LASER INTERFEROMETER GRAVITATIONAL WAVE OBSERVATORY

LIGO Laboratory / LIGO Scientific Collaboration

LIGO- T080252 -01 *ADVANCED LIGO* 6th November, 2008

Performance of a commercial pneumatic isolation system for the ISC Output Chambers (HAMs 1 & 7)

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Rev 00: Initial Draft

Rev 01: Includes updated FEA

1.1 Introduction

The purpose of this study was to investigate the performance of a commercially available pneumatic isolation system for HAM 1/7. The HAM1 chamber (for the H1 and L1 interferometers; HAM7 for the H2 interferometer) is an optical readout chamber for the Interferometer Sensing and Control (ISC) subsystem. Unlike the other HAM chambers, which have more stringent isolation requirements, HAM1/7 was not base-lined with the same high performance seismic isolation system; These chambers have neither a Hydraulic External Pre-Isolator (HEPI) system, not an Internal Seismic Isolation System (ISI). The baseline notion is to employ a commercially available pneumatic isolation system. The isolation performance goals are to obtain a factor of 10 reduction in ground noise at 10 Hz with little (no) amplification below 10 Hz

We started by obtaining the properties of a simple rigid model of the HAM seismic support platform. We then selected a catalogue pneumatic isolator that could handle the weight of this system and which had the lowest available isolation frequency. Using MATLAB we next calculated (not reported here) the expected isolation performance for this rigid system approximation and compared it a simple spring/damper in a finite element analysis (FEA), as a check. Lastly to calculate the transfer function, of the full system, from ground to optics table we performed an FEA in ANSYS.

1.2 Mass of HAM (Simple rigid) Seismic Support System

The following mass properties of all of the components making up this HAM configuration were obtained from drawings supplied by Andy Stein and Ken Mason, the references of the key items are shown below.

Support Table (D972711)	168 kg	(1x 168 kg)
Support Tubes (D972610)	183 kg	(2x 91.5 kg)
Optical Table (D972710)	382 kg	(1 x 382 kg)
Mounting Bases	20 kg	(4 x 5 kg)
Bearings	9 kg	(4 x 2.3 kg)
HAM spacers	9 kg	(2x 4.5 kg)
Updated Crossbeams*	292 kg	(2 x 146 kg)

TOTAL (with updated crossbeam)	1372 kg
Implies per leg (4 legs)	343 kg

* Information from SolidWorks assembly supplied by Andy Stein via link in an e-mail on 29th August 2008. Additional information on the new crossbeam designs can also be found in LIGO-E080328 and LIGO-E080166

The mass per leg of 343 kg was then used to select a catalogue isolator (see section 1.3) and to build a model representing the mass of the HAM Seismic Support System (see section 1.4).

It should be noted that after our analysis was complete we realized that mass for the payload on the HAM optics table was not included. If our estimate of 100 kg (total) is correct it would only be a small correction to the results quoted in this report. For comparison the payload should be far less than that for the HAM-ISI which is 510 kg, as per LIGO-E040136-00.

1.3 Newport Isolator

After researching several isolators we decided to select the Newport I-2000 Series Isolator. The following specifications, in figure (1) below, were obtained from the Newport web site. The published transfer functions are also shown in Appendix 1. It should be noted that these specifications are for 900 kg per leg.

Specifications

Specifications

Isolation Specifications

Model	Vertical Isolation*			Horizontal Isolation*			Amplification at Resonance		Damping Element Airflow	Horz. Damping	Load per Isolator [lb (kg)]
	Res (Hz)	5Hz (%)	10Hz (%)	Res (Hz)	5Hz (%)	10Hz (%)	Vert (dB)	Horz (dB)			
I-2000 Series	1	94	98	1.5	85	95	13	13	Normal	Oil	2000 (900)
CI-2000 Series	1	94	98	1.5	85	95	13	13	Normal	Oil	2000 (900)
I-2000S-428TC	1	94	98	1.5	85	95	13	13	Normal	Oil	2000 (900)

Figure (1): - Extract from Newport Web-site of Specifications of Isolator I-2000 series CI-2000 is the cleanroom version. I-2000S is a square cross-section version. The performance of all of the I-2000 series are identical. The load capacity ranges from 165 lb (83 kg) to 2000 lb (900 kg) per isolator.

Figure (2):- I-2000 series Newport Isolator

These I-2000 isolators come with automatic leveling and have a footprint of 8 inches [203.2 mm] diameter by a height of 10 to 28 inches [714mm] high, depending upon model. Full details can be found at the following url: -

http://www.newport.com/store/genproduct.aspx?id=139763&lang=1033&Section=Pricing

The vertical isolation is due to a spring, and air damper, and hence depends upon the mass of the system. The resonant frequency, f_v is

$$f_v = \left(\frac{1}{2\pi}\right)\sqrt{\frac{k}{m}}$$

where k is the effective spring stiffness and m is isolated mass. For the maximum load (900 kg per isolator), the resonant frequency is 1 Hz. The horizontal isolation is due to a pendulum, and oil damper, and hence does not depend upon the mass of the system. Using our payload of 343 kg per leg and the data from figure (1) we calculated, a vertical frequency of 1.64 Hz. In order to achieve a vertical isolation of 1 Hz, we would have to add ~900 – (343+100/4) = ~530 kg per leg. This is equivalent to a cube of steel which is 15 inch (390 mm) on each side.

The resonant frequency, f_h is

$$f_h = \left(\frac{1}{2\pi}\right)\sqrt{\frac{g}{L}}$$

Where g is the gravitational acceleration constant and L is the pendulum length. The horizontal isolation frequency is 1.5 Hz, independent of isolated mass, implying an L of 110 mm. In section 1.5.3 we tweak the horizontal (X and Z) ‘k” in the model in order to obtain a horizontal frequency of ~ 1.5 Hz.

1.4 Mass representing HAM Seismic Support System

Using SolidWorks we created a mass representing the HAM Seismic Support System equivalent to that calculated in section 1.3 and based on the actual CAD assembly obtained from Mason and Stein. The simplified model created for the forthcoming FEA work can be seen in figure (3) below. As an example we created a representation of the Support Table (D972711) and the Optical Table (D972710) that matched the real system in mass and approximately (better than factor of 2) in mass moment of intertia. The mass and moments of inertia (taken at the center of mass and aligned with the output coordinate system) of the entire isolated mass model are compared to the moments of inertia of the seismic support structure in the table below.

1) Isolated Mass (SW)			
Mass = 1439.92 kg			
Moments of inertia (kg.m^2)	Lxx = 1248.90	Lxy = 2.08	Lxz = -0.11
	Lyx = 2.08	Lyy = 2135.43	Lyz = -2.92
	Lzx = -0.11	Lzy = -2.92	Lzz = 1080.98
2) Modal Mass (SW)			
Mass = 1419.61 kg			
Moments of inertia (kg.m^2)	Lxx = 1199.95	Lxy = 0.00	Lxz = -0.00
	Lyx = 0.00	Lyy = 2070.35	Lyz = 0.00
	Lzx = -0.00	Lzy = 0.00	Lzz = 1066.62

Table (1): Comparison of Moments of Inertia of the model and actual isolated mass

The location of the spring/damper support points are (as indicated in Figure 3) spaced as per the actual support structure, the pier supports are spaced 2.392 m by 3.039 m apart (see D972501-B, sheet 5).

From now on this representation will be referred to as the modal mass. The ballooned section in figure (3) bellows shows an additional item that had to be added to the model prior to exporting it to ANSYS. These 4 cubes along us to better represent the real interface between the isolated mass and the pneumatic isolators. As a result the mass and moments of inertia used in the FEA model are as follows: -

Mass = 1625.95 kg			
Moments of inertia * (kg.m^2)	Lxx = 1354.85	Lxy = 0.00	Lxz = -0.00
	Lyx = 0.00	Lyy = 2724.58	Lyz = 0.00
	Lzx = -0.00	Lzy = 0.00	Lzz = 1569.41

Table (2): result the mass and moments of inertia used in the FEA model

The ballooned section, in the figure (3) below, also shows an example of the small pads that were created in SolidWorks. These are used in ANSYS to connect the modal mass to the springs that will represent the Newport Isolator.

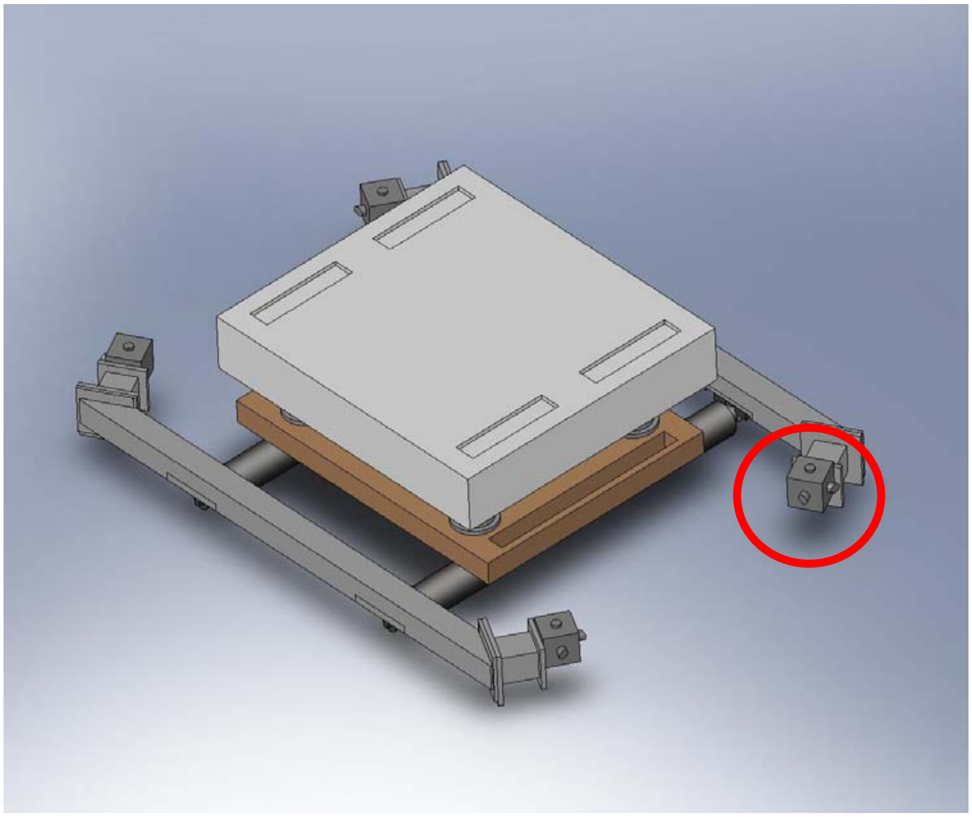


Figure (3): - Modal Mass representing the HAM Seismic Support System. The ballooned sections, in the figure below, shows an example of the small pads that were created in SolidWorks.

1.5 FEA Analysis without the Bellows

1.5.1 Co-ordinate system

For all of the FE analysis carried out in this report the following co-ordinate system was assumed: - Y vertical, X longitudinal and Z transverse.

1.5.2 Adding the springs and a representation of the ground

The modal mass, shown in figure (3) above, was then imported from SolidWorks into ANSYS Workbench 11.1. As shown in figure (4) we added a series of springs representing the Newport Pneumatic Isolators. At each corner of the modal mass we connected 3 springs, to the pads described in section 1.4. Each set of 3 springs is equivalent to one Pneumatic Isolator.

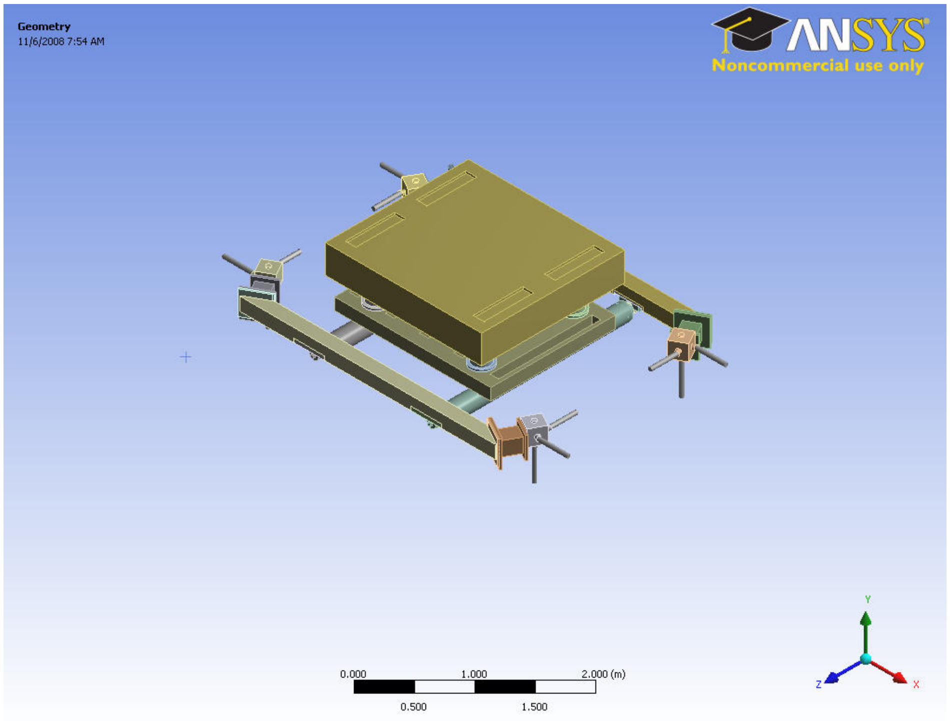


Figure (4i): - Modal Mass with springs representing the Newport Pneumatic Isolator

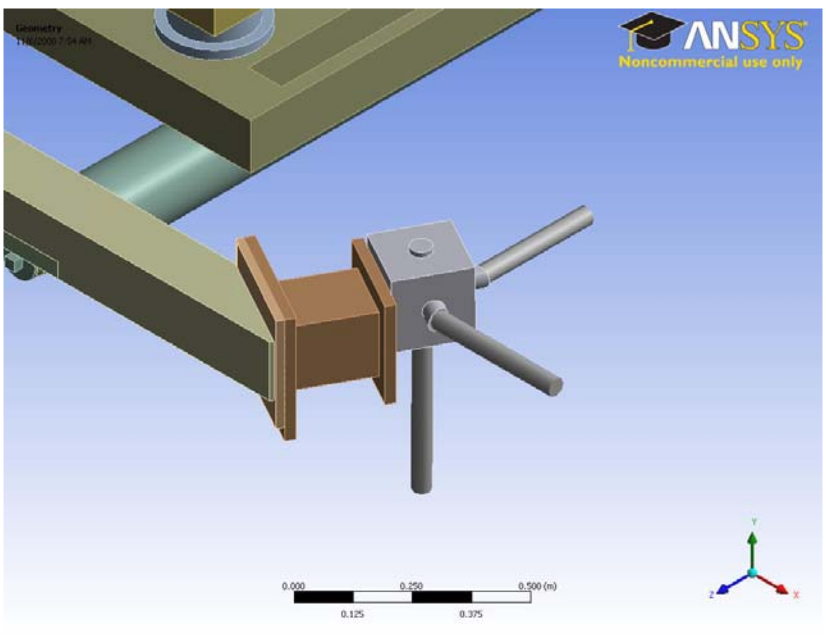


Figure (4ii): -Close up of springs representing the Newport Pneumatic Isolator

In order to obtain the transfer function (magnitude and phase) of this system in X, Y and Z the other end of each spring was connected to a large mass representing the ground. The ground mass was several thousand time the weight of the modal mass. In figure (5) all of the mass representing the "ground" is shown along with the modal mass and the springs.

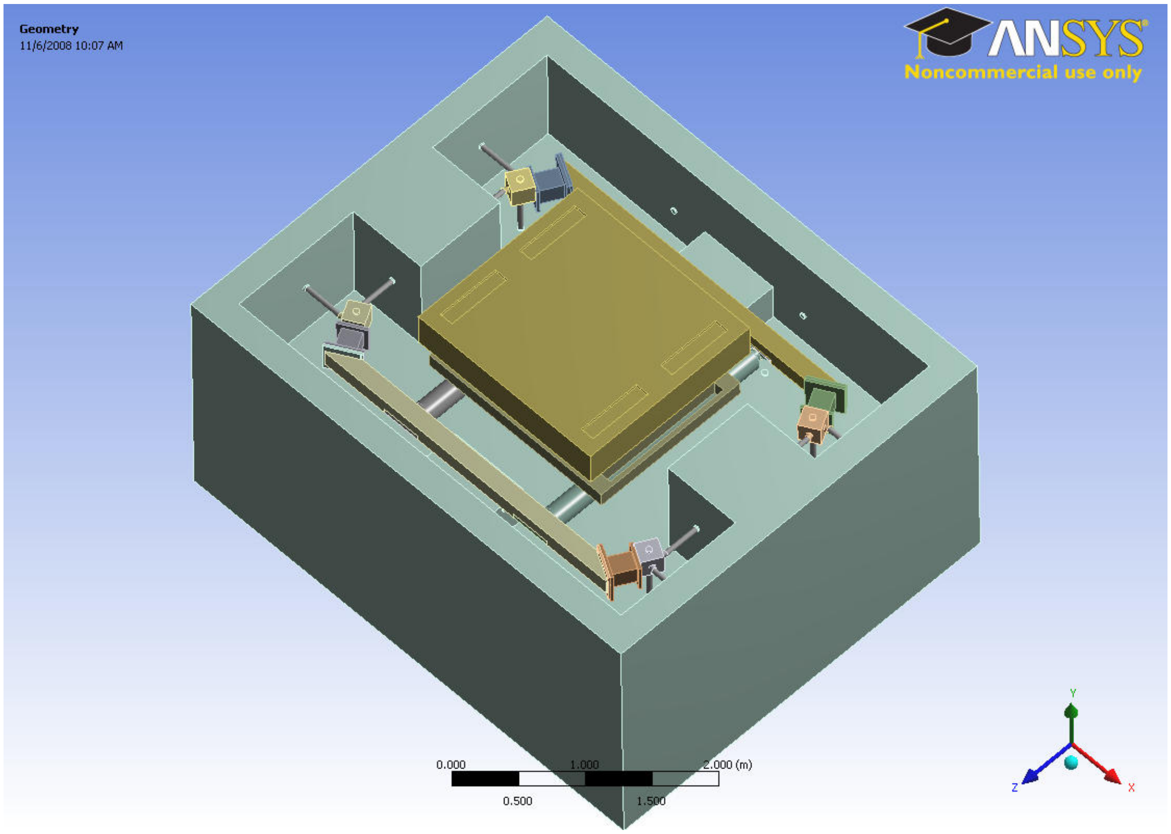


Figure (5) –"Ground" mass shown along with the Modal Mass and the 12 springs representing the 4 Newport Pneumatic Isolators

1.5.3 Matching modal frequency of isolator & calculating the correct damping

Each spring in our model requires a stiffness value and a damping factor, the values used are outlined in the figure below. The vertical stiffness, k quoted is calculated from the data in section 1.4. At this point a modal analysis of the model produced a vertical frequency of ~ 1.6 Hz, agreeing with that calculated in section 1.4 and thus proving the validity of our model to this point.

In order to obtain horizontal frequencies (X and Z) of 1.5 Hz we tweaked the horizontal (X and Z) ‘k” of the pneumatic isolator in the model in order to obtain a horizontal frequency of ~ 1.5 Hz. The numbers are quote in the table below.

The next step was to vary the damping factor until the amplification at resonance in each direction matched those published by Newport, please refer to appendix 1. The damping factors, b quoted below are these matched values.

	k (N/m)	b (N.s/m)
X		
Pneumatic Isolator	38000 *	600
Y (Vertical)		
Pneumatic Isolator	35530	750
Z (compliant direction of the bellows)		
Pneumatic Isolator	38000 *	600

*Figure (6): - Values of k and b for the Pneumatic Isolator * In order to obtain horizontal frequencies (X and Z) of 1.5 Hz we tweaked the horizontal (X and Z) ‘k” of the pneumatic isolator in the model in order to obtain a horizontal frequency of ~ 1.5 Hz.*

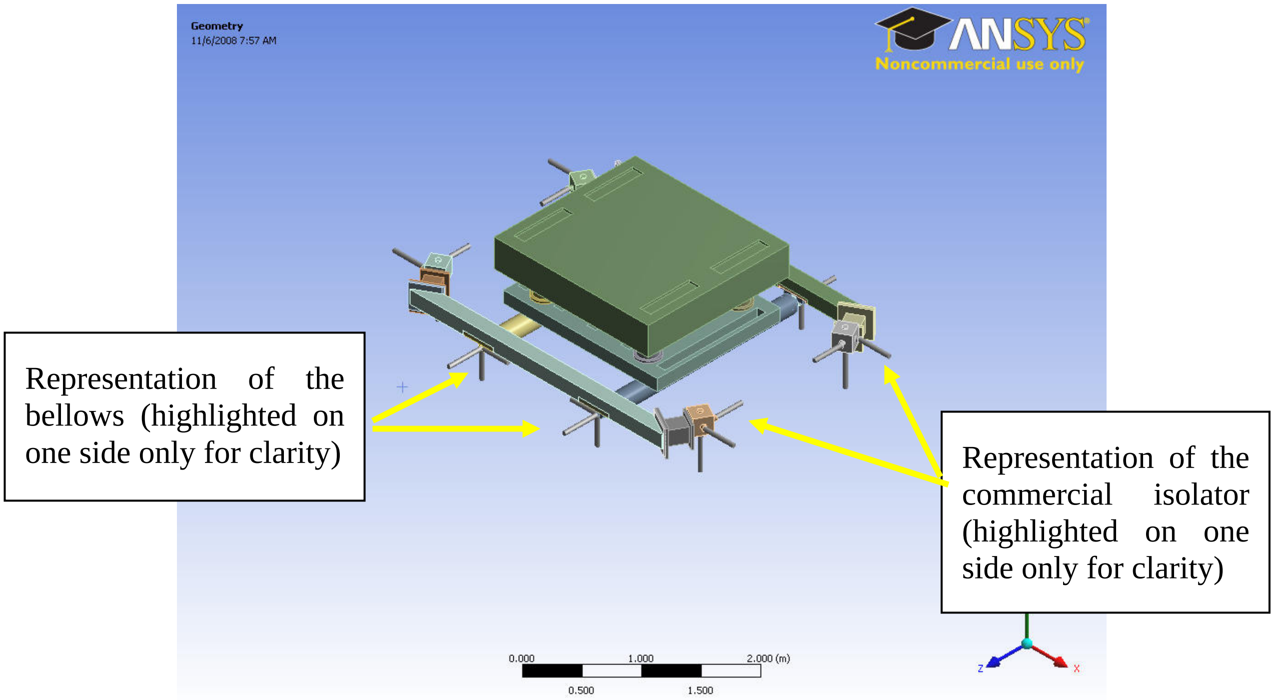
1.6 Adding the HAM bellows

We used the bellow spring stiffness from LIGO-T980123-A. It should be noted that these values are for the measured BSC bellows. However, since we do not have data on the HAM bellows we will use the (higher) BSC data for this analysis. These numbers are shown below along with an estimated value for the damping factor, b.

	k (N/m)	b (N.s/m)
X		
Bellow	43658	1e-4
Y (Vertical)		
Bellow	43658	1e-4
Z (compliant direction of the bellows)		
Bellow	6017	1e-4

Figure (7): - Values of k and b for the Bellows

Figure (8) shows the representation of the bellows as (12) spring/dampers along with the "ground" mass, the Modal Mass and the 12 springs representing the 4 Newport Pneumatic Isolators. It should be noted that in each direction the springs and bellows are added symmetrically.



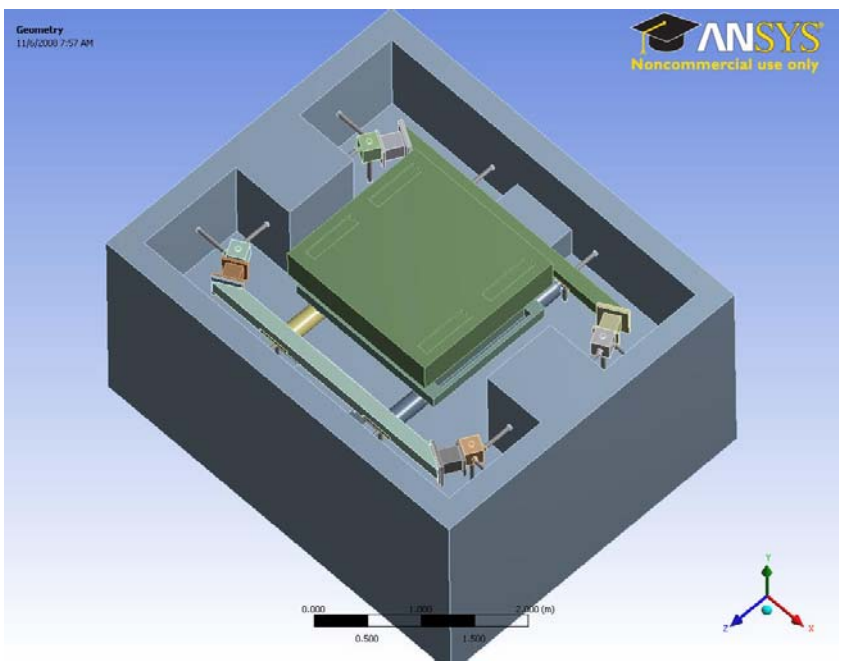


Figure (8) (i) Upper image shows 24 springs - 4 sets of 3 for bellows and 4 sets of 3 for the Newport Isolator – and modal mass. (ii) Lower image shows 24 springs, modal mass and "ground" mass are also shown

1.6.1 Modal Analysis (Isolator and Bellows)

The following set of modal frequencies was obtained from the FEA analysis of the modal mass connected to ground via the 2 sets of springs representing both the pneumatic isolators and HAM bellows. The "ground" mass surrounding the modal mass was clamped with a fixed support on all of the external surfaces; the clamped area is shown in blue below in figure (9).

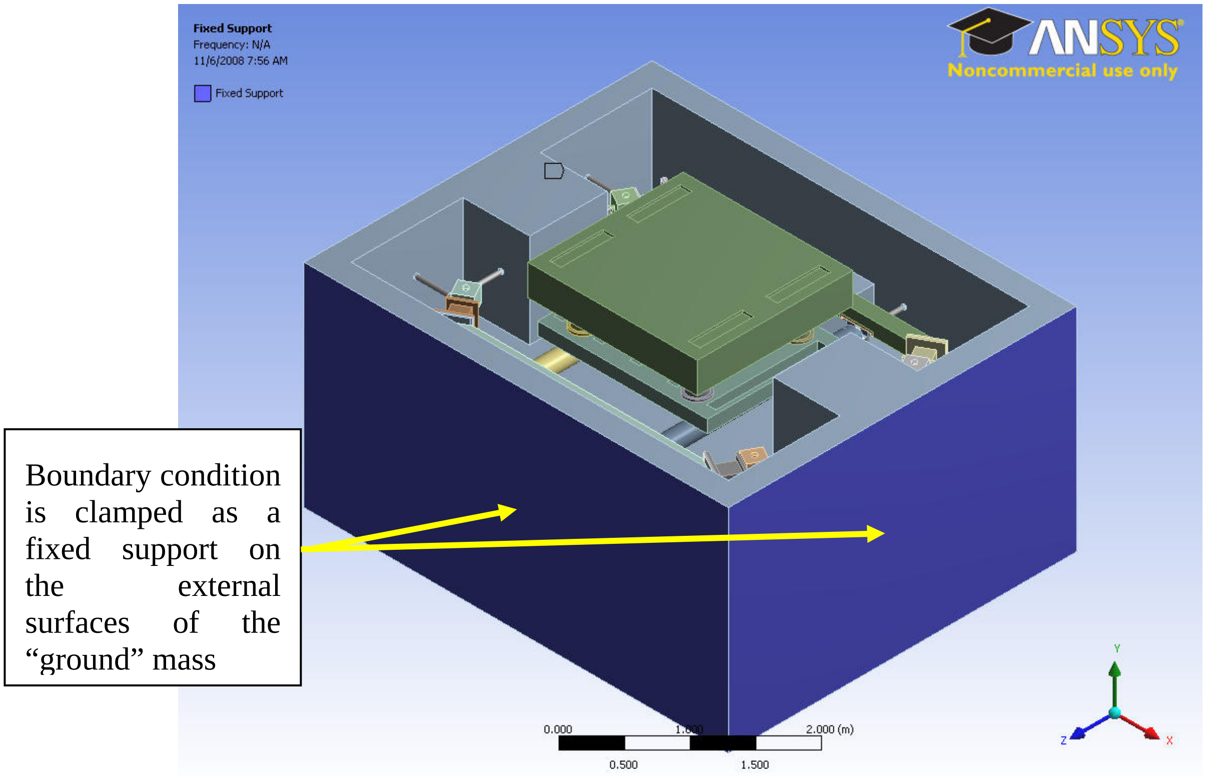


Figure (9): - In the above image of the model the area (all external surfaces of "ground" mass) clamped with a fixed support are highlighted in blue.

The analysis was carried out up to around 50 Hz and all of the modes obtained are shown in figure (10) below.

#	Mode Frequency (Hz)	Mode Shape (Description)
1.	1.6543	Transverse (Z)
2.	2.2203	Vertical (Y)
3.	2.2546	Longitudinal (X)
4.	3.4621	Yaw
5.	15.339	Tilt (Tilt in X)
6.	16.397	Roll (Tilt in Z direction)
7.	34.033	Crossbeam bending
8.	46.232	Crossbeam bending

Figure (10): - Mode Frequencies from FEA analysis described above Please reference appendix 2 for images of the mode shapes.

1.6.2 Using Harmonic Analysis to obtain Transfer Function

In order to obtain the transfer function we performed a harmonic analysis in each direction using the setup described in the modal analysis above and applying a force to the ground. Using the frequency response of both the table and the ground we obtained the transfer function in each direction up to 40 Hz, these are shown below in figure (11), (12) and (13). Figure (10) below shows the setup for the harmonic analysis in the X direction with the force applied to the ground shown. The areas on both the table and the ground used for the frequency response are also shown. The fixed support of the "ground" mass is not shown.

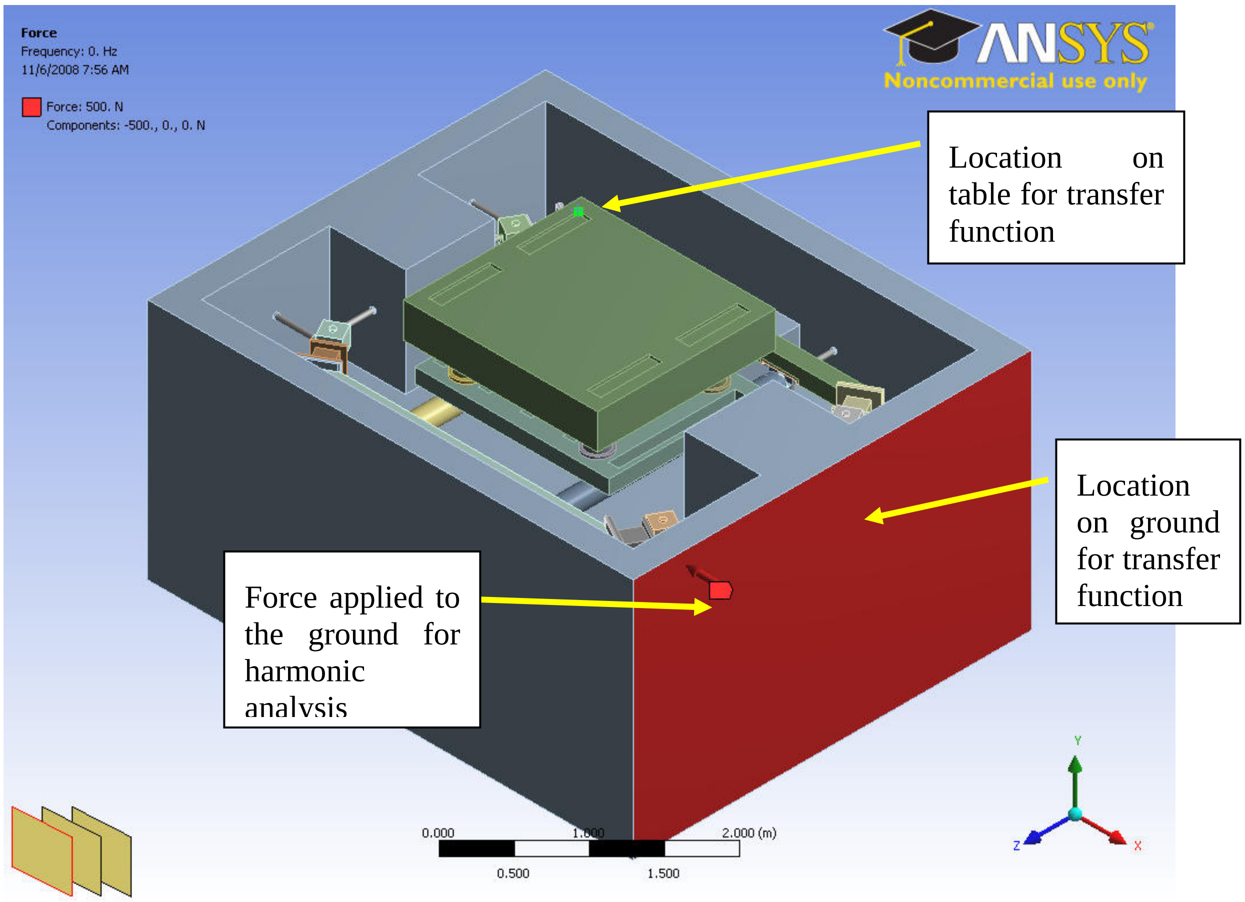
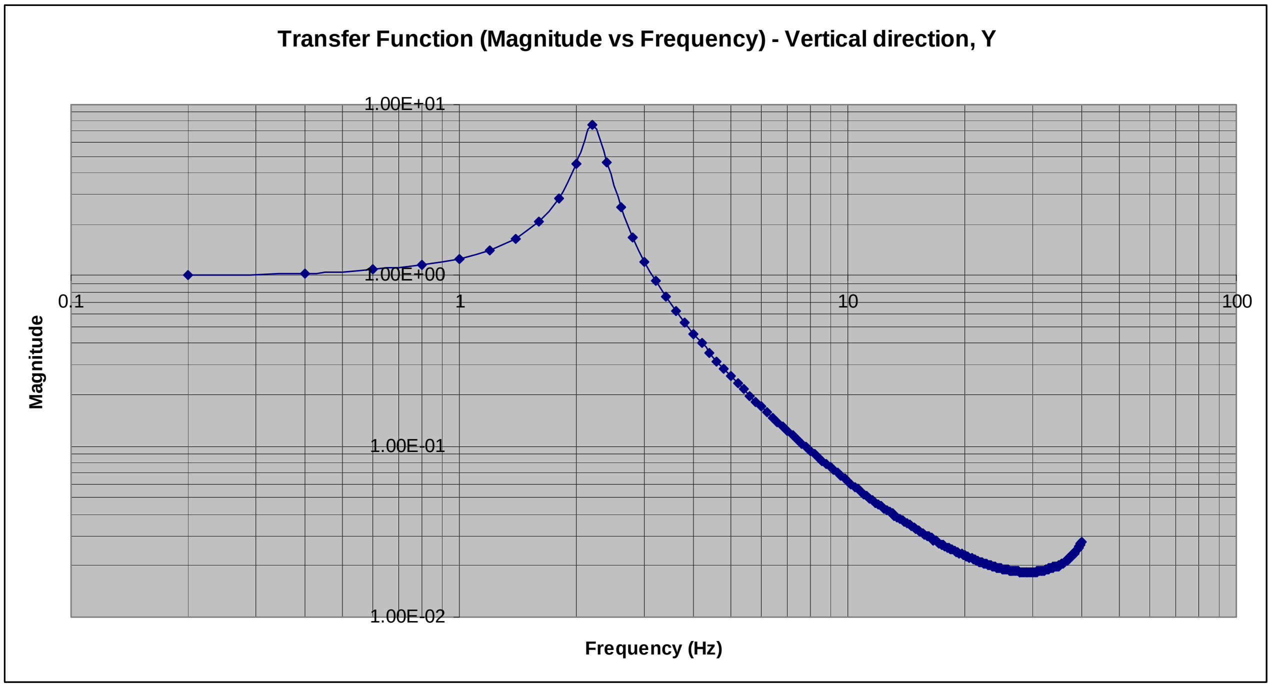


Figure (10): The setup for the harmonic analysis in the X direction with the force applied to the ground shown. The areas on both the table and the ground used for the frequency response are also shown. The fixed support of the "ground" mass is not shown.

1.6.3 Transfer function magnitude and phase



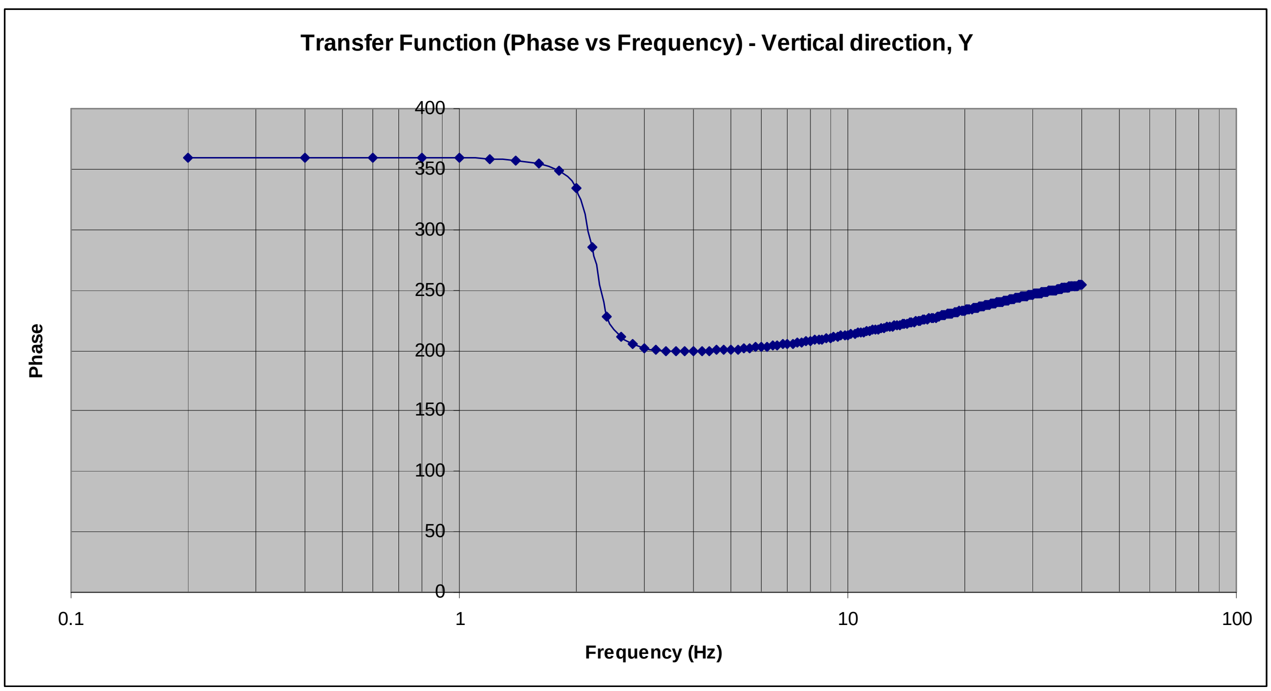
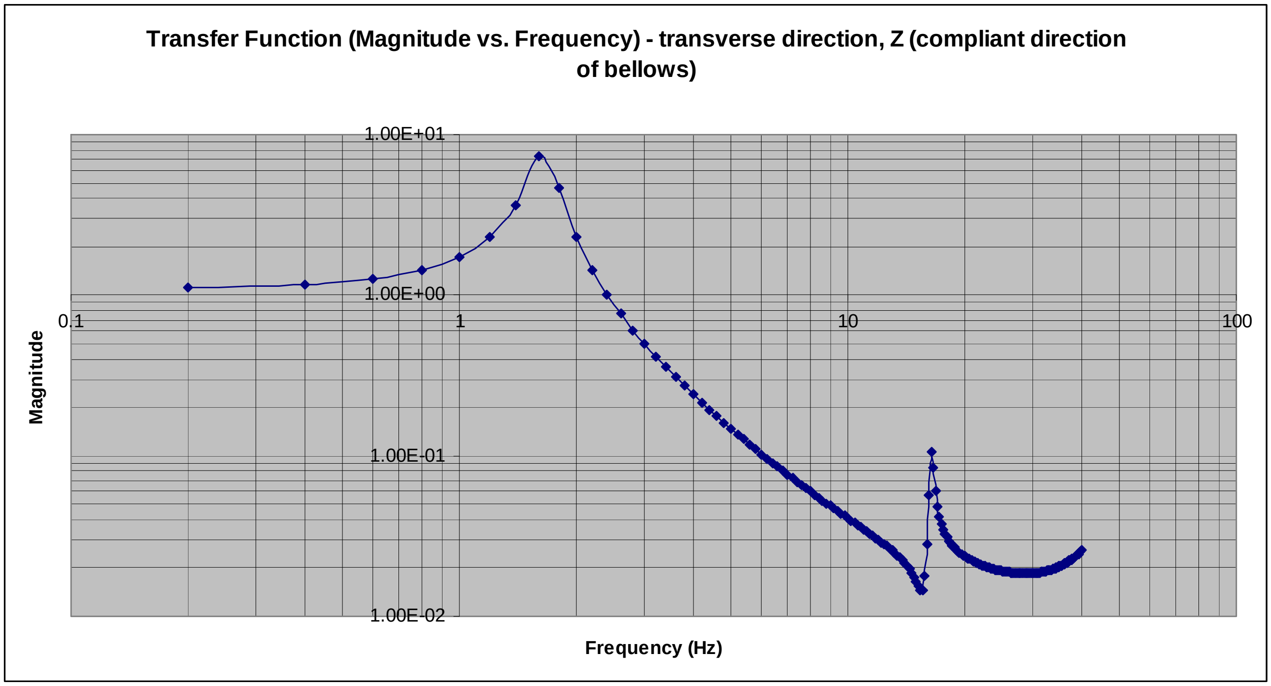


Figure (11):Transfer Function (magnitude and phase) – vertical direction, Y



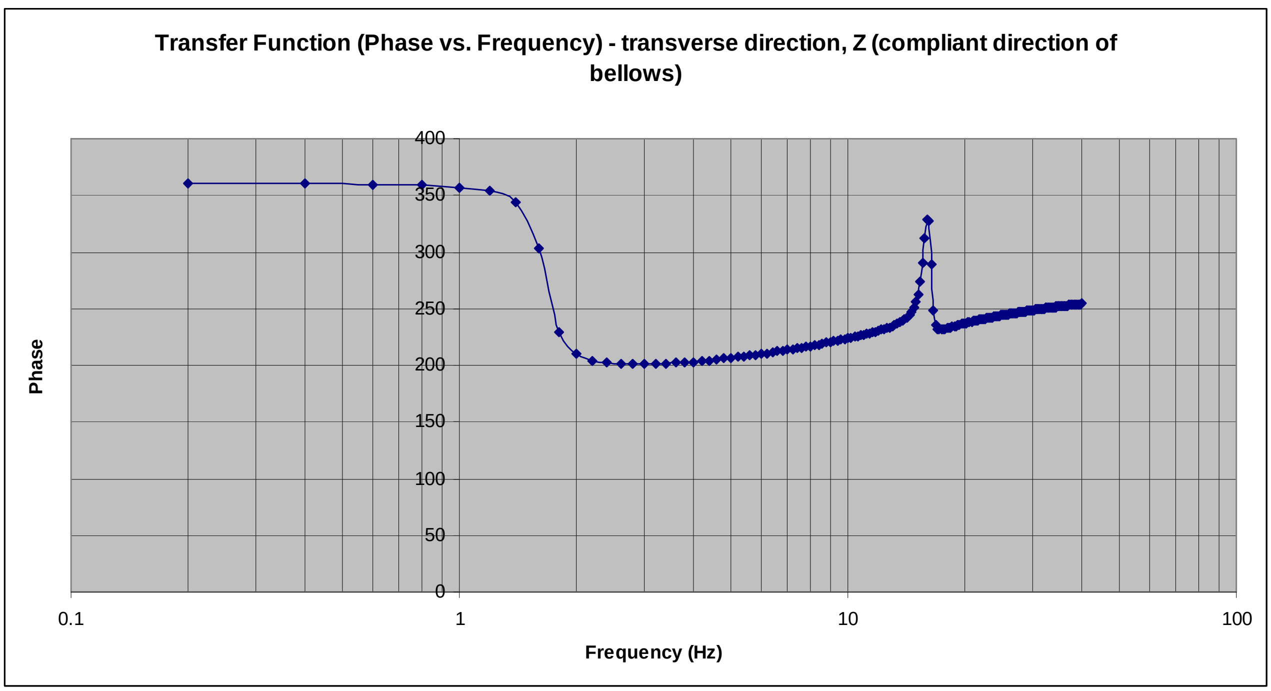
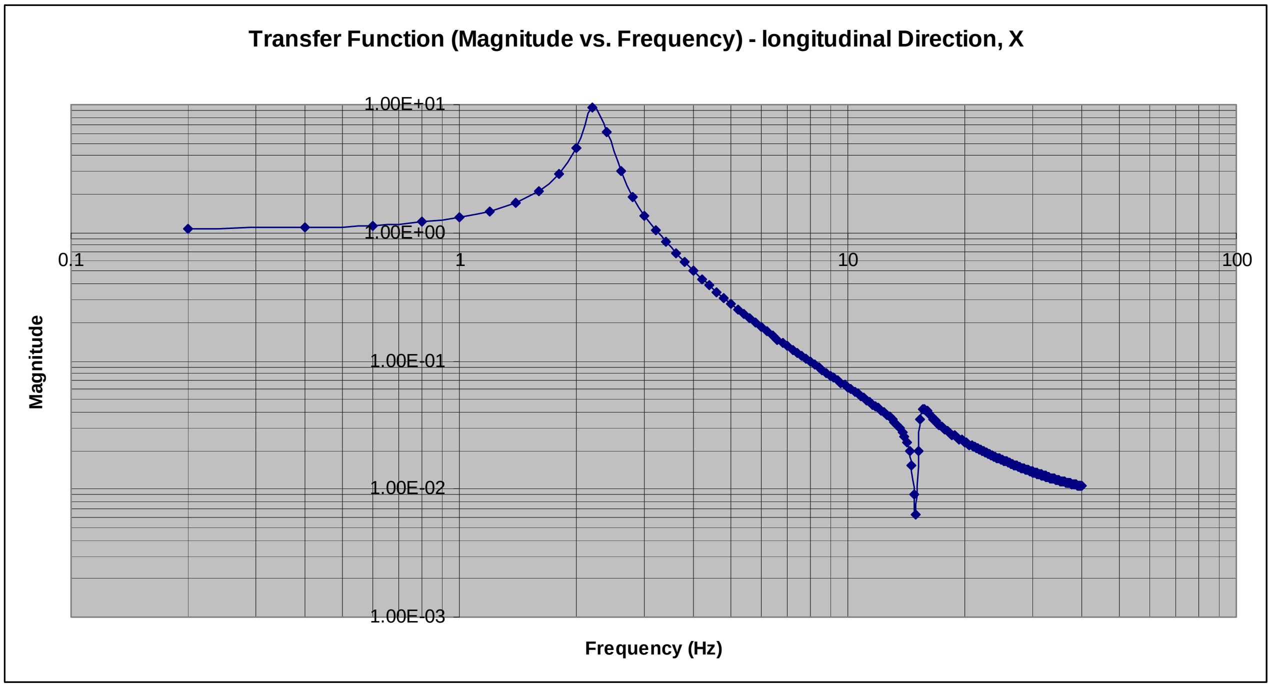


Figure (12): Transfer Function (magnitude and phase) – Transverse direction, Z – compliant direction of bellows



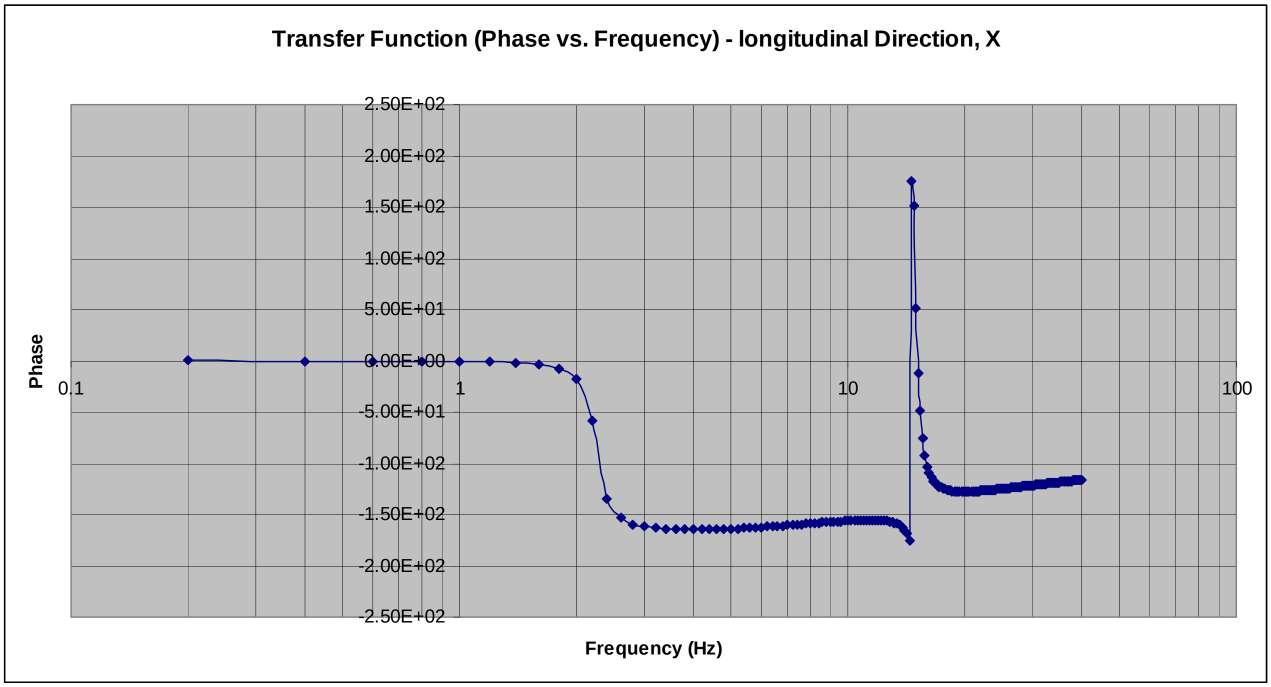


Figure (13): Transfer Function (magnitude and phase) – longitudinal direction, X

1.6.4 Conclusions

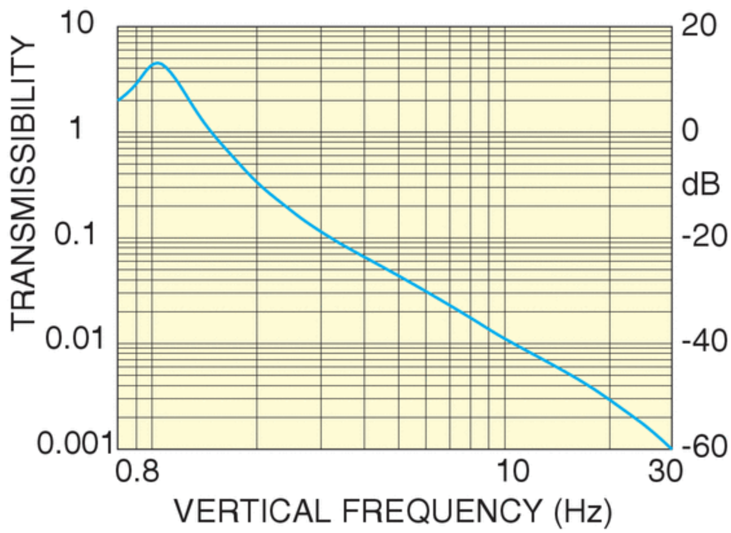
Although there are number of significant approximations in the analysis above, we can draw some conclusions:

1) To achieve as low an isolation frequency as possible, requires the addition of huge blocks of steel atop each of the four isolators. This is likely not impractical.
2) The stiffnesses of the bellows are comparable to the pneumatic isolator stiffnesses, which will result in significantly higher resonant frequencies, reduced isolation at 10 Hz and higher Q at resonance.
3) It seems likely that the isolation performance at 10 Hz will only marginally meet the desired isolation factor of 10 and the amplification of low frequency ground noise at resonance may be significant.

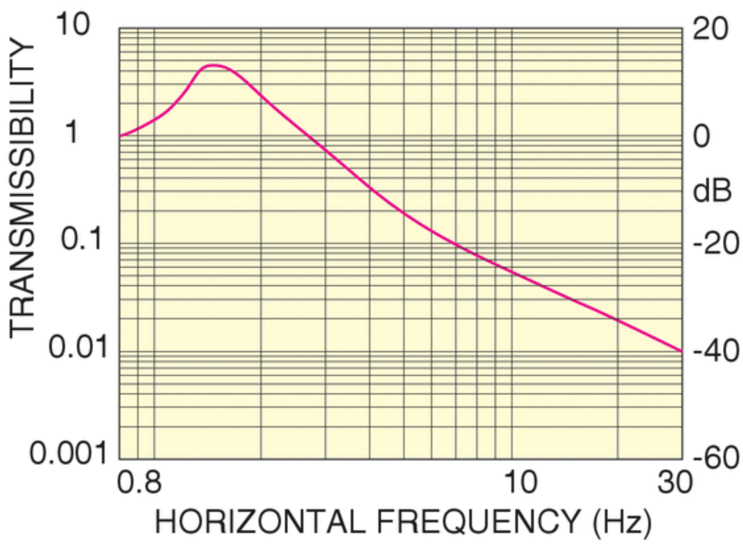
In addition, a simple pneumatic isolator system does not provide alignment control for the optics table or positioning control relative to the other isolated tables. Alignment functionality could be provided with tip/tilt steering mirrors on each optical beam, but the capability to maintain alignment control of the optics table seems prudent.

Appendix 1

The following transfer functions were obtained from the Newport Web-site for I-2000 series assuming 900 kg per leg.



I-2000 Vertical Transmissibility (book value)



I-2000 Horizontal Transmissibility (book value)