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LE GUIDE FOR SUPPORT: A COOKBOOK FOR MODELING OF ACCELERATOR STRUCTURES*

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Abstract

The Advanced Photon Source-Upgrade (APS-U) project has stringent specifications and a 12 month installation schedule. Some form of these constraints appear to be common at all multi-bend achromat upgrade projects. At the APS-U, no full tests will be made of the final accelerator support design. The evaluation of the final design against the specifications will be based primarily on computer simulations using virtual inputs.

Ensuring that the final designs meet specifications solely based on simulations is much like cooking a complex, multicourse meal without a trial run. Producing a successful meal on the first try requires a prior understanding of the ingredients, techniques, and interactions between the constituents. A good cookbook can be essential in providing this understanding. Likewise, producing an accelerator support final design that meets the requirements requires a prior understanding of the materials, components, techniques, and interactions between them. This poster describes a cookbook-style approach that any design team can use to confidently predict important characteristics such as natural frequency and ambient vibration response with an error of around 10%.

MOTIVATION

Many third-generation synchrotrons are on a path to a high-brightness, multi-bend achromat (MBA) upgrade [1-3]. In addition to the orders-of-magnitude increase in brightness, these MBA upgrades are unusual in that they involve shutting down very productive machines for a period, removing and replacing the storage ring components, commissioning, and making them again available to the users in the shortest time possible. There is precedent for this in the United States with the SPEAR 3 upgrade project [4]. However, the aggressive project schedules of these MBA upgrades impose additional engineering constraints on the magnet support systems, beyond those driven by physics requirements or those found in greenfield synchrotron projects.

From the beginning of the design process the APS-U supports team assumed there would be a large reliance on computer simulation of magnet support mechanical behavior. Table 1 shows the stringent APS-U mechanical motion tolerances. In addition to these mechanical tolerances, there are space constraints dictated by installation, utilities, and front requirements. The short installation time requires magnets be grouped into larger structures and installed fully aligned and assembled, reducing in-tunnel assembly time. These constraints all have to be met within the project schedule.

Table 1: APS-U Vibration Tolerances

Specified over 1-100 Hz	X (rms)	Y (rms)
Girder Vibration	20 nm	20 nm
Quadrupole Vibration	10 nm	10 nm

The APS-U magnet support design has evolved in parallel with the magnet, vacuum system, and revised physics requirements, moving beyond that described previously [5, 6]. The only full-scale magnet support prototype has a structure quite different that the final design.

Evaluating the final APS-U magnet support design against Table 1, from ref. [1], using only a numerical model, is challenging. Finite element (FE) model and vibration modelling is convenient due to the ease of incorporating CAD data. However, it is common for synchrotron engineers to expect FE results to be unrealistic [7], to tune a FE model *a posteriori* [8], or to use the FE model to confirm the behaviour of one particular design [9]. The APS-U supports team wanted a model and a process by which the final design could be checked against requirements, the effect of design trade-offs could be evaluated, and that could be used to accurately predict beam motions [10].

RECIPE

Ingredients

The APS-U magnet support systems consist of the following components, listed in order from the floor to the magnets: 1) a thin epoxy grout line, 2) a steel-reinforced concrete 'plinth', 3) a set of support and alignment mechanisms, 4) a cast iron 'girder', and 5) individual magnets. Measurements of the APS have shown the floor can be considered to be rigid. The most basic ingredients in any mechanical dynamic model are geometry, mass, stiffness, and damping. To obtain the final result, one more ingredient is necessary, the facility vibration. In this section we explain how these ingredients can be mixed to capture the behaviour of accelerator magnet support systems, using the APS-U magnet support system as an example.

The appropriate mixing of the ingredients for each component is key to producing an accurate FE model. For the epoxy grout, plinth, and girder, CAD geometry, combined with the material property data are sufficient to accurately capture the component behaviour. The epoxy manufacturer supplies modulus data. For the steel and concrete plinth, the steel reinforcing geometry is captured in the CAD model and can be used with common steel material properties, while the concrete vendor supplied modulus data on the proprietary concrete mix. Likewise for the girder, the

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CAD data accurately define the geometry and the foundry supplied the cast iron material properties. The APS-U magnets are fixed to the girder and can be well-represented with simplified CAD data that capture the mass and inertia properties.

In contrast, the support and alignment mechanisms behaviour *are not* accurately captured using CAD data and available material properties. The use of CAD geometry and material properties are the root source of the unreliability of the FE modal analyses such as those of references [7] and [8]. Regardless of the support and alignment mechanism type, threaded rods, wedge jacks, cam movers, etc., these components exhibit nonlinear stiffness and damping with respect to load. This is due to all of the interfaces within these components (threads, sliding surfaces, Hertzian contacts, spherical bearings, etc.). When relatively lightly loaded, these interfaces are fairly compliant. In addition, the slope of the stiffness vs. load curve can be steep at light loads. This means small changes in load can greatly influence behaviour. The use of CAD geometry vastly over predicts the component stiffness, and without *a posteriori* model tuning results in over predicting modal response and under predicting vibration response. Typically manufacturers do not or cannot supply the necessary component data.

The best way to capture the support and alignment component behaviour is to replace the geometric representation with a discrete stiffnesses in the FE model. The alignment component masses can be neglected if sufficiently small (m is generally <<<10% of total system mass), or easily represented with discrete masses. Simple vibration measurements [11] can then be used to determine the load-dependent, linearized, diagonal stiffness coefficient matrix for a support component (three translational and three torsional stiffnesses).

The final ingredient to predict the overall system vibration response is the facility vibration. This is one area where upgrading an existing facility has an advantage over a greenfield site. As an existing facility, the APS has been stable and the vibration characteristics can and have been well characterized. The storage ring floor motions have been measured at various locations in three directions.

Measures

Like baking French pastry, an accurate mechanical model is highly dependent both the form (CAD vs. discrete) and amount (geometry derived vs. measured stiffness and damping) of the ingredients. Experimental modal analysis (EMA) measurements were made of the FODO plinth and girder protoypes to confirm the assumption that geometry and material properties were sufficient to predict their behavior. Whereas measurements were made of the alignment components to generate the necessary linearized stiffness coefficients.

A full (measured FRFs, curve fitting (natural frequency and damping), modal visualization) free boundary condition (BC) EMA was conducted on the plinth and girder. Figure 1 shows the girder casting rigged for an EMA. Triaxial, roving accelerometers were used to measure the impact frequency response functions (FRFs) at 28 points, enough to properly resolve the first 6 modes. The slings provided a very soft support, approximating free BCs.

The match between the FE and EMA modal results are a metric of dynamic model performance. A FE modal model that matches the EMA well will yield accurate random vibration results when combined with measured damping values. The model will also provide accurate static and thermal deflection results. The first five modes of the plinth and six modes of the girder matched to an average of 1.98% and 5.40% respectively. Table 2 shows the first three modes for the girder.

Figure 1: The FODO prototype girder casting at the manufacturer, rigged for a free BC EMA.

Table 2: Girder Modal Results, Average Difference 1.99%

Mode	EMA (Hz)	FEA (Hz)	% Diff.
1	106	104	1.89
2	157	154	1.91
3	232	227	2.16

Linearized stiffness coefficients were determined for a variety of wedge jack adjusters, spherical bearings, and the metal-polymer bearings used in both the vertical and inplane adjustment stacks. Dynamic testing proved to be more reliable, flexible, and provided more information (diagonal stiffness matrix) that static stiffness measurements used in [9].

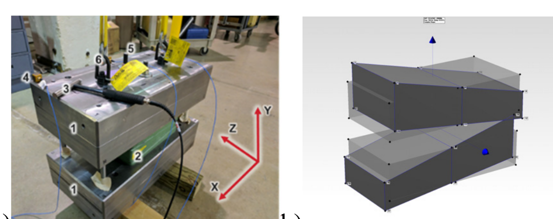
a)  b)

Figure 2: Stiffness test rig shown in a). The components are: 1) masses, 2) wedge jack, 3) impact hammer, 4) accelerometers, 5) threaded rods, and 6) slings. In b) The mode involving rotation about the Z axis is shown.

The component is clamped between the two rigid (in frequency range of interest) masses shown in Figure 2 and a free BC modal analysis is done at many load. A load cell measures the applied force. For each load, three translational and three rotational stiffness are determined. The assembly is symmetric and has six uncoupled modes, relating

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to each of the translational and torsional stiffnesses. Modal analysis software is used to curve fit the data for each load, determining natural frequencies and damping. Modes shapes are also visualized. Stiffnesses are initially estimated at the highest applied load using a single-degree of freedom model. The results are refined with a FE model of the test rig. The stiffness at the lower load levels can then be determined simply by scaling the high-load stiffnesses by the square of the ratio of natural frequencies. In this manner a table of stiffnesses is determined for each component. Figure 3 shows one set of stiffness vs. load curves.

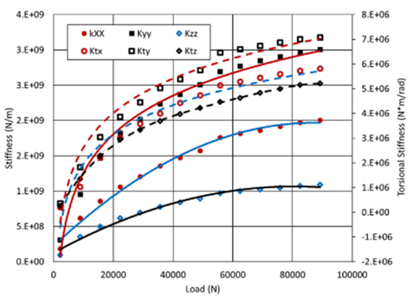
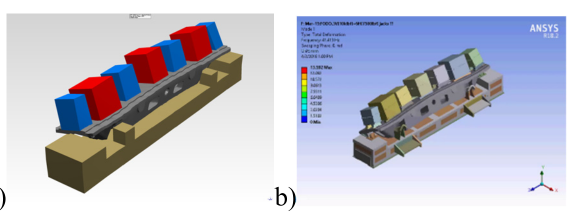


Figure 3: Airloc 2012-KSKCV translational (Kxx, Kyy, Kzz) and torsional (Ktx, Kty, Ktz) stiffnesses.

RESULTS

The first analysis of the complete model was a modal analysis both to verify the modelling process and to estimate the first natural frequency, and provide a set of mode shapes for random vibration analysis. The first three modes matched well, with an average error of 7.3%. This is with only *a priori* knowledge. Figure 4 shows the first mode comparison.



a) b)

Figure 4: Grouted FODO prototype, first mode shape comparison, a) is EMA at 41 Hz, while b) is FE modal analysis at 42 Hz (2.4% difference).

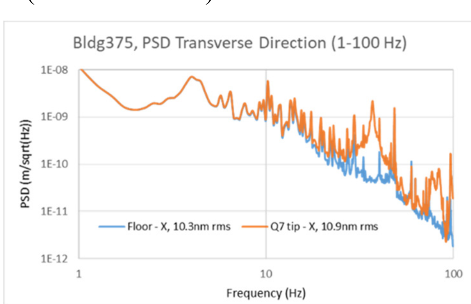


Figure 5: Predicted FODO magnet pole tip response using random vibration analysis. Blue is floor (input) while orange is magnet (response).

The modal model can then be used in a random vibration analysis to estimate magnet pole tip motion, as in Figure 5. Work is ongoing to measure and compare the experimental response. Work is also ongoing to perform a transient dynamic analysis, from which support transfer functions and magnet-to-magnet phase relations can be estimated.

CONCLUSIONS

This paper describes a process by which accelerator magnet support structures can be accurately modelled using only data from material properties, CAD geometry, and sub-component dynamic stiffness tests. The expected error is less than 10%. The accuracy of this method allows for using only FE model results to evaluate the APS-U supports final design against requirements both dynamically and statically. It also allows for a novel approach to estimating mechanical motion-related orbit distortions [10]. The approach is also readily applied to beamline instrumentation as well.

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