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HORIZONTAL-GAP VERTICALLY-POLARIZING UNDULATOR (HGVPU) DESIGN CHALLENGES AND RESOLUTIONS*

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Abstract

The Horizontal-Gap Vertically-Polarizing Undulator (HGVPU) is a compact, innovative, variable-gap insertion device developed by Argonne National Laboratory for the LCLS-II HXR beamline at SLAC. A full sized 3.4-meter-long prototype has been built and fully tested meeting all LCLS-II undulator specifications. An array of conical springs compensates the attractive magnetic forces of the undulator jaws. These springs are designed to exhibit non-linear spring characteristics that can be closely tuned to match the force curve exerted by the magnetic field, thereby minimizing the overall deflection of the strongbacks. The HGVPU also utilizes the existing LCLS-I support and motion system along with other existing equipment and infrastructure, thus lowering overall cost and installation downtime.

Figure 1: Rendering of final 3.4m HGVPU prototype.

INTRODUCTION

The 3.4-meter-long prototype HGVPU, shown in Figure 1, was successfully developed and built but not without its own unique challenges [1,2]. Very stringent straightness requirements of the strongbacks, temperamental properties of the spring compensation mechanism, mechanical stability over a large temperature range, installation and handling logistics, and unique control system demands all contributed to the development process of this device. Some of the more significant of these challenges will be examined as well as their final resolutions.

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DESIGN OVERVIEW

The HGVPU is a variable gap undulator that uses an array of springs to compensate for the large attractive magnetic forces between the horizontally opposed undulator jaws. In order to minimize deflection of the strongbacks and to reduce the forces on the actuators, the conical springs are designed with non-linear spring characteristics that approximate the exponential-like force curve induced by the magnetic attractive forces over the variable gap range. Figure 2 shows the cross-sectional view of a single undulator jaw. The magnets and poles are mounted to the magnet/pole mounts which are bolted to the undulator strongback from the back side. Each jaw rests on and is guided by two linear slides that are bolted to the girder surface.

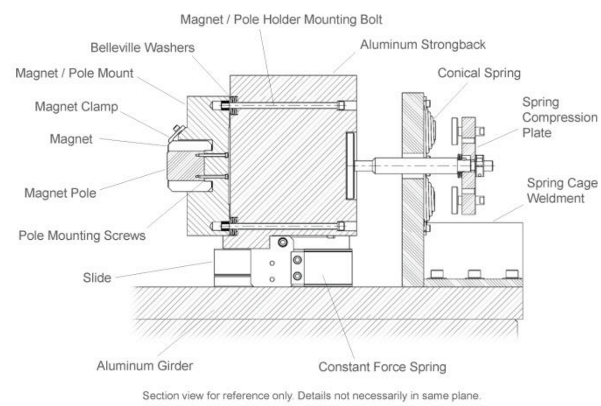

Figure 2: Cross-Section View of Undulator Jaw.

Two actuators, which are not shown in this view, position each undulator jaw at fixed engagement points along the length of the strongback to provide accurate positioning and allow for tapering capability. The springs are affixed to stationary spring cage brackets where they are engaged by the spring compression plates which are directly coupled to the strongbacks. For gaps >21mm, the attractive magnetic forces are negligible and spring compensation is unnecessary. At 21mm, the springs are engaged and increasingly compressed as the gap is decreased to a minimum gap of 7mm.

STRONGBACK STRAIGHTNESS AND ACTUATOR POSITION

Minimizing deflection of the strongbacks was the single greatest design challenge to overcome. The physics requirement for this particular undulator is that the strongbacks deflect less than 19 microns over the full gap range of 200mm to 7mm [3]. The attractive magnetic

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force of the two opposing undulator jaws exceeds 6000lbs at minimum gap. Without an extremely rigid strongback the magnetic arrays would deflect under these forces rendering the undulator ineffective. Typically, this would be resolved by utilizing a very wide beam with a high moment of inertia.

As this is a horizontal gap device, the overall width was limited to one meter in order to fit within the undulator hall while still allowing the proper egress. A very wide cross-sectioned beam, was not a viable option, therefore the idea of using springs to compensate for the magnetic forces was conceived.

In a perfectly balanced system, the spring forces would have the same force curve as the magnetic forces and would be equally spread across the length of the undulator. The beam would have zero deflection, and the actuators could, in theory, be mounted anywhere along the length of the strongback.

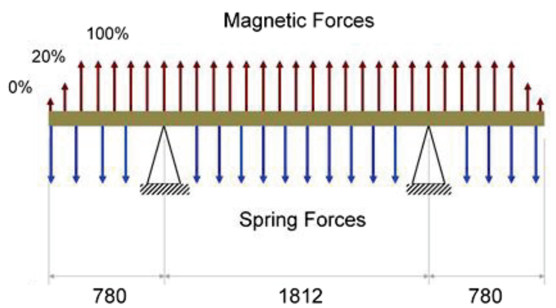

Figure 3: Free-Body Diagram of Strongback.

In reality, the magnetic force is evenly distributed across 260 poles with slightly lower field strength at the end poles. The corresponding spring cage units themselves take up a finite amount of space limiting the number of force compensation points to 18. Position and space to accommodate the actuators had to be considered also.

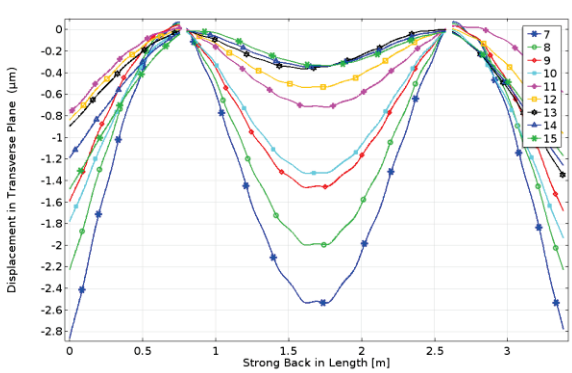

Figure 4: Simulated strongback deflections at gaps of 7mm to 15mm.

For a perfectly evenly distributed force, the ideal position for the actuator engagement points, to minimize deflection of a beam, would be close to the Airy points ($L/\sqrt{3}$). Due to the non-ideal factors of the system, and after some iteration and compromises where made, the final arrangement, shown in Figure 3, was chosen. A slightly tighter spacing of the spring cages between the actuators as opposed to those at the ends was necessary due to the space constraints of the components.

Figure 4 shows simulated deflections of the strongback over its length for gaps of 7mm to 15mm. Finite element analysis of the system showed the maximum deflection of the strongback would be less than 6μm per side in its worst case scenario with a gap of 7mm, which is well within the straightness requirements of the system. The simulation results were later confirmed by magnetic measurement.

SPRING CAGE CALIBRATION AND INSTALLATION

The attractive magnetic field strength drops off exponentially over distance. Unlike standard springs which exhibit a linear force change with displacement, conical springs can be designed to mimic the exponential force curve of a magnetic field. Working with a spring manufacturer, two separate types of conical springs were designed. One set of springs with slightly higher forces and the other with slightly lower forces than the ideal calculated magnetic force curve.

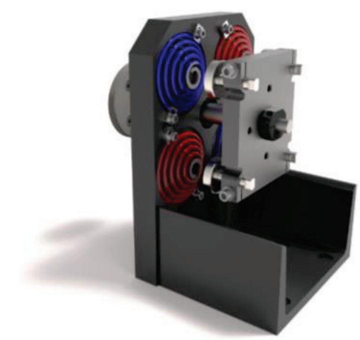

Figure 5: Individual Spring Cage Assembly.

The springs were initially numbered, measured, and sorted in pairs for both the strong and weak type. A special calibration fixture was designed to measure both the spring curves of the individual springs and also the full spring cage assemblies. Each spring cage is comprised of opposing strong and weak pairs, identified as red and blue in figure 5, that were previously sorted in order to achieve an ideal spring rate. Each spring cage is then individually calibrated over the full gap range on the calibration fixture, then moved to a specific gap of 30mm where a transfer clamp is installed to lock the position of the spring compression plate for final transfer to the undulator.

Figure 6 shows the force vs. compression plot of the four springs in a typical spring cage assembly. It is evident that any slight misalignment of the spring cages at the smallest gaps can significantly affect the forces at that gap. Displacements of 10μm, corresponding to forces of 2lbs or more were deemed enough to induce unacceptable

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deformations in the strongback. Transferring the calibrated spring cage to the undulator at final assembly therefore requires that the position of the spring compression plate be in the exact same position relative to the mounting surface of the strongback as its corresponding position in the calibration fixture. Installation of each spring cage is done individually while the encoder readouts are monitored to ensure that the gap of the undulator is the same as that of the spring cage calibration and that there are no binding forces. Even tightening the bolts that fasten the spring cage to the girder surface can create positional errors so this step is critical to the performance of the HGVPU.

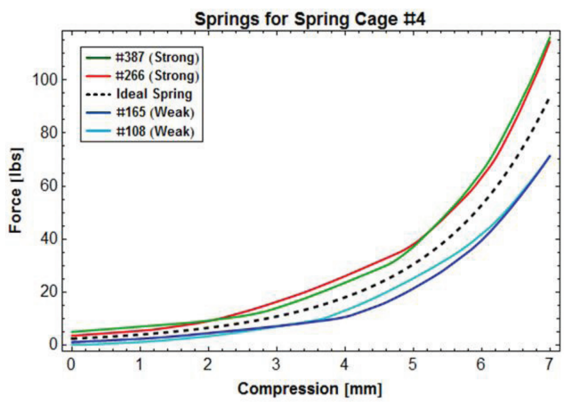

Figure 6: Spring Cage Load Curves.

GIRDER OPTIMIZATION & DESIGN

Initially, the prototype HGVPU utilized the original LCLS steel girder, however the steel girder was less than ideal for a number of reasons.

- The elevation of the girder top surface was around 3.5 inches too short and therefore required an array of spacers to make up for the difference (Figure 7).

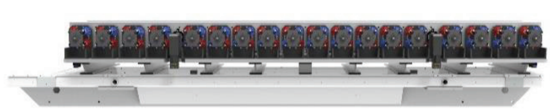

Figure 7: HGVPU Prototype on Steel Girder.

- Because of the offset loading condition of the girder, the spacers had to cantilever off of the girder on one side yet provide enough rigidity to resist the magnetic loads. This required the spacers to be very thick and heavy.
- The combined weight of the new gap-separation mechanism, base plate, and spacers increased the overall weight of the undulator by roughly 1500lbs. to a total of 6500lbs which is more than the capacity of the crane in the LCLS magnetic measurement facility.
- The dissimilar thermal expansion coefficients of the steel girder and aluminium strongbacks and magnet structures could potentially cause issues when undergoing temperature variations.

To meet the vigorous installation schedule of LCLS-II, it was decided early on that rather than reuse the old girders, new girders would be fabricated so that the undulator could be preassembled and ready for installation. This provided the opportunity to design new girders that would be optimized for the unique requirements of the HGVPU. Four initial concepts were presented;

- Option 0; Keep the existing steel girder design using the spacers.
- Option 1; A one-piece integrated undulator/girder assembly requiring the girder, undulator, and vacuum chamber to be moved as a single unit.
- Option 2; A girder with a rigid undulator support plate that can be separated from the girder, allowing the girder to remain in place, but still required that the vacuum chamber would have to be removed and re-aligned.
- Option 3; A two-piece gap separation mechanism with separate right and left side undulator jaws/spring compensation units, that could be removed leaving the vacuum chamber installed so that chamber realignment and breaking vacuum would not be necessary. The jaws would be tuned as a matched pair on a separate fixture, and then reassembled onto the girder in the undulator hall. This third option, although unique from a maintenance perspective, would have been much too difficult to align with the precision necessary to meet the magnetic requirements.

Ultimately, Option 1 was chosen as it was the most robust and stable design, the most cost effective, and any minor shortcomings in maintenance were deemed negligible.

Figure 8: Optimized Aluminum Girder Design.

The new girder is a welded aluminum structure that is much lighter than the steel girder and has the same thermal expansion characteristics as the undulator jaws. It is taller to make up for the height of the spacers, which also adds to its stiffness, and has identical interfaces for the LCLS pedestal supports. As shown in figure 8, the girder structure underneath the top plate is designed so that a rib is centered on each spring cage in order to stiffen the structure directly where the forces are concentrated.

Figure 9 shows the simulated surface profile plot of deflection for the force-bearing components of the undulator. The maximum deflection of the girder baseplate is around 12µm, which occurs a little bit off-center in the central region towards the aisle side. A deflection of around 80µm is seen at the highest point of the vertical

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plate of the spring cages, and although the spring cage deflection is much greater than the maximum allowable deflection of the girder, this deflection is already compensated for in the individual spring cage calibration. The linear bending of the cage itself could also conceivably help to smooth out the approximated exponential force curve of the springs.

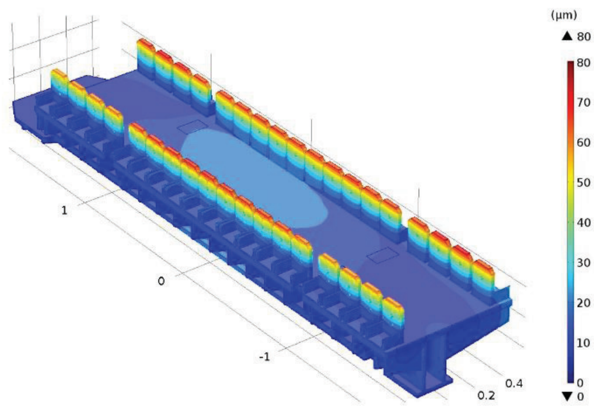

Figure 9: Simulation results of deformation.

MECHANICAL TUNING

Mechanical tuning is essentially levelling of the pole tips as measured with a very precise gap measurement probe. Typically, at the APS, undulator magnetic arrays are mechanically tuned by shimming in between the magnet/pole holders and the strongback. For the very stringent requirements of this undulator, this process became more difficult as minor adjustments lead to adjacent shims losing contact or falling out entirely. The idea of using Belleville washers in place of shims was born. Inserted at each of the hold down bolts, the washers allow continuous alignment of the strongbacks as they are designed hold position rather than having to loosen, shim, and re-tighten the bolts. The cross-section view of Figure 2 shows the position of the three spring washers. Stacked in series, they provided roughly 950lbs of force at each bolt. As these springs act in the same direction as the magnetic force, their deflection does not vary due to a change in gap. Initially, the assembly started with a 100µm gap so that tuning could be done in both directions. Later it became easier to measure and identify the highest pole tip as the nominal position and correspondingly adjust all of the others out to this point. This seemed like a sound idea however we later discovered that with changing gap, the points where the magnet base and strongbacks were in contact behaved differently due to the tensile forces within the bolts. The original plan with the 100µm gap would likely eliminate this. Since the mechanical specifications had been met and due to the tight delivery schedule, we did not have time to investigate the issue as we would have had to recalibrate all of the spring cages to compensate for the 100µm shift.

SUPPORT COMPATIBILITY

To ensure compatibility with the existing LCLS pedestal supports and cam movers, the HGVPU was fully evaluated for both overall gross weight and seismic performance. The original fixed-gap LCLS undulator girder was centered on the undulator strongbacks rather than the beam centerline. The HGVPU gap-separation mechanism is however centered around the electron beam which requires a non-symmetric girder. The unique off-center loading condition of the undulator was investigated and since the gap changes symmetrically, and the center of gravity stays within the footprint of the cam movers, it was determined to be a non-issue. The new girder design also gave us the opportunity to optimize for this condition as shown in figure 10.

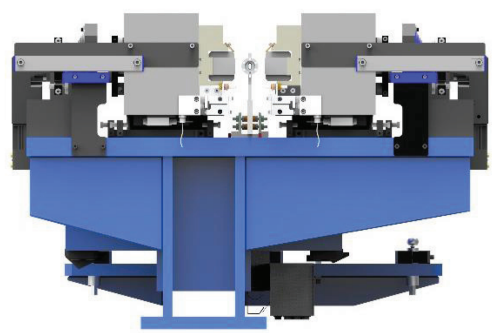

Figure 10: Upstream End View of HGVPU.

ENVIRONMENTAL TESTING

The goal for mechanical stability of the HGVPU was that it be able to remain tuned after undergoing a temperature variation of 20°C +/- 15°C (5°C-35°C). The concern was that the dissimilar thermal expansion coefficients of the steel girder relative to the aluminum strongbacks and magnet arrays would cause stresses in the system as it underwent temperature variations during transport and storage.

For temperature cycling of the undulator, a temperature controlled heating enclosure was purchased from Hemi Heating in Sweden. The heating enclosure has an operating range of up to 240°C. Six heating units, controlled by 12 thermocouples, regulate an even temperature distribution inside the tent. The temperature profile can also be programmed for ramp-up and cool-down times. The direct temperature of the HGVPU was measured separately via six thermocouples attached directly to the device itself. Readings from these sensors were recorded in LabView over the duration of the test. To cool the undulator below ambient temperature, the enclosure was specified with two slots to install standard residential A/C window units. Two 15,000 Btu, 115V units were purchased. The units have an operating range down to 16°C. In order to reach temperatures below 16°C, two line-voltage thermostats were purchased and wired in-line with the 120V power input of the units. We tested that the A/C unit came back on with its previous settings when we interrupted the

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power by unplugging the device from the wall. The thermocouple internal to the A/C unit was then re-routed to remain outside of the enclosure, therefore making the unit think that the temperature was higher, and in-turn tricking the unit into staying on indefinitely. The inline thermostats then regulated the temperature inside the tent by cycling the power to the unit. This worked very well.

At the time of testing, the HGVPU was still assembled on the original LCLS steel girder. Although it was assumed that the dissimilar thermal expansion coefficient of the steel girder relative to the aluminum magnet structures could cause stresses and mechanical tuning issues, this appears not to be the case. Furthermore, the final design of the optimized girder is aluminum so this is no longer an issue. Ultimately, the HGVPU was shipped to SLAC and measurements showed that it remained tuned.

GAP SYMMETRY

As much as possible of the existing LCLS control system was reused and/or modified. The cam mover controls remained the same, and a few motors, limit switches and their respective control chassis were added for the new variable gap design. The gap symmetry of the HGVPU is critical. If left uncompensated, the extremely high internal forces could severely damage the device. In one possible scenario where the undulator is at minimum gap and one of the jaws were allowed to open, the remaining jaw would be subjected to the full spring compensation force without the balancing magnetic force. In this scenario, the strongback would surely deform and the motors and drive components would likely undergo significant damage. Symmetry monitoring is therefore incorporated into the control logic.

CONCLUSION

As with any project, typical time constraints, cost constraints, and space constrains all had to be balanced and compromises were made. The end result however produced a very robust device that lead to the HGVPU being chosen as the baseline for the LCLS-II project hard x-ray undulator line shown in Figure 11.

ACKNOWLEDGMENT

The success of this project was the culmination of work of many talented individuals that I had the great pleasure of working with. As the project manager, and integration engineer for the 3.4-meter prototype, I became involved with the project after much of the initial concepts had been ironed out and two smaller scale prototypes had previously been built and tested. In addition to this paper's co-authors, Rich Voogd and Marty Smith did exceptional work developing the control system for the device, Jie Liu and Vic Guarino provided additional finite element analysis, Greg Wiemerslage and Jason Lerch developed the integrated vacuum chamber, Bill Jansma, Kristine Mietsner, Keith Knight and Rolando Gwekoh, provided survey and alignment support, John Terrhar, Joe Gagliano, Eric McCarthy and Mike Merritt provided technician support, and Pat Den Hartog, Marian White, and Patricia Mast were instrumental in costing and scheduling. We also appreciate all of the correspondence with our peers at SLAC National Accelerator Laboratory and Lawrence Berkeley National Laboratory.

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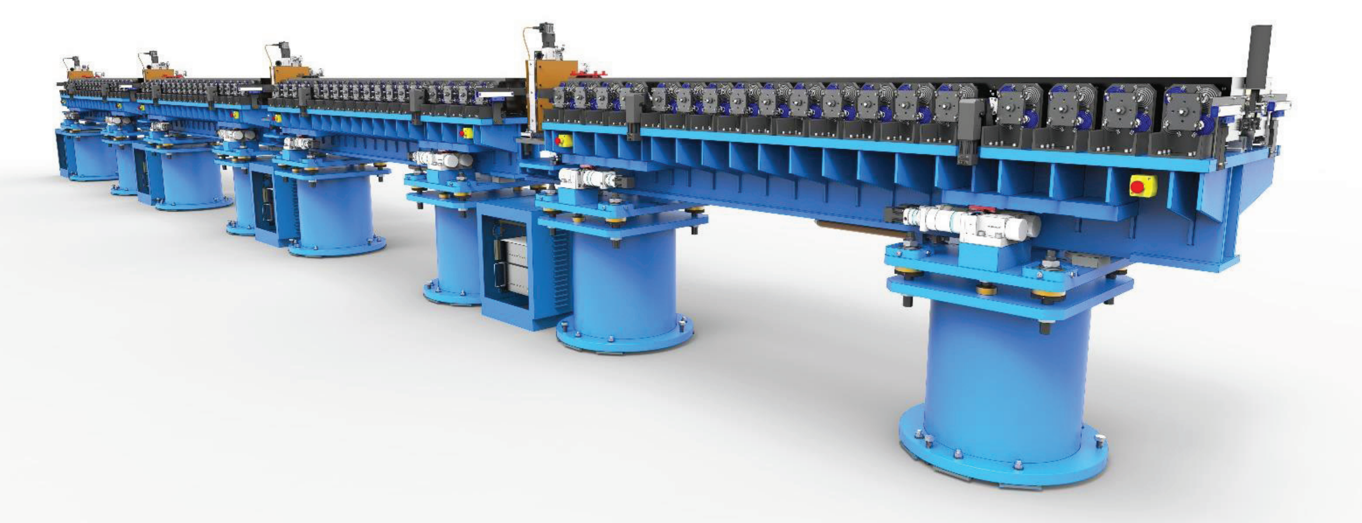

Figure 11: Rendered Image of Hard X-ray Undulator Beamline.