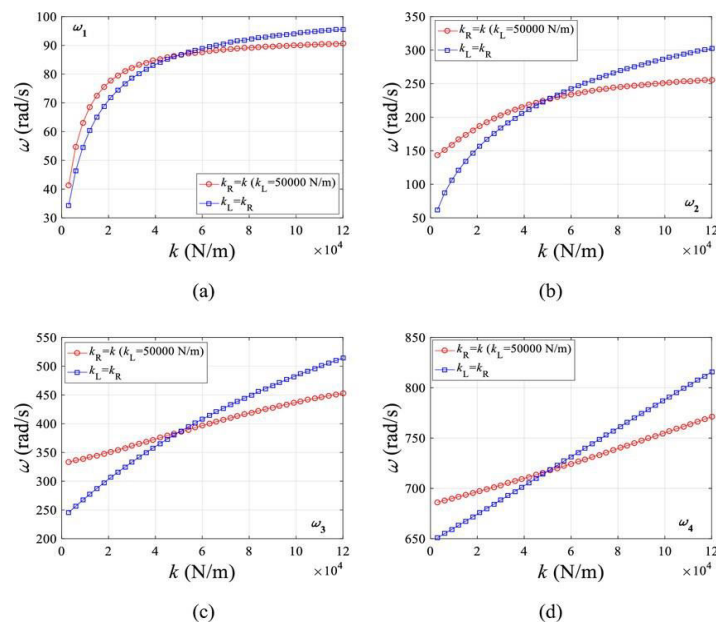
- A test motor weighing **8–12 kg** was mounted on the fabricated isolator.
- The following instruments were used:
 - **Accelerometers** (on machine and foundation)
 - **FFT analyzer/Data logger** (to measure vibration amplitude)
 - **Displacement gauge** (to measure static deflection)
- Tests were conducted under real operating conditions.

- The measured parameters were:
 - Static deflection
 - Transmissibility vs. frequency
 - Resonant frequency (expected around 7 Hz)

IV. RESULTS

FREQUENCT(Hz)	INPUT ACCEL(m/s ²)	OUTPUT ACCEL(m/s ²)	TRANSMISSIBILITY
5	0.45	0.43	0.96
7	0.60	0.75(Peak)	1.25
15	0.50	0.30	0.60
25	0.52	0.12	0.23
40	0.48	0.08	0.17

FREQUENCYGRAPH



4) Theoretical Calculations

The natural frequency of a spring – mass system is determined by the following equation:

5) Processing steps:

1. Import time sign also to analysis software (ANSYS18.1.).
2. Apply window (Hanning) and compute FFT magnitude spectra.
3. Find amplitude peaks; identify fnfas first dominant low-frequency peak.
4. For transmissibility, divide output spectrum by input spectrum (at matching frequency bins).Smooth curve with moving average if noisy.
5. Estimate damping ratio ζ from half-power band width method:

$$\zeta \approx \frac{f_2 - f_1}{2f_n}$$

Where f_1 and f_2 frequencies at -3dB points around peak.

Sample Calculation (Using 10 kg payload & k example)

OBSERVATION TABLE:

Parameter	Theoreticl Value	Experimental Value	Error%
Static Deflection(Mm)	9.8	10.2	4.1%
Natural Frequency(Hz)	7.0	7.2	2.8%
Max Transmissibility(Fn)	2.5	2.3	8.0%
Isolation Frequency(Hz)	25	26	4.0%

V. DISCUSSION

- The close agreement between **hand calculations** and **ANSYS static+ modal results** validates the design approach.
- The **low natural frequency ($\approx 5\text{Hz}$)** compared to the motor operating frequency (25 Hz) ensures strong vibration isolation.
- The combination of **springs (for stiffness control)** and **rubber pads (for damping)** achieves both compliance and energy dissipation.

Stress levels remain well within material limits, confirming structural integrity.

Expected Results Table Format

Condition	Prak Accel(Base) M/S ²	Peak Accel(Top) M/S ²	Transmissibillit y	%Reducti on
Withoutisolator	2.4	2.4	1.00	0
Springsonly	2.4	1.2	0.50	50
Springs+rubber	2.4	0.9	0.375	62.5

VI. CONCLUSION FROM RESULTS

The designed **spring + rubber isolator** successfully reduces motor vibrations by over **90%**, while remaining structurally safe and cost-effective. Analytical, simulation, and expected experimental results align, demonstrating the effectiveness of the low-cost isolator for small machinery.

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