Optimal Dynamic Performance for a High-Precision Stage System

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Abstract

Scientists at the Advanced Photon Source (APS) are continually stretching the limits of x-ray measurement techniques. To this end, mechanical structural dynamic stability is of primary importance. System dynamic performance of multidimensional stage groups, such as those found in optical instrument positioning systems, is dependent upon both individual component behavior and the system configuration. Experimental modal-analysis techniques have been implemented to determine the six-degree-of-freedom stiffnesses and damping for individual stages and groups. The key to the approach is a combination of experimental measurements and numerical simulation. Experimental dynamic-performance data, such as displacement, dynamic stiffness, and damping, are used in computer models to home in on problem areas faster than either a computational approach or experimental approach alone would allow.

In this paper, two recent examples of this methodology are presented: the diagnosing of a multiaxis goniometer installed in the 2-ID-D experimental station, and the investigation of actuator/stage group optimization for multiaxis positioning systems.

Keywords: high-precision, dynamic stability, actuator, stage, goniometer

1. Introduction and Motivation

At the Advanced Photon Source (APS), as at all synchrotron radiation light source facilities, the ability of investigators to carry out their work rests on the assumption that the x-ray beam, the sample it impinges upon, and the x-ray detector remain stable, with known relative positions and orientations. As it requires considerable effort and resources to produce the x-ray beam, maximizing the light use is essential. To this end, instrumentation engineers are introducing automated positioning systems and robots to reduce the need for manual manipulation of optics, samples and detectors and increase the beam utilization. Although this approach enables researchers to generate more data, the introduction of multibody flexible manipulators and massive multiaxis goniometers can confound data acquisition due to the increased structural flexibility of the system. Engineers within the Experimental Facilities Division (XFD) have undertaken projects to improve the structural stability of existing systems and ensure the structural stability of systems currently under design.

A history of work exists to ensure optimal dynamic performance of x-ray components at the APS. It can be divided into three basic areas: ground-motion studies [1-3], existing equipment analysis and remediation [4-14], and new design consideration [10-11]. The current work investigates the design and installation of a viscoelastic damper for a microdiffraction goniometer and the modeling of a recently implemented robotic x-ray sample handling system. The objective of this research is to further the application of multibody vibration and multibody dynamics in improving the performance of support and positioning structures at the APS. Specific goals include identifying component stiffness and damping properties, construction of finite element (FE) multibody *structural vibration* models of the goniometer and Cartesian robot systems, comparison of the models to experimentally measured behavior, modification of the FE models to include proposed changes, and eventual implementation of a large displacement multibody *dynamic* model of the robot system.

A brief introduction to the two systems under investigation is necessary. The first subject is a multiaxis goniometer installed in the APS 2-ID-D beamline hutch. Ground borne and structure borne vibration have been determined to reduce the resolution of the instrument. Fig. 1 depicts the goniometer.

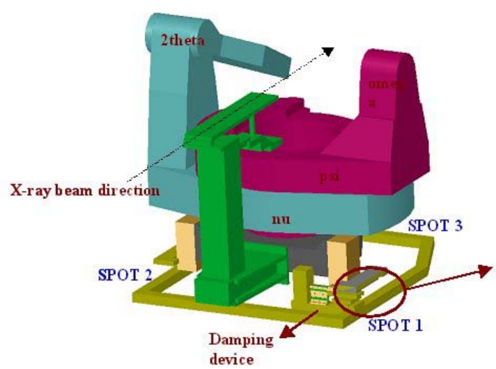

Fig. 1: Multiaxis goniometer and measurement spots.

It is composed of three functional groups: the coarse positioning support structure, sample kinematic support structure, and the detector kinematic support structure. The coarse positioning system also comprises three parts: a support frame, a triad of precision stages, and a granite table. Large displacements and rotations of the goniometer are effected through the vertical, horizontal and longitudinal motion of these stages. The stage groups are located in a triangle under the goniometer, at spots 1, 2 and 3, in Fig. 1. Vibration diagnostics and analysis focused on modeling the stages and their contribution to overall vibration performance, with the goal being the application of an effective viscoelastic damper.

The second subject is a 3-axis Cartesian robotic manipulator, which is to be installed at the APS 2-BM beamline to perform sample positioning for high throughput microtomography research. Fig. 2 illustrates the robot system. The robot consists of three orthogonally oriented linear stages, with an end effector attached to the end of the Y stage. This particular robot will fulfill a pick-and-place type mission, moving samples from a magazine to the target area in the beamline.

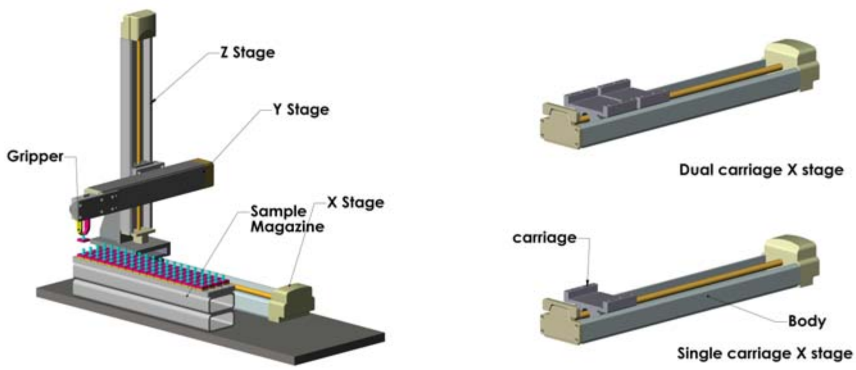

Fig. 2: Cartesian robotic manipulator and X stage comparison.

The investigation centered on determining the modal parameters, mass, stiffness and damping, of the three stages in order to build a computer model that could accurately predict a proposed change to the system configuration. The change is the replacement of the current single-carriage X stage with an X stage that has two carriages, as in the right hand side of Fig. 2. In addition, the mass, stiffness and damping data will be used in the future to construct a multibody dynamic model of the robot, in order to verify behavior during sample transit.

2. Methodology

2.1 Experimental Quantification of Existing Structure

The initial experimental survey of the goniometer consisted of autopower spectrum measurements at several points on the structure and on the floor near the instrument. These measurements were used to define the ambient vibration level and to gain a basic understanding of the dynamic characteristics of the goniometer. In these tests, high-sensitivity, low-frequency, piezo-based accelerometers and a dynamic signal analyzer were used to measure horizontal, longitudinal and vertical accelerations from 0-100 Hz. Testing indicated significant motion in the horizontal direction, with peaks well above the 20 nm baseline scientists would like to see throughout the low-frequency range.

Transmissibility measurements, frequency response functions (FRFs) that quantifie the change in motion from one point to another point, were then used to deduce the suspect components. Each coarse positioning stage group was surveyed by measuring the 0-100 Hz transmissibility between the ground and points within the stage system; both vertical and horizontal transmissibility functions were obtained. We determined that the ground motion was amplified in both the horizontal and vertical directions, although the horizontal direction transmissibility was. Damping was also estimated using the transmissibility FRFs.

Impact modal analysis was then performed in anticipation of building a FE model. Frequency response functions were measured using an impact hammer and the above-mentioned accelerometers. The data were then processed using ME-Scope software to extract natural frequencies, modes shapes and damping. The data were subsequently used to guide the analytical finite element analysis.

Modal analysis was the experimental tool used to experimentally investigate the Cartesian robot. Both the individual stages and the complete robot underwent modal analysis. Once again, FRFs between the force input and the acceleration response output were generated through impact testing. The FRFs were processed using ME-Scope software. Damping and stiffness data for each stage were then calculated using the modal analysis results.

2.2 *Computer FE Models*

A finite element model of the goniometer was built, using the experimental data for guidance and corroboration. Stage stiffness and damping data were also incorporated from previous research [9-12]. Consolidating the complex structure above the granite table resulted in a lumped mass FE model, as shown in Fig. 3. Modal analysis and harmonic analysis of the model were used to investigate the addition of the viscoelastic damper, making it possible to determine the proper damping coefficient.

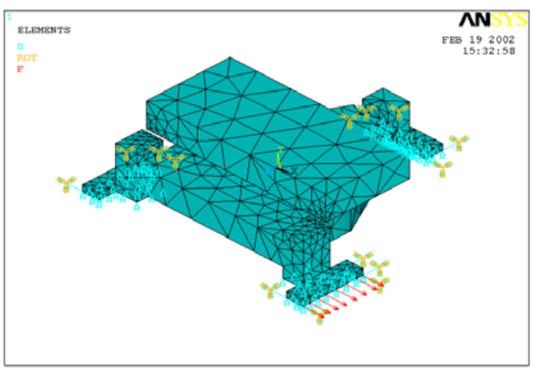

Fig. 3: Goniometer FE model.

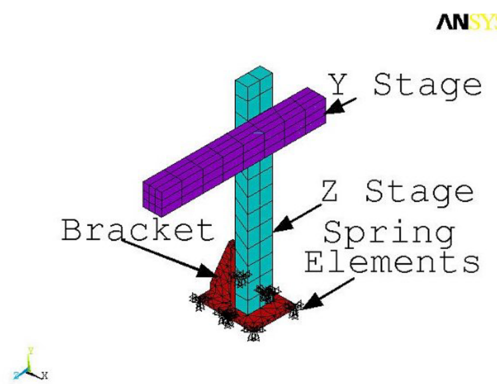

Fig. 4: Cartesian robot FE model.

Similar to the technique used for the goniometer, a lumped mass FE model of the Cartesian robot was constructed. Stiffness and damping data for the individual stages were used to accurately model the interconnections between stages. Two FE models were built, with the difference being additional stiffness and damping elements for the dual stage model. The resulting model is shown in Fig. 4. The X stage was not included in the model, as it was considered rigid and constrained by the ground.

2.3 *Solution Implementation*

The viscoelastic damper for the goniometer needed to have a minimum stiffness of 1E6 N/m and a damping ratio of at least 20% to achieve the desired vibratory response, which would achieve the desired reduction in transmissibility from 9 to 6. The viscoelastic material was selected to be WM 468MP, because this material has been used for vibration suppression for APS storage ring girders. It has been subjected to tests to verify the material behavior in an x-ray environment. The final design was a sandwich of stainless-steel plates with a layer of viscoelastic material between each plate (Fig. 5). The damping device will be connected to the goniometer across the horizontal stages, adding the required stiffness and damping to reduce the goniometer vibration amplitude.

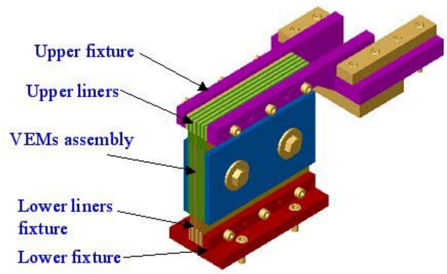

Fig. 5: Viscoelastic damper device.

2.4 Experimental Verification

After the viscoelastic damper was installed to the goniometer, transmissibility and autopower spectrum measurements were performed once again. The autopower spectrum measurements displayed an appreciable reduction in amplitude, although the environment was extremely quiet on the day of the test. Transmissibility measurements reveal a more important fact; the amplification ratio was reduced close to the target of 6. The last measurements were made with the coarse-positioning stages replaced with solid supports, this solution provides the largest improvement but no goniometer adjustment.

Unfortunately, a physical model of the dual carriage XYZ robot did not exist at the time of experimental testing. Comparison could only be made between the single carriage design and experiment and between the two FE models. The FE model indicated changing the X stage should increase the system stiffness; actual performance will need to be assessed when the design change is implemented.

3. Results

3.1 Goniometer

3.1.1 Before Damper

Goniometer baseline performance, at spot 1 (see Fig. 1), is illustrated by Fig. 6 and Fig. 7.

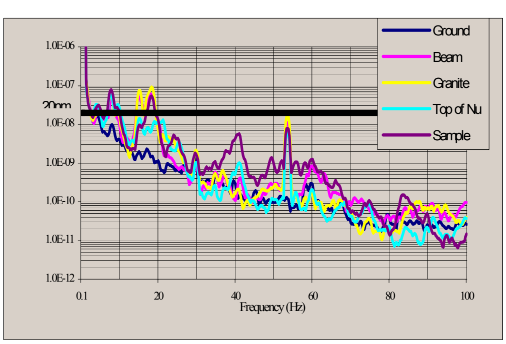

Fig. 6: Horizontal spectrum.

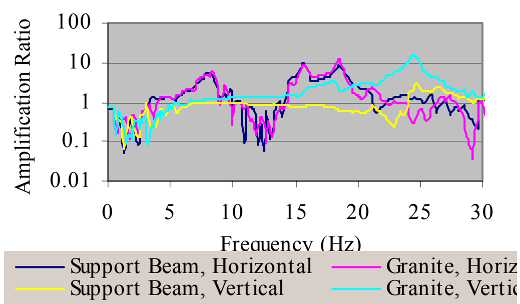

Fig. 7: Horizontal transmisibility.

3.1.2 After Damper

Goniometer performance, at spot 1 (see Fig. 1), after installation of the viscoelastic damper is illustrated by Fig. 8 and Fig. 9.

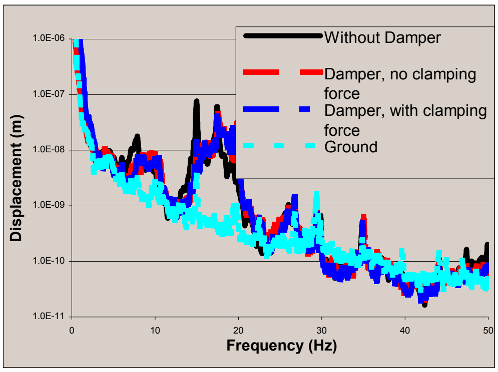

Fig. 8: Horizontal spectrum.

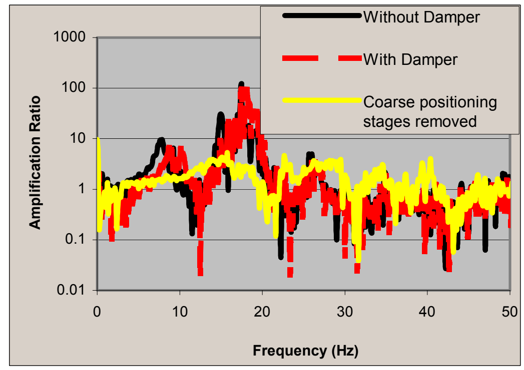

Fig. 9: Horizontal transmissibility.

3.2 Cartesian Robot

3.2.1 Single-Carriage Configuration

The FE model agrees with the experimental modal analysis as shown by the first mode in the FE analysis, Fig. 10 and experiment, Fig. 11.

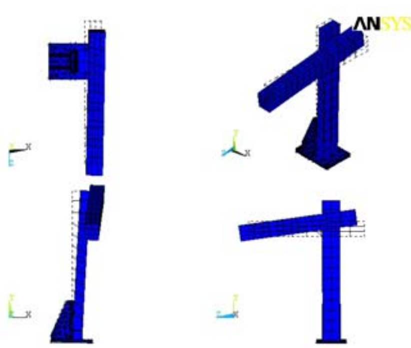

Fig. 10: FE mode 1, 25.3 Hz.

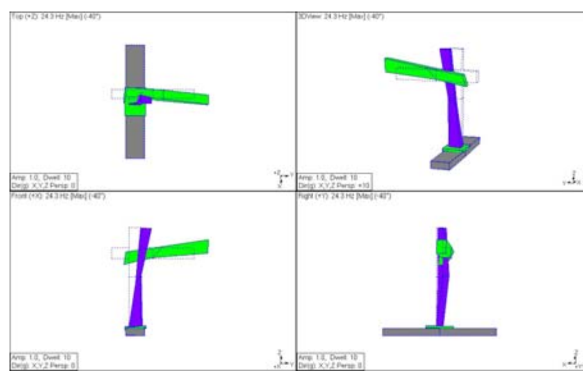

Fig. 11: Experiment, mode 1, 24.3 Hz.

3.2.2 Dual-Carriage Configuration

The FE model indicates an increase in stiffness for the dual carriage model with respect to the single-carriage configuration. Figure 12 illustrates the shift in frequency due to the added stiffness of the two-carriage stage. Figure 13 illustrates the second mode of the dual-carriage configuration.

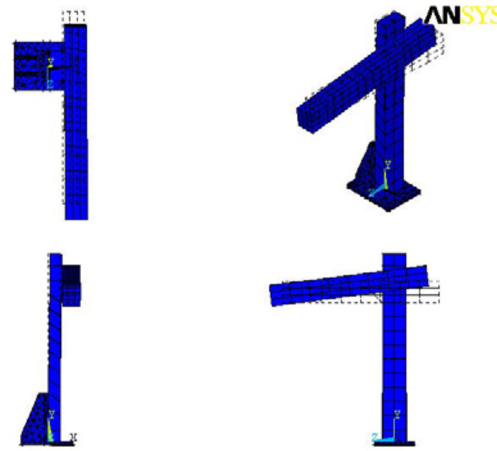

Fig. 12: FE mode 1, 26.9 Hz.

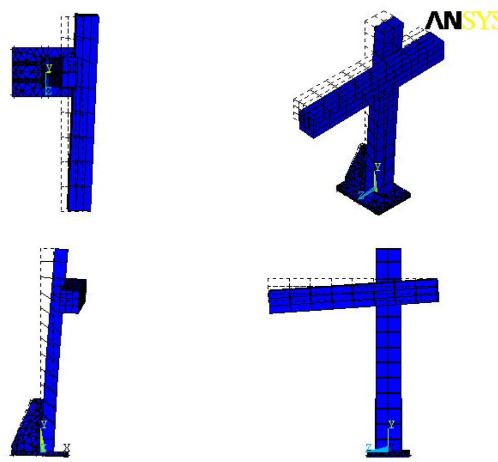

Fig. 13: FE mode 2, 57.6 Hz.

4. Conclusions

In the diagnosis of the goniometer and the design of the viscoelastic damping device experimental vibration measurements were valuable in three areas: initial investigation, computer model construction, and solution verification. Accurately evaluating system parameters would not have been possible if only a computer modeling approach had been taken. In addition, a properly devised viscoelastic damper can be effectively used to increase low-frequency vibration performance of large beamline positioning components. Modal analysis is a successful method for determining important system parameters, which can then be inserted into a computer simulation.

5. Acknowledgments

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6. References

[1] J. A. Jendrzejczyk, M. W. Wambsganss, R. K. Smith, "Surveillance of 18 Hz components of ground motion at the APS site," Argonne National Laboratory (1990) unpublished.

[2] J. A. Jendrzejczyk, M. W. Wambsganss, R. K. Smith, "Surface wave attenuation characteristics at the APS site," Argonne National Laboratory (1992) unpublished.

[3] T. J. Royston, "Vibration characteristics of an APS lab facility in building 401," ANL Report APS/LS Note 263, 1996.

[4] D. Mangra, S. Sharma, and J. Jendrzejczyk, "Passive vibration damping of the APS machine components," ANL Report ANL/ASD/CP_88981, 1995.

[5] C. Ader, J Jendrzejczyk, and D. Mangra, "The application of viscoelastic damping materials to control the vibration of magnets in a synchrotron radiation facility," ANL Report ANL/ET/CP-85273, 1995.

[6] D. Mangra, S. Sharma, and C Doose, " Performance of the vibration damping pads in the APS storage ring," 1st International Workshop on Mechanical Engineering Design of Synchrotron Radiation Equipment and Instruments, Paul Scherrer Institute, Wurenlingen/Villigen, Switzerland, July 13/14, 2000.

[7] J. A. Jendrzejczyk, M. W. Wambsganss, R. K. Smith, "Vibration study of the APS storage ring 0.8 meter quadrupole maget/magnet support assembly," Argonne National Laboratory (1992) unpublished.

[8] S. S. Chen, S. Zhu, M. W. Wambsganss, J. A. Jendrzejczyk, "Identification of root cause of vibration of a liquid gallium-cooled silicon monochromator and recommendations for abatement," Argonne National Laboratory (1994) unpublished.

[9] I. Basdogan, "A theoretical and experimental study of the dynamic response of high precision optical positioning systems," Ph.D. diss., Department of Mechanical Engineering, University of Illinois at Chicago (1997).

[10] I. Basdogan, T. J. Royston, A. A. Shabana, D. Shu, T. Kuzay, "Vibratory response of a mirror support/positioning system for the Advanced Photon Source Project at Argonne National Laboratory," Proc. SPIE 2865, 2-13 (1996).

[11] J. Barraza, D. Shu, T. Kuzay, T. J. Royston, "Vibratory response modeling and verification of a high-precision optical positioning system," Proc. SPIE 3786, 86-97 (1999).

[12] J. Barraza, J., D. Shu, T. Kuzay, T. J. Royston, "Vibratory response of a precision double-multilayer-monochromator positioning system using a generic modeling program with experimental verification," Proc. SPIE 3429, 265-270 (1998).

[13] S. H. Lee, C. Preissner, B. Lai, Z. Cai, D. Shu, "Enhancement of the vibration stability of a microdiffraction goniometer," Proc. SPIE 4771, 100-110 (2002).

[14] C. Preissner, S. H. Lee,T. J. Royston, D. Shu, "Optimal dynamic performance for high-precision actuators/stages," Proc. SPIE 4771, 91-99 (2002).