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HAM Optics Table
Mechanical Design and Analysis

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Abstract

A preliminary analytical design for the HAM optics table is developed. The design is checked for natural frequencies, thermal noise, and static stresses.

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LIGO PROJECT

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1. Design Requirements

1.1 Dimensions

The optics table must have a rectangular and plane upper surface for mounting optical components. The minimum dimensions are 1.70 m x 1.90 m.

1.2 Response to Residual Seismic Noise

Figure 1 shows the isolation requirements for the HAM stacks[1], compared to the expected performance of 3-stage[2] isolation stacks using Viton springs[2] (worst case since the performance of metal spring stacks is far superior) and coil springs[2].

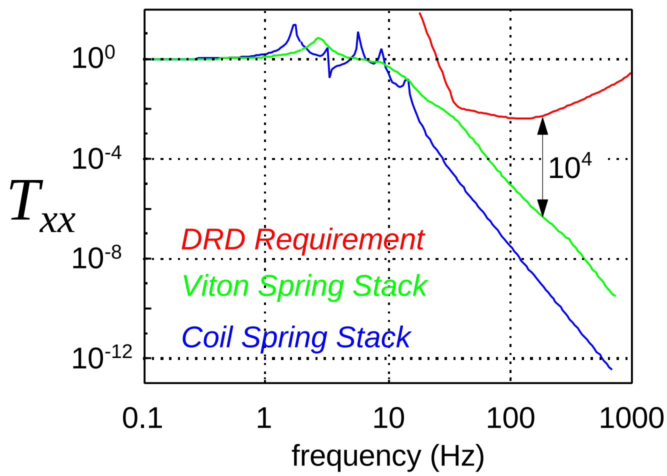

Figure 1: Approximate isolation performance with 3-stage Viton stack and with 3-stage coil spring stack.

The figure shows that even with a viton spring stack, a factor 10000 margin exists between the expected performance and the design requirement, at frequencies above 200 Hz. Resonances at 200 Hz or more will therefore not produce violations of those requirements (with Q's of 10000 or less and participation factors <1).

1.3 Response to Thermal Noise

Fred Raab has derived a limit on thermal noise-induced horizontal motion spectrum (m/√Hz) for the HAM optics table[3]. This limit $x_{platform}(f)$ is shown in Fig. 2 and can be used to derive an analytical expression for the maximum allowable quality factor Q_i of a natural mode #i of the optics table as

$$Q_i \leq Q_i^{\max} = \frac{8p^3}{4k_B T} \cdot \left| x_{platform}(f_i) \right|^2 \cdot m_i \cdot f_i^3 , \qquad (1)$$

where T is the optics table temperature (293 °K typ.), k_B is Boltzmann's constant (1.381×10^{-23} J/°K), m_i is the effective mass of mode #i at the attachment point of the mirror pendulum, and f_i is the natural frequency of mode #i in Hz.

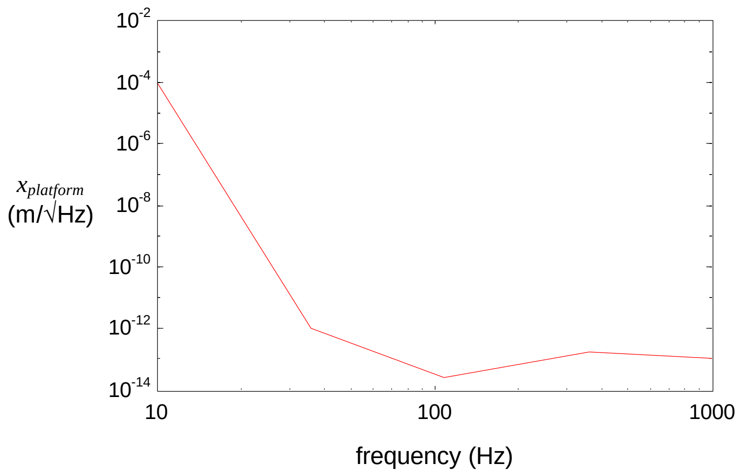

Figure 2: Thermal noise limit for HAM optics table[3].

2. Analysis

The optics table is supported by soft springs (Viton, coil or, leaf). The natural frequencies associated with these springs (stack component oscillating as a rigid body on the springs) range from about 2 to 30 Hz, an order of magnitude below the expected natural frequencies of the optics table. This decouples the dynamic behavior of the stack components from their supports, essentially simulating free-free boundary conditions. Because of this, all modal analysis reported in this document was performed in free-free conditions.

The optics table is analyzed with finite elements (NASTRAN), using shell elements. The entire top face of the optics table is given an artificially high density to simulate a payload of 227 kg (500 lbs) uniformly smeared on the optics table.

3. Description of Current Design

Figure 3 shows the current configuration of the optics table. It consists of a welded aluminum sandwich plate 1.7m long by 1.9m wide by 34 cm thick, weighing approximately 410 kg (903 lbs). The top and bottom plates are 19.1mm (.75”) and 12.7 (.5”) thick, respectively; They are separated by a total of 14 web plates 6.35 mm (.25”) thick, forming an array of 6×6 cells.

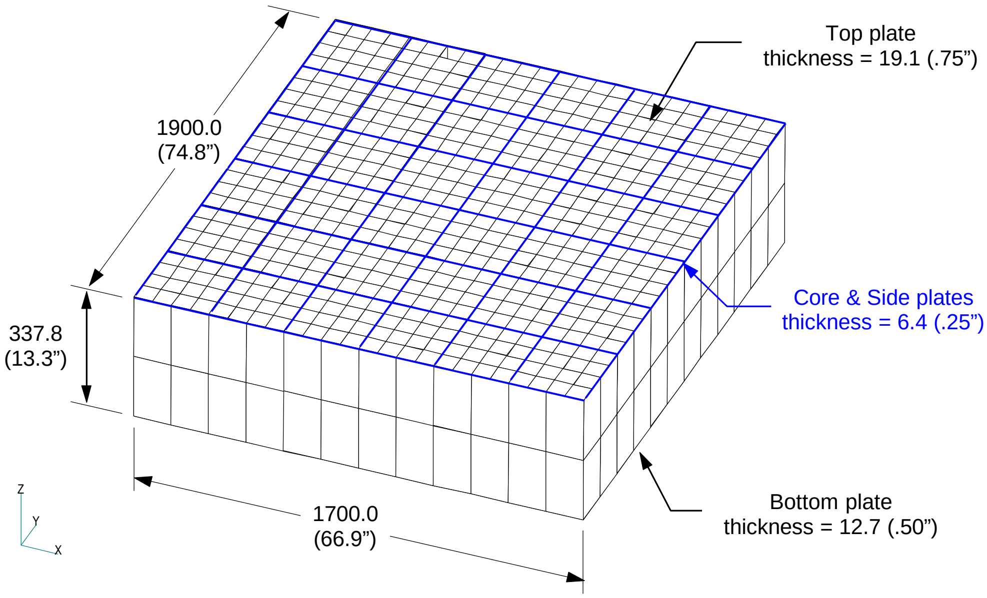

Figure 3: HAM optics table design geometry; all dimensions in mm (inches); blue lines show location of internal web plates.

4. Performance

4.1 Natural Modes

The first natural frequency is 250 hertz, comfortably above the requirements of the system (200 Hz). Higher modes occur at 342, 397, 457 Hz and higher (Fig. 4).

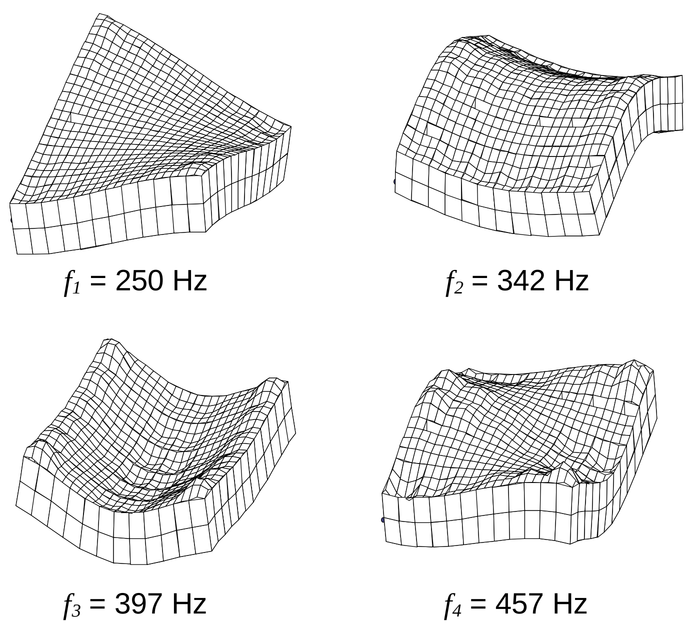

Figure 4: First 4 natural modes.

4.2 Thermal Noise Evaluation

Free-free natural modes obtained from NASTRAN runs (Fig. 4) are used to evaluate effective masses m_i of the optics table. Because the locations of the mode cleaner mirrors attachment points are not completely defined at this point, the effective masses were conservatively calculated for the largest displacement in the horizontal plane at any of 9 points located at the corners, centerpoints of the edges, and center of the optics table upper surface. Using equation (1), maximum allowable *Q*'s are estimated for each mode and listed in Table 1.

mode i	f_i (Hz)	m_i (kg)	Q_i^{max}
1	250	8074	1.78×10^{7}
2	342	3737	5.88×10^{7}
3	397	7803	2.02×10^{8}
4	457	4447	1.51×10^{8}
5	474	4350	1.59×10^{8}
6	559	7082	3.56×10^{8}
7	584	21456	1.17×10^{9}
8	584	8216	4.50×10^{8}
9	596	7388	4.21×10^{8}
10	615	7412	4.49×10^{8}
11	622	114457	7.08×10^{9}
12	622	74332	4.60×10^{9}
13	623	24995	1.55×10^{9}
14	628	593637	3.75×10^{10}
15	639	66730	4.35×10^{9}
16	643	85754	5.66×10^{9}
17	645	63563	4.23×10^{9}
18	654	12253	8.35×10^{8}

Table 1: Maximum allowable downtube *Q*'s for thermal noise response.

First note that most modes listed in the table have very large effective masses for horizontal motion of the optics table surface. This is because those modes have very small transverse amplitudes (see Fig 4); they involve primarily overall bending deflections of the optics table (normal to the optics table surface).

The lowest of all *Q* requirements applies to the first elastic mode and is about 10^7 (0.00001% loss factor), which is orders of magnitude above typical *Q*'s for large welded structures (100-500) or even material damping at nanostrain level in solid aluminum (400-8400[4]).

4.3 Static Stresses in Spring Maintenance Condition

As a means to allow complete unloading of the springs for maintenance or vacuum bake, we examine the option of suspending the stacks from the bottom plate of the optics table sandwich through a number of specially designed pins. Analysis is performed to verify that the static stresses created in the sandwich structure in those conditions are acceptable.

As expected, stresses due to gravity loads with the stacks hanging from the optics table lower plate are extremely low. In the analysis, the optics table is simply supported at each corner of the bottom plate. Nodes of the bottom plate in the vicinity of the legs are given increased mass to represent the mass of the stacks (2 nodes per leg). A 500 lbs payload is also smeared on the top plate and a standard gravity acceleration field is imposed on the model. Maximum nodal stress is about 3 MPa (435 psi). The yield strength for annealed aluminum alloys range from about 55 MPa (8 ksi) for 6061-O to 117 MPa (17 ksi) for 5086-O.

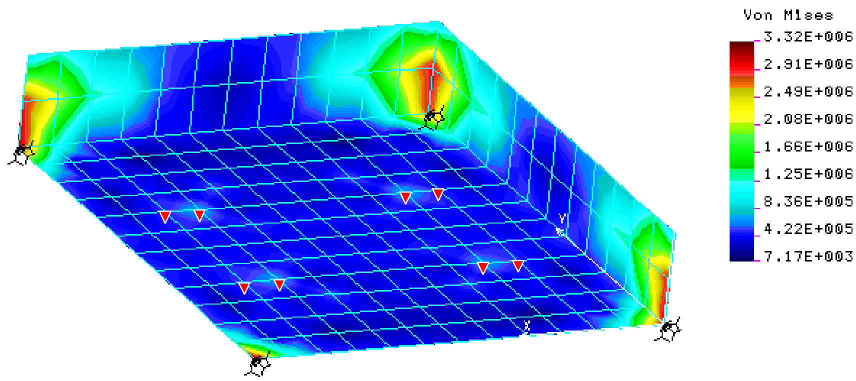

Figure 5: Nodal Von Mises stresses (Pa) due to gravity with optics table hanging by 4 corners, 500 lb payload smeared on top plate, and stacks hanging from bottom plate (seen from below).

5. References

1. F. Raab and N. Solomonson, "Seismic Isolation Design Requirements Document," LIGO-T960065-02-D, April 15, 1996.
2. E. Ponslet, "HAM Seismic Isolation - Projected Performance," HYTEC-TN-LIGO-13, January 14, 1997.
3. F. Raab, "Thermal-Noise Requirements for HAM Seismic Isolation," LIGO-T960188-00-D, January 7, 1997.
4. J. M. Ting and E. F. Crawley, "Characterization of Damping of Materials and Structures from Nanostrain Levels to One Thousand Microstrain," *AIAA Journal*, Vol. **30**, No. 7, July 1992.

Note 1, Linda Turner, 09/03/99 11:36:42 AM
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