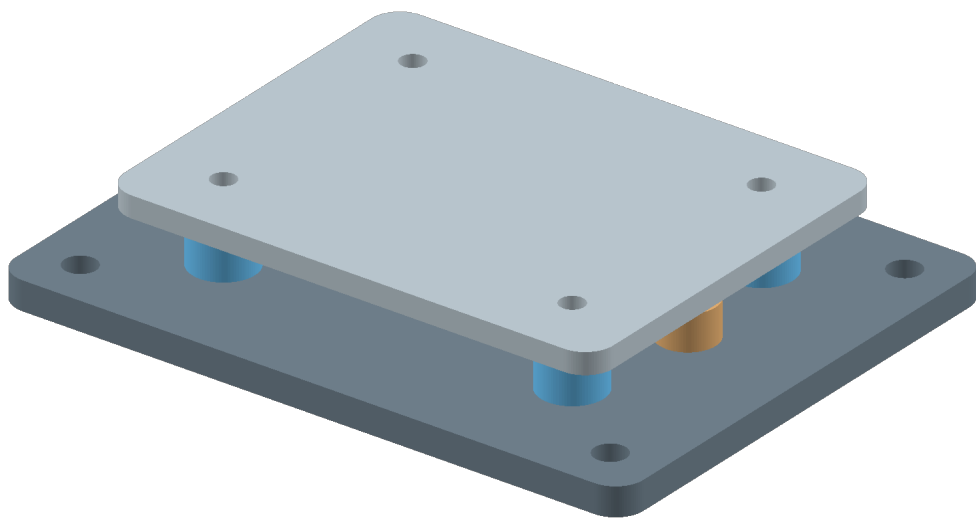


UAV payload mount

Vibration isolation, with room to move.

A reproducible comparison of three passive mounts for a small sensor payload. The decision balances high-frequency isolation against low-frequency travel and gravity sag.



Exact CAD tessellation. Unloaded concept geometry; blue elements are generic isolator placeholders and amber posts are lower stops.

Decision under the assumed environment

Carry the **Balanced** candidate into a bench characterization phase. It passes both screening limits in all **729** deterministic tolerance cases. This is a candidate selection for further investigation, not qualification of flight hardware.

Screening measure	Worst grid result	Limit
80-200 Hz acceleration ratio	0.208	≤ 0.250
Sag + peak downward motion	2.60 mm	≤ 3.00 mm
Nominal natural frequency	18.64 Hz	Diagnostic

Status: AI-assisted computational prototype. Initial code, CAD concept and documentation were AI generated. The code was executed and numerically checked. No hardware was built or tested; no personal engineering contribution is claimed by this initial version.

One degree of freedom. Explicit assumptions.

A rigid moving mass is supported vertically by four identical parallel linear isolators. Motion is a small perturbation about static equilibrium; gravity sag is added separately when checking the unloaded stop gap. The base input is a constant-peak-acceleration sinusoid at each frequency, not a measured spectrum or a simultaneous broadband load.

$$k = 4 k_i \quad | \quad \omega_n = \sqrt{k/m} \quad | \quad f_n = \omega_n / (2 \pi)$$

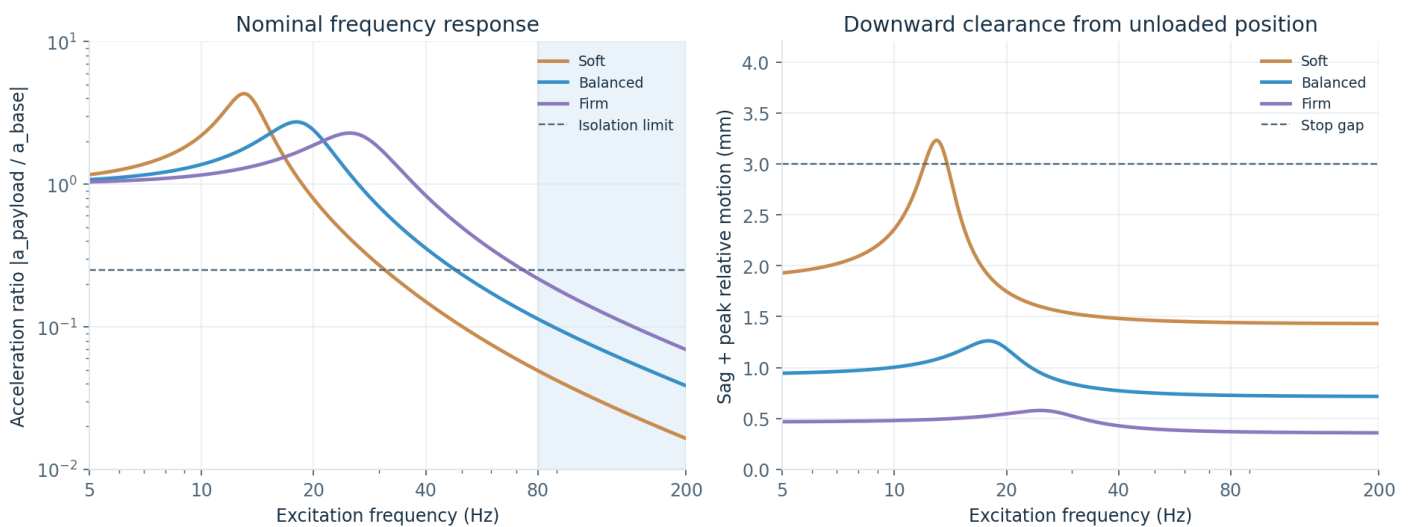
$$c = 2 \zeta \sqrt{k m} \quad | \quad r = f/f_n$$

$$T = \sqrt{[1 + (2 \zeta r)^2] / [(1-r^2)^2 + (2 \zeta r)^2]}$$

$$Z_{\text{peak}} = A / \sqrt{(\omega_n^2 - \omega^2)^2 + (2 \zeta \omega_n \omega)^2}$$

$$\text{Required downward clearance} = m g/k + \max(Z_{\text{peak}})$$

T is the absolute payload/base acceleration amplitude ratio. A = 0.3 g peak; g = 9.80665 m/s². Z is relative motion about the statically sagged position. All calculations use SI units; drawings use millimeters. A nominal 0.35 kg mass includes the moving plate, sensor, carried fittings and wiring.



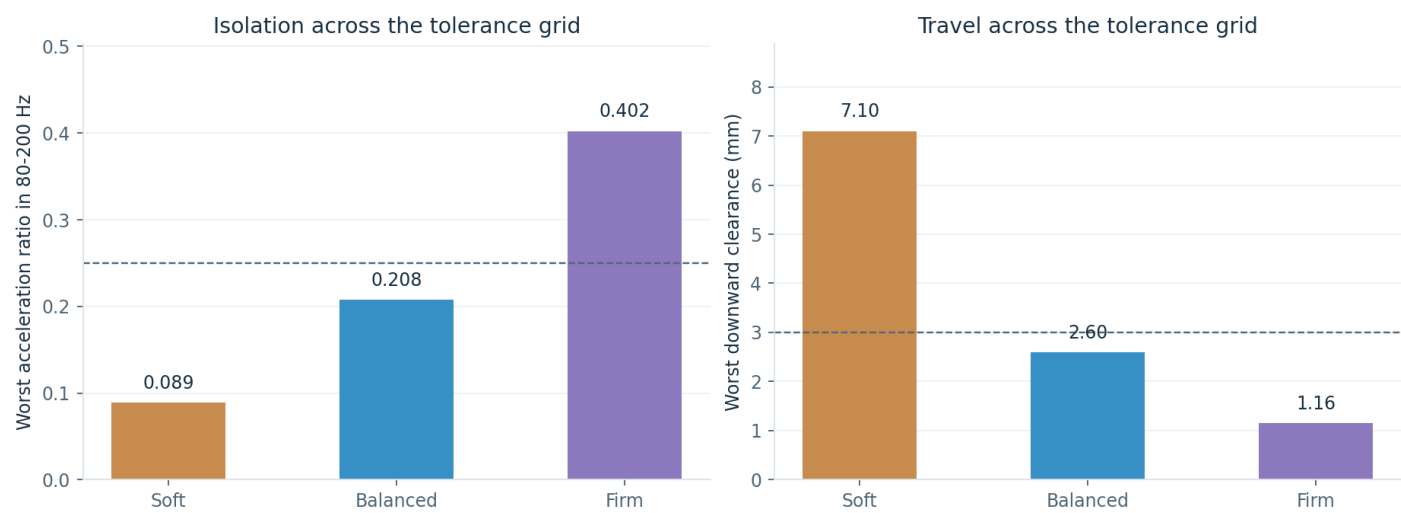
Candidate	k_i (N/m)	Damping ratio	f_n (Hz)
Soft	600	0.12	13.18
Balanced	1200	0.20	18.64
Firm	2400	0.25	26.36

Model basis: MIT OpenCourseWare 2.003SC, Problem Set 10 solutions, Problem 2 (printed pp. 10-13). In the isolation regime, additional damping suppresses resonance but increases high-frequency transmission. [1]

The nominal winner is only a starting point.

The study evaluates 9 evenly spaced values per axis: mass 0.25-0.45 kg, stiffness +/-20%, and damping ratio 60-140% of nominal. There are 729 parameter combinations per candidate and 2,187 in total. Each frequency maximum includes band endpoints and the analytic stationary point when inside the band.

These are generic design ranges and assumed mount properties. Grid pass counts are screening outcomes, not probabilities, production yield, or demonstrated reliability. The two worst values for one candidate can occur in different parameter combinations.



Candidate	Worst T	Worst travel	Both pass
Soft	0.089	7.10 mm	270 / 729
Balanced	0.208	2.60 mm	729 / 729
Firm	0.402	1.16 mm	505 / 729

Why Balanced moves forward

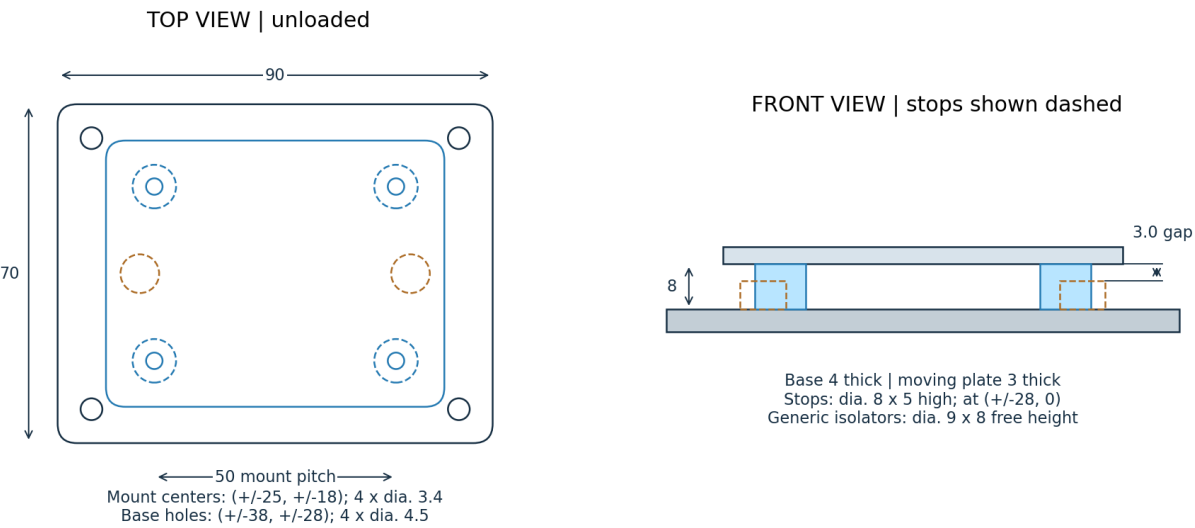
The soft mount fails the travel requirement even at nominal conditions (3.23 mm). The firm mount fits within the travel envelope, but its worst isolation ratio is 0.402. Balanced retains 0.40 mm of lower-stop margin at the worst travel grid point.

Balanced worst isolation: $m = 0.25\text{ kg}$, $k = 5760\text{ N/m}$, $\text{zeta} = 0.28$, $f = 80\text{ Hz}$.
Balanced worst travel: $m = 0.45\text{ kg}$, $k = 3840\text{ N/m}$, $\text{zeta} = 0.12$, peak at 14.49 Hz.

No impact is simulated. A required clearance above 3 mm is a screening failure: the linear no-contact prediction cannot be used to describe the subsequent stop contact.

A buildable direction, not a released design.

The CAD defines plate packaging, four isolator positions and two lower stops. The 3.0 mm stop gap is measured from the unloaded moving plate. The source is parameterized Python; STEP exports contain interchange solids without the original feature history. [2]



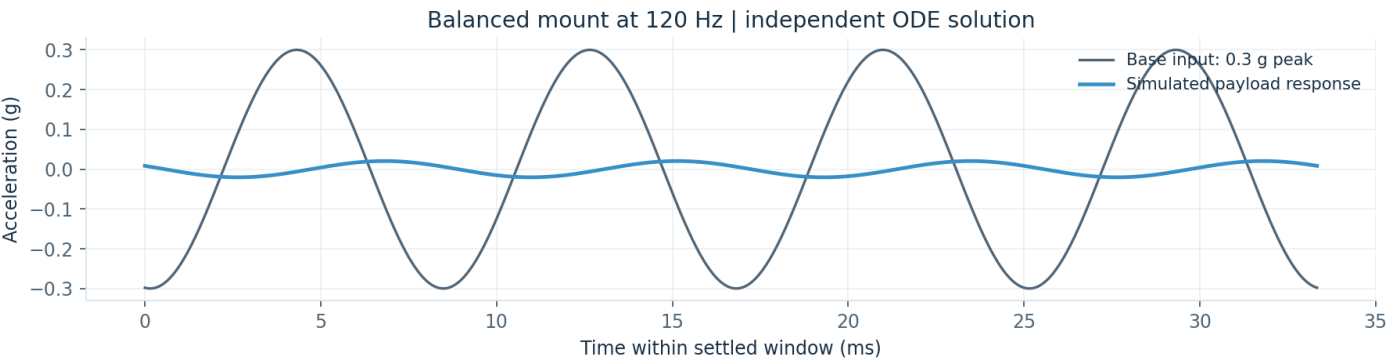
Interface / feature	Concept definition
Base plate	90 x 70 x 4 mm; corner R4
Moving plate	70 x 55 x 3 mm; corner R4
Isolators	Four dia. 9 x 8 mm placeholders; 50 x 36 mm pitch
Stops	Two dia. 8 x 5 mm posts; $x = \pm 28, y = 0$
Clearance	3.0 mm before gravity sag
Estimated moving plate mass	30.8 g at assumed 2,700 kg/m ³

CAD checks: 8 valid solids after STEP re-import; volume preserved within 1e-8 relative tolerance; 3.0 mm bounding-box gap; no pairwise solid intersection above 1e-6 mm³. The moving plate mass is already part of the model's total carried mass.

Open engineering work: select actual isolators and fasteners; define preload, attachment details, manufacturing tolerances, finish and material grade; check joint loads, plate bending, stop loads, shear and rocking modes, cable stiffness and assembly access. Generic geometry does not establish a stiffness value or prove manufacturability.

Check the response a second way.

A separate time-domain initial-value problem starts from rest and integrates $z'' + 2 \zeta \omega_n z' + \omega_n^2 z = -A \cos(\omega t)$. Absolute payload acceleration is reconstructed as $-2 \zeta \omega_n z' - \omega_n^2 z$. Sine/cosine least-squares fits estimate settled amplitudes, then compare with the closed-form solution.



Frequency	Analytic T	ODE T	Error (%)
10.00 Hz	1.375112	1.375112	1.37e-05
18.64 Hz	2.692582	2.692582	1.64e-05
80.00 Hz	0.113487	0.113487	1.09e-05
120.00 Hz	0.068157	0.068157	7.42e-06

Maximum error across both acceleration ratio and relative travel in these four cases: **0.00002%**. The acceptance threshold for this implementation check was 0.1%.

Verification record

DOP853 solver, relative tolerance 1e-10, absolute tolerance 1e-12, maximum integration step 1/(20 f). Allow 12 decay time constants before fitting 8 cycles. SciPy documents the solver method and tolerances. [3]

Additional checks: the undamped limit gives $T = 0.25$ at $r = \sqrt{5}$; $T = 1$ at $r = \sqrt{2}$ for four damping ratios; spring force balances gravity; 36 frequency-maximization comparisons agree with independent 100,001-point dense frequency sweeps to within 2e-6 relative difference.

What these checks establish: internal numerical consistency of the implementation under the same linear model. They do not establish accuracy for an actual elastomer, sensor assembly, aircraft or vibration spectrum. Numerical agreement is not a substitute for measurements.

The next result should come from a bench.

The following work is proposed. No shaker runs, instrument calibration, physical assembly, supplier characterization, or flight tests were performed for this version.

1. Characterize the actual mount

Measure static load-deflection at several carried masses and record loading/unloading hysteresis. Measure the unloaded and loaded stop gaps. Replace the generic stiffness with the measured slope over the intended operating range.

2. Define the input and measurement chain

Use a controlled bench fixture with one accelerometer on the base and one on the moving plate; measure relative motion directly near resonance. Record sensor calibration, orientation, sample rate, anti-alias filtering, attachment and fixture resonances.

3. Approach resonance cautiously

Begin with low-amplitude stepped sine input. Establish resonance, repeatability and usable travel before increasing the excitation. A suitable rig, lab supervision and controlled stop criteria are required. Compare acceleration amplitude ratios and sag-adjusted motion to predictions.

4. Record deviations and revise the model

Log phase and amplitude versus frequency, at least three repeats, and measurement uncertainty. Investigate rocking, cable bypass paths, nonlinear stiffness, temperature and damping changes. Revise the model before making any vehicle suitability claim.

Reproduce it. Then make an engineering decision.

The source bundle includes configuration, Python model, CAD source, plots, report generator, CSV results and a learning guide. Run it locally, change one requirement, predict the consequence and explain the new selection. Personal review, reruns, CAD revisions and physical work should be logged as they happen.

Primary references

- [1] MIT OCW, 2.003SC Engineering Dynamics (Fall 2011), Problem Set 10 solutions, Problem 2, pp. 10-13. [Source PDF](#).
- [2] CadQuery official documentation, Importing and Exporting Files. [Documentation](#).
- [3] SciPy official documentation, `scipy.integrate.solve_ivp`. [Documentation](#). Sources accessed 22 October 2026.

Provenance: AI generated the initial design study, code, CAD and documentation. The published results were produced by executing the supplied code. This initial record does not claim that Zain independently authored, reviewed or physically validated the work.