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THE JOY OF VIBRATION MITIGATION

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Abstract

As part of the Diamond-II facility upgrade, a new Optics Metrology Lab has been built at Diamond Light Source. This replaced the old lab which will be demolished to make space for a "flagship" beamline. However, the location for the new lab has intermittent 100 times higher floor velocity in the range 50-150 Hz compared to the original. This paper describes the engineering developments to successfully mitigate vibrations within the new lab. The raft of measures includes: 'skyhook' damping (i.e. active damping using geophone velocity feedback) and novel 2-stage passive vibration isolation. New vibration isolation & damping systems have been installed and will enable ultra-sensitive metrology tests to continue in the new lab.

INTRODUCTION

Diamond Light Source is the UK's national synchrotron light facility. Each beamline uses a range of optics to focus and monochromate the ultra-intense X-ray beams created by the synchrotron. Prior to beamline installation, all X-ray optics are assembled and characterised in the Optics Metrology Laboratory (OML1). This cleanroom lab contains a suite of state-of-the-art metrology instruments to measure X-ray optics with sub-nanometre precision. These sensitive instruments require a mechanically and thermally stable environment. After > 15 years of operation, OML1 is to be demolished to make space for a new flagship beamline. To continue optical metrology operations and prepare for the improved-quality X-ray optical systems required for Diamond-II beamlines, a new lab (OML2) has been built. However, due to space limitations within the Experimental Hall, the location for OML2 has an intermittent 100 times higher floor velocity in the range 50-150 Hz compared the original. Such vibrations are caused by nearby plant, including a large, motorised dewar store. The engineers were given the task of finding isolation solutions to mitigate these increased levels of disturbances and provide an ultra-stable environment for the optical metrology instruments. Commercial passive vibration isolated tables only provide transmission data over a limited frequency range (e.g., Newport™ S-2000A from 0.8 to 30 Hz) and the supplied plots look like simple 1D lump mass models. Active damping options were also investigated, but they were not considered to be cost-effective, or provide the required damping bandwidth. Therefore, in-house damping solutions were designed and built.

PASSIVE VIBRATION ISOLATION

To replace existing air-isolation optical benches, a concept using spring isolators (from Farrat) was developed. The design re-used unwanted optical breadboards, which were supported via an intermediate granite block with two isolation stages i.e. floor to granite, and granite to table. This provided an attachment method that did not over-constrain the breadboard as well as providing a steeper isolation slope with frequency, as shown in Fig. 1. The performance of the vibration isolation system is significantly different from a simple 1D lump mass model since the higher-order vibration modes cross-couple. Rotation modes are measured as translations with amplitudes that depend upon the modal lever [1]. A 3D modal analysis of the proposed design was performed using ANSYS software to both visualise the mode shapes and to generate a reduced order model (ROM) for input into Simulink®. The sensor location was close to, but not exactly at, the centre of the table, as with the measured data. This prevents symmetry from making modes unobservable. The simulated vibration transfer function depends upon the point of measurement and the variation of spring rate across isolators. A random distribution of a realistic ± 10% spring rate causes higher-order modes to become more significant.

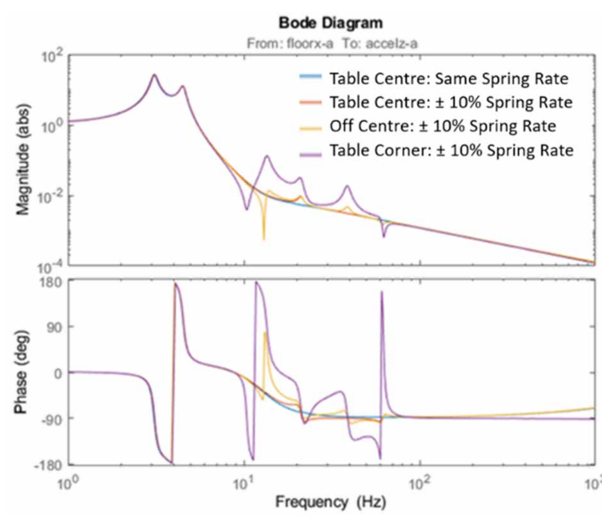

Figure 1: Simulated floor to tabletop transfer functions showing the effect of varying the spring stiffness and sensor location.

Passive Vibration Test Results

A measured transmissibility plot of the installed table, to compare with Fig. 1, is given in Fig. 2. The very low coherence is caused by the acoustic disturbance which is comparable to the floor vibration. Significant energy is passing through the air which is not measured by the floor accelerometer corrupting the transmissibility ratio. That said the measured data does show the simulated 2 resonant peaks below 10 Hz which amplify the floor acceleration and vibration isolation above 6-8 Hz.

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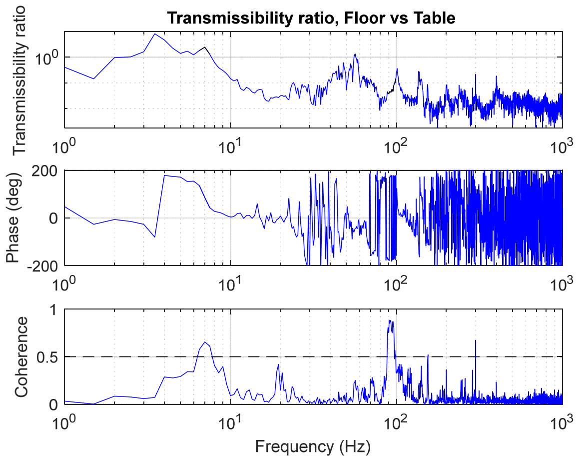

Figure 2: Measured accelerometer transmissibility plot from the floor to the table top. The low coherence indicates that there is no correlation between floor and table motion at those frequencies

To give a clearer view of the isolation performance an integrated displacement plot which runs from high to low frequency is presented in Fig. 3. The data plotted in blue above with the dewar store running is the same colour below.

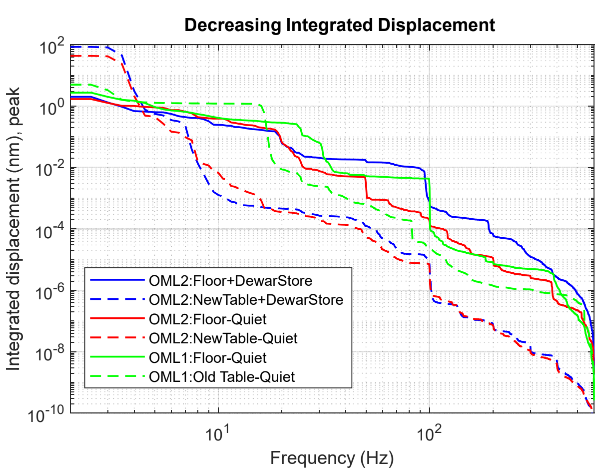

Figure 3: Measured accelerometer data plotted as an integrated displacement running from high to low frequency to compare the new table performance with the old commercial solution

The ratio of floor to table spectra does not make sense when significant energy is also passing through the air. The effect of the turning off the lab air-conditioning was clearly demonstrated in preliminary tests in OML1, when the peak sound level was reduced from 60 dB to 55 dB, causing the table peak acceleration to reduce by an order of magnitude.

The region of interest i.e., the typical equipment resonant frequency range coincides with the increased floor vibration over the 50-150 Hz range. The new table with dewar store disturbance, exhibits an isolation factor of over 200 at frequencies above 10 Hz. The new table design is significantly more stable above 10 Hz than the old commercial air isolation table in OML1.

ACTIVE DAMPING

The Diamond-NOM (Nano Optical Metrology) is a non-contact, slope profiler capable of characterizing optical surfaces with sub-nm repeatability. Air-bearing stages are used to translate the autocollimator beams across the surface under test [2]. As supplied by the manufacturer (Q-Sys), the granite base was supported on steel wedges with an elastomer layer. The original installation did not exhibit any significant vibration isolation, and so was replaced with Farrat spring mount isolators. The new passive vibration isolation was effective above 6 Hz. However, an undesirable effect was that the acceleration reaction forces from the scanning stage caused the granite system to resonate. Oscillation at ~2 Hz created secondary reaction forces back to the scanning stage, corrupting the dynamic, measurement of X-ray optics.

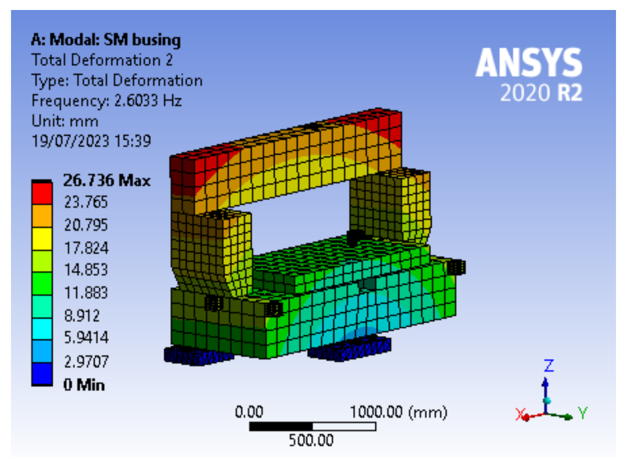

Figure 4: ANSYS modal analysis showing the dominant resonant mode driven by the scanning stage reaction forces.

Options considered to mitigate the resonance were: reduce the stage acceleration (at the expense of measurement throughput); modify the system by adding a balancing mass (but this would be a complex task for a commercial system with a granite structure); add additional passive dampers (which would corrupt the vibration isolation); or the chosen solution of developing an active damping system.

A ROM was exported from ANSYS see Fig. 4. The ROM was used as the plant block in a Simulink model to capture the 6 DOF (Degree of Freedom) complexity of the lowest 20 mode shapes. A simple ‘skyhook’ damping (i.e., active damping using geophone velocity feedback) was simulated [1]. Disturbance to position transfer functions were exported to create a dynamic error budget (DEB) [3]. It may be seen from Fig. 5 that neither amplifier current noise (Trust Automation TA105), 16-bit DAC quantisation noise, nor the geophone noise, limit the system stability. It was critical that the active system did not worsen the stability. The floor motion trace is hidden under the total trace.

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Above 100 Hz, a first order low pass filter was added as the sensor noise became dominant.

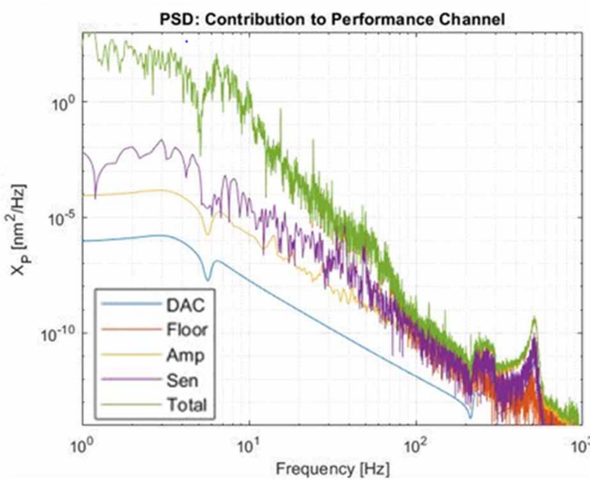

Figure 5: Dynamic error budget showing that the amplifier, sensor, and DAC noise would not degrade the position stability

Active Damping Test Results

The apparatus for the active damping system is shown in Fig. 6. The system implemented was a single axis, however it effectively damps multiple modes. The contact point with the granite was carefully chosen to maximise the number of modes which could be controlled. The measurement and actuator systems were closely located to enable closed-loop stability.

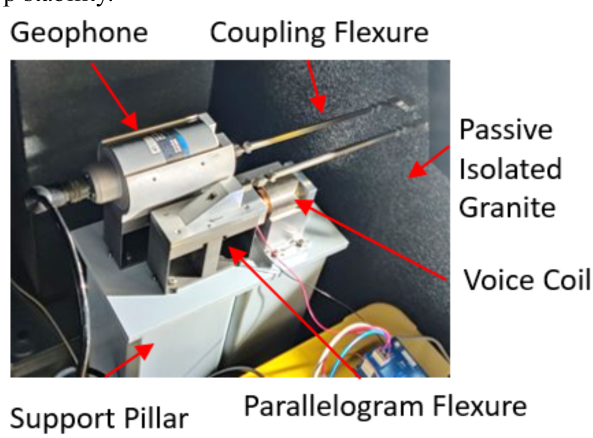

Figure 6: The main apparatus of the active damping system for the Diamond-NOM slope profiler.

The measured performance was in good agreement with the simulation, i.e., the resonant frequencies and vibration decay times were very similar. Representative active damping results are shown in Fig. 7. The damping ratio has improved by an order of magnitude, from 0.009 to 0.24, and settling time (to within 5%) are reduced from 47 s to 2.4 s.

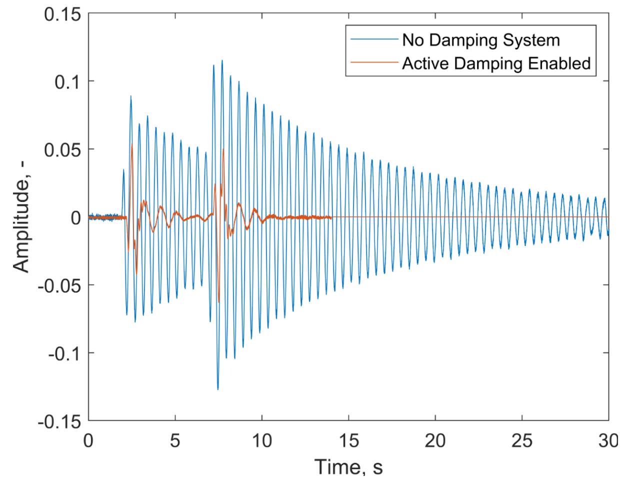

Figure 7: Accelerometer scans (velocity versus time) to show the effect of moving the scanning stage of the Diamond-NOM, with and without the active damping system.

CONCLUSION

The active and passive isolation systems were successful in reducing vibrations transmitted to the metrology instruments in the new Optics Metrology Lab (OML2).

- The passive system successfully isolated the floor vibrations, leaving acoustics as the dominant disturbance force.
- The active damping system applied to the Diamond-NOM reduced the settle time between steps by an order of magnitude. This enables scanning to be performed more quickly, increasing measurement throughput while maintaining the passive vibration isolation.

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