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EXPERIMENTAL MODAL ANALYSIS VIBRATION MEASUREMENT TO INFORM ENGINEERING DESIGN

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Abstract

Experimental Modal Analysis was performed on an existing 5 degree of freedom mirror system on beamline I08 at the Diamond Light Source, by impacting the structure and measuring the response at locations of interest. Commercial software was used to generate the frequency response functions and mode shape animations. This experimental information was used to inform and optimise a design iteration for a new mirror system. The new mechanism was designed, installed and tested on the J08 branch line at The Diamond Light Source to validate the expected improvements in stability, stiffness and resonant frequency. The mirror system fundamental resonant frequency was significantly increased from 20 Hz to 49 Hz.

INTRODUCTION

Experimental Modal Analysis is a technique which allows a mechanical system's vibration mode shapes to be identified, quantified and visualised. This information can then be used to inform future design & operation. It is commonly used for everything from aircraft to tennis rackets. Modal Analysis was used at the Advanced Photon Source during the development of new magnet support structures for example [1]. There are numerous commercial systems based upon different data capture methods from large scanning laser Doppler vibrometry systems [2] to single accelerometer techniques [3].

5 Degree of Freedom Mirror System

The I08 X-ray Microscopy beamline at the Diamond Light Source uses 3 mirror systems with identical supports. The new J08 branchline project required a new mirror system. Rather than order an identical copy of an existing mirror system, the opportunity for a design iteration was taken. The original systems which were procured from Instrument Design Technology Ltd.™ (IDT), have been in service since 2013. They have operated without fault and exhibit excellent thermal stability. The design uses an orthogonal system of slides, rails and vertical jacks to provide the 5 degrees of freedom (DOF). The mirror system moving platform is mounted without over constraint. A spherical bearing forms the "cone", a single free rail & spherical bearing forms the "vee" and 2 crossed free rails & a spherical bearing forms the "flat". A horizontally deflecting mechanical mirror bender is mounted within the vessel with a fine pitch piezo mechanism incorporated into the support. The vessel is pumped by a relatively small 200 l/s ion pump, mounted in top of the system, as there was no space underneath.

An image of the system under test is given in Figure 1. Up-stream of the mirror is a diagnostic vessel and downstream is the radiation shutter.

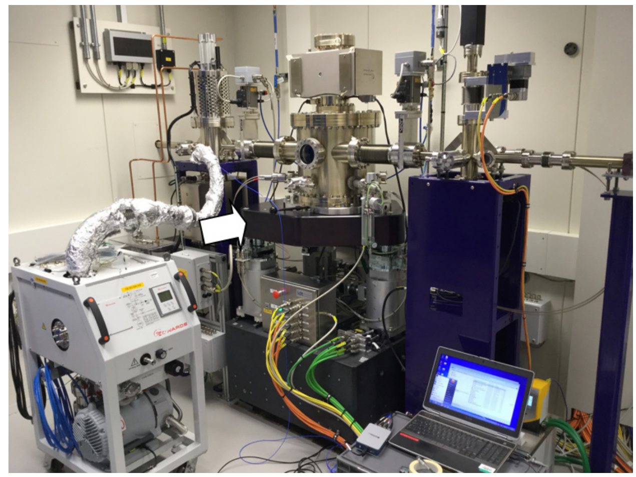

Figure 1: Mirror System I08-M4 at the Diamond Light Source. Impact location indicated by arrow.

Data Capture & Processing

The data presented within this paper was taken using a single triaxial accelerometer (PCB™ LW214478), an impact hammer (Bruel & Kjaer™ 8206), a 4 channel signal amplifier (National Instruments Ltd.™ cDAQ-9171) and software from M+P International Ltd.™ (Analyzer 5.1.0 software [3]).

The method chosen for this study was to manually impact the system at the same point and move the triaxial accelerometer around. The impact location was chosen to drive the resonant mode shapes most likely to cause X-ray beam motion. The direction and location are indicated in the image above (Figure 1). This force vector drives both horizontal and rotation modes of the horizontally deflecting optic. A course 3D model of the mirror system was created with node locations either corresponding to physical data capture locations or simply following data capture locations. For example, the granite block was modelled with 4 data points on the top surface close to the corners with an extra 4 virtual slave points to form the bottom surface. A total of 26 data points (5 impact average per point) were measured to give a clear picture of how the mirror system base, jacks, moving platform, vacuum vessel ports, ion pump and neighbouring vessel ports move. The software processes the accelerometer data using the time, amplitude and phase information from the calibrated impact hammer to create frequency response functions (FRF) & mode shapes. The modal analysis 3D model is shown below (Figure 2). Multiple data points were taken on the

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jacks & slides to enable the clear observation of the motion contribution of each rail & slide.

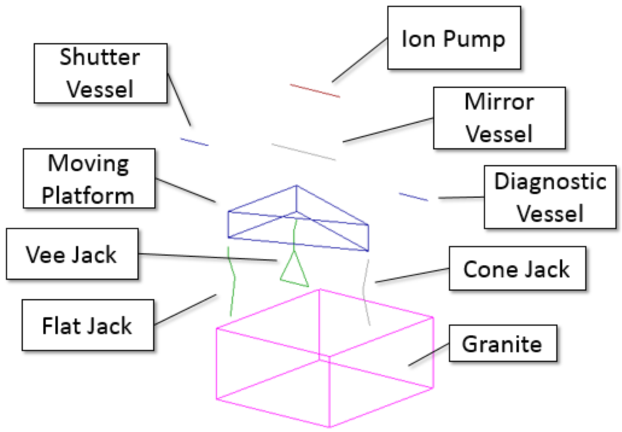

Figure 2: Modal Analysis Model of the mirror system within the M+P software.

I08-M4 MODAL ANALYSIS RESULTS

The software plots the frequency response functions (FRF) of the requested locations and gives a mode shape animation for the selected frequency. The FRF is a graph of measured acceleration per unit impact force vs. frequency. The I08-M4 system FRF for the impact location is given as the blue trace in Figure 3.

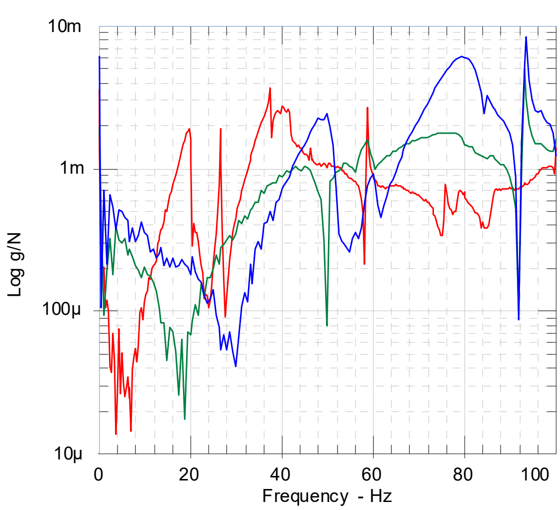

Figure 3: Impact Test Comparison of X Direction FRF: Red I08-M4, Green J08-M2 Clamps Off, Blue J08-M2 Clamps On.

It is very interesting to note that the physical acceleration of the system varies by more than an order of magnitude for the same peak force. Only the frequency is different.

The mirror system fundamental vibration mode was a translation of moving platform, normal to the optic at 19.7 Hz. Only a modal analysis will show you that the peak in the FRF is a translation rather than a rotation. An image of the mode shape is given below in Figure 4. There are a number of points to note about this image. The granite and neighbouring vessels are stationary at this frequency. The "cone" and "vee" jacks are pivoting at their base around the preloaded ballscrew slides; whereas the "flat" jack is translating on its free slide. The mirror system vacuum vessel ports are translating less than the moving platform i.e. being restrained by the bellows. The ion pump is rocking with a larger amplitude than the moving platform due to the addition of the rotation component around the jack bases.

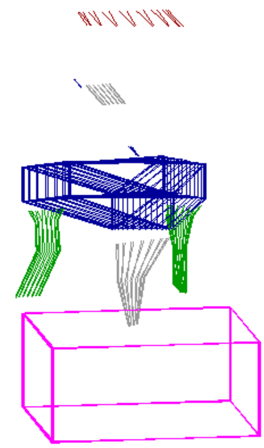

Figure 4: Fundamental Vibration Mode Shape at 19.7 Hz, looking down-stream.

A large table of system resonant characteristics was created using this quality data. The frequencies ranged from the shutter resonance at 14.4 Hz up to the granite block resonance at 111.3 Hz. It is perhaps surprising that even on the nanometre level, the preloaded rails are rolling i.e. the vibration force is greater than the friction.

A second example mode shape is given below (Figure 5) to show the effect of the ion pump resonances on the mirror stability. At 58.75 Hz the ion pump resonated with minimal damping dragging the moving platform with it.

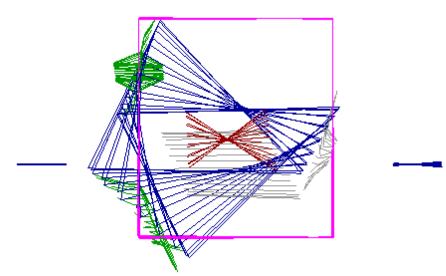

Figure 5: I08-M4 Ion Pump Resonant Mode at 58.75 Hz, plan view.

Modal Analysis Design Conclusions

The author created a list of design suggestions based upon the experimental modal analysis data. IDT used this to perform a low risk design iteration, building on the existing good performance.

1. Move the ion pump to the diagnostic assembly to reduce the moving mass & remove a resonator
2. Integrate pneumatic slide locks to fix the kinematic system during data capture
3. Redesign the system to use rails with a higher moment stiffness to increase rocking resonance frequencies

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4. Reduce the mass of the motion platform
5. Recess the spherical bearings into the moving platform to reduce leg length
6. Reduce the jack travel & hence length to optimise for this application
7. Mount the granite firmly to the concrete floor

J08-M2 MIRROR SYSTEM

The new improved system was delivered by IDT and installed at the Diamond Light Source on the J08 branch line. An image of the system is given below in Figure 6.

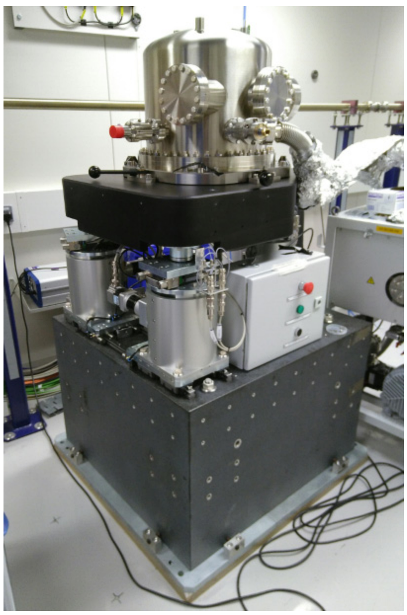

Figure 6: J08-M2 Mirror System from Instrument Design Technology Ltd.

J08-M2 Mirror System Vibration Tests

The FRF of the new system was measured in the same way as for the original system. The data is presented in the same plot for direct comparison in Figure 3. The original fundamental of 19.7 Hz compared well with systems from other commercial suppliers. The new system resonant frequency has been pushed up to 45 Hz without clamps and 50 Hz with clamps engaged. This is particularly impressive as the resonant frequency is proportional to the square root of the stiffness [4]. The system moving mass was reduced by a factor of 0.6 and the stiffness was increased by a factor of 3 (Assuming a simple spring mass system). It is also clear from the FRF that the width of the fundamental and hence damping is significantly increased. The damping must be caused by the increased rolling element preload and contact areas. The high resonant frequency enables the support to accurately track the synchrotron floor motion and the high damping minimises the amplitude at resonance.

The new and the old systems are only mounted 2 m apart which allows a direct comparison of background stability. The system position has been calculated by a double integration of the accelerometer data. As may be seen from Figure 7 the old system position oscillated with an amplitude of ~ 100 nm whereas the new system stability is at the background vibration level of ~ 20 nm (Measured using PCB™ 393B31). Both traces nicely illustrate how the largest amplitude motion of the floor is at the lowest frequency with the beamline optics moving together.

The slide clamps clearly stiffen the system and reduce the damping at resonance under a FRF impact measurement. However, in practice the background stability data shows little difference.

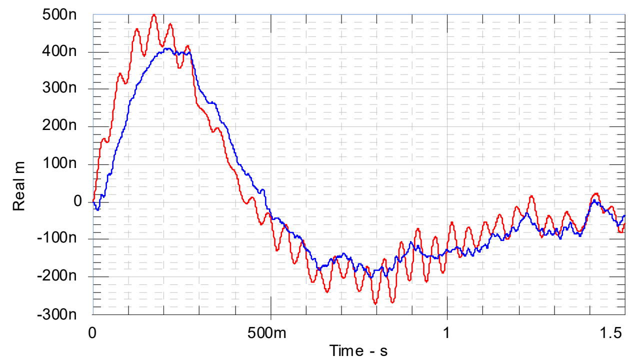

Figure 7: Comparison of Mirror System stability: Red = I08-M4, Blue = J08-M2.

CONCLUSION

An ultra-stable mirror system has been engineered, installed and tested using the experimental modal analysis technique to inform the design evolution of an existing system. The fundamental vibration mode of the system is ~ 45 Hz with the clamps off and ~ 50 Hz with the clamps on. The J08 system will form a stable focus for the J08 endstation.

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