# Advanced High-Frequency 6-DOF Vibration Testing Using the Tensor

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**Commercially available vibration test systems able to reproduce and accurately control multiple-input, multiple-output vibration tests are often constrained by a limited-frequency band due to excessive mass and low natural frequencies of the fixtures. Consequently, their use in Department of Defense test facilities is limited to a select number of test profiles found in MIL-STD documents and/or platform-specific tests where the frequency band of interest is below 500 Hz. This frequency limitation has now been addressed with the introduction of a new vibration test system based on redundant electro-dynamic shakers hydrostatically coupled to the specimen mounting table. This system, called the Tensor™ 18kN, designed, built, and tested by Team Corporation, has been delivered to two sites in the U.S. This article discusses the design, with an emphasis on mechanical solutions that increase the frequency bandwidth and provide an over-determined number of control points for performance to 2,000 Hz. Additionally, the system response for two specific vibration tests is presented in detail, with a discussion of the system control and its ability to minimize the response of particular vibration modes.**

What is a tensor? Wikipedia defines a tensor as "a geometric object that describes the linear relations between vectors, scalars, and other tensors. Elementary examples of such relations include the dot product, the cross product, and linear map. A tensor can be represented as a multi-dimensional array of numerical values."[1] In basic engineering you may think of a tensor in terms of the three-dimensional stress state of a solid object. Remember the cube element showing the normal and shear stresses on each face? Now, in vibration testing you can think of a tensor as the solution to reproducing high-frequency, multi-axis vibration "stress states" in the lab. Team Corporation's new Tensor™ 18kN is the most advanced commercially available vibration test system capable of replicating field vibration environments out to 2,000 Hz. Twelve independent excitation inputs are linearly mapped into six controlled degrees of freedom (DOF).

The basic concept of the Tensor 18kN originated in a much smaller system developed in 2007, namely the Tensor™ 900. This system was novel in that 12 electro-dynamic (ED) shakers were configured in such a way, with the proper bearing arrangement, to provide 6-DOF control out to 2,000 Hz. In fact, the users of this system have successfully operated it past 3,000 Hz. This is a small system and was used to develop the basic premise of the high-frequency, over-actuated, 6-DOF vibration test system. It has 200 lbf RMS per axis and an 8 × 8-inch table. The first customers were Sandia National Laboratories and the University of Maryland's Center for Advanced Lifecycle Engineering (CALCE). CALCE uses the Tensor 900 for accelerated stress testing and researching the reliability of printed circuit boards subjected to various vibration environments. Both of these sites have published papers over the last several years discussing how this system has improved their testing capability.[2-5] Figure 1 shows the Tensor 900.

The design of the Tensor™ 18kN vibration test system is covered under the following patents:
- U.S. Patent: 6 860 152
- China Patent: ZL 03 809 374.X
- Japan Patent: 4 217 210

## Mechanical System

The Tensor 18kN expands the design of the Tensor 900 to a size more practical for the typical user. The system is still designed to be over-determined and uses 12 custom-made ED shakers for excitation out to 2,000 Hz. However, now 900 lbf RMS shakers are used to drive a 30-inch-square table. With four shakers in each axis, this system produces 3,600 lbf RMS per axis and has a bare table moving mass of nominally 430 lbm. Figure 2 is a photo of the Tensor 18kN system and Table 1 lists the system's performance specifications.


*Figure 1. Tensor™ 900 vibration test system.*

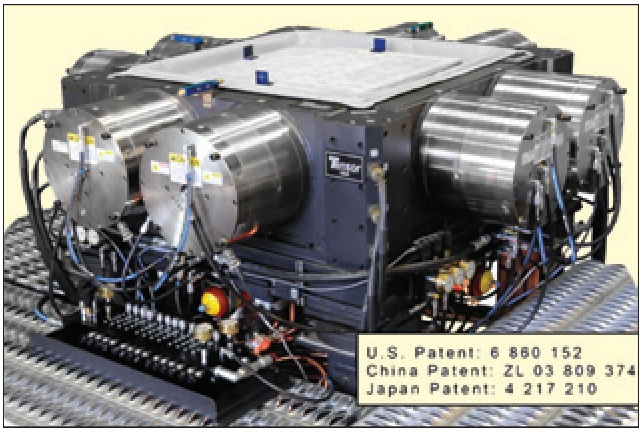


*Figure 2. Tensor™ 18kN vibration test system.*

The main components of the Tensor 18kN system are:
- Twelve custom ED shakers
- 30 × 30-inch vibration table
- Highly damped reaction mass
- Vertical preload actuator
- Hydraulic power supply
- Power amplifier set

*Table 1. Tensor 18kN performance specifications.*

| Specification (per Axis) | English Units | SI Units |
|---|---|---|
| Peak Sine Force | 4,800 lbf | 21.4 kN |
| RMS Random Force | 3,600 lbf | 16,0 kN |
| Moving Mass | 430 lbm | 195 kg |
| Peak Velocity | 50 in/sec | 1.3 m/sec |
| Dynamic Stroke | 1.00 in. p-p | 25 mm p-p |
| Static Stroke | 1.50 in. p-p | 38 mm p-p |
| Maximum Rotation | ±4.0 deg | ±4.0 deg |

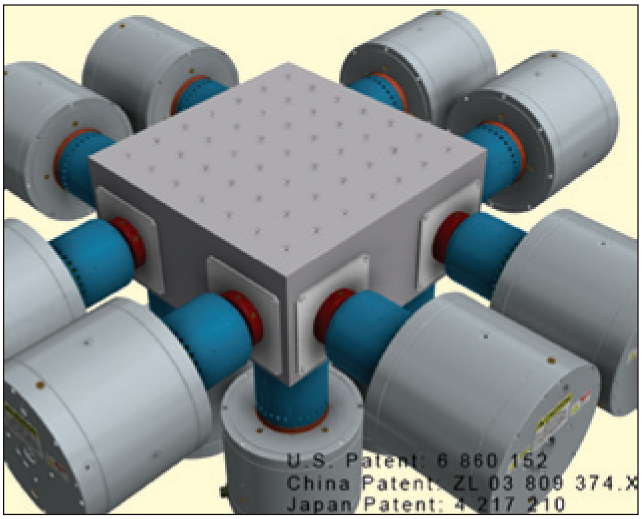


*Figure 3. Tensor™ electro-dynamic shaker configuration.*

The design of the shakers is based on a standard field coil and voice coil driver set. But beyond this, the similarities with standard ED shakers end. The armature flexures have been replaced with hydrostatic journal bearings. There is no specimen mounting surface, but rather, the end of the armature incorporates a version of Team Corporation's signature hydrostatic pad bearings. These bearings only transmit the armature force through the pad's axis and allow the table to move unrestrained in the other five DOFs about each armature. Incorporating hydrostatic pad bearings directly into the armature frees up the necessary DOFs to the system so that 12 shakers can be used to drive all six DOFs without mechanically locking up. The hydrostatic connection to the table is an extremely stiff and friction-free interface, providing for high-frequency response and no mechanical wear.

The mechanics of the Tensor 18kN require the shaker armatures to be preloaded against the table for proper operation. To accomplish this, each shaker has a preload mechanism integrated into its armature. In the horizontal axes, opposing shakers react each other's preload to remain in static equilibrium. In the vertical axis, a preload actuator is required to hold the table down against the armature preload. This actuator is hydraulically controlled and includes a variety of hydrostatic bearings to keep the system kinematically sound. The vertical preload actuator is essentially a soft spring with a large static load capacity and has minimal effect on the control of the system. Figure 3 shows the layout of the ED shakers relative to the vibration table.

In addition to pressurizing the hydrostatic bearings, hydraulic oil is also used to remove the heat generated from both the field and voice coils of the shakers. Hydraulic oil provides more effective heat transfer than forced-air convection used in most shakers, and the Tensor 18kN is configured to handle the flow of oil and keep it properly contained. A further benefit of oil-cooled coils is that the shakers are much quieter than conventional shakers because an external blower is not required.

The ED shakers, vibration table and vertical preload actuator are all integrated into a highly damped reaction mass. This reaction mass rests on air isolators and creates a system that is self-contained and isolated from the existing facility. No additional reaction mass is required for operation, only a floor capable of supporting the system weight of nominally 17,000 lbm [7,700 kg]. This allows for a system that is easily integrated into an existing laboratory facility.

A hydraulic power supply provides the required hydraulic pressure and flow, while a bank of 12 power amplifiers provides the electric voltage and current to the shakers. Both of these subsystems can be located remotely from the Tensor 18kN to minimize the noise levels present in the lab. This, along with the oil-cooled coils, provides for low ambient noise levels inside the laboratory.

## Control Scheme

For proper operation, the Tensor 18kN requires an advanced multiple-input, multiple-output (MIMO) vibration test controller, capable of driving 12 shakers using, at a minimum, 12 response accelerometers. The data presented here were collected using the Data Physics SignalStar Matrix controller.

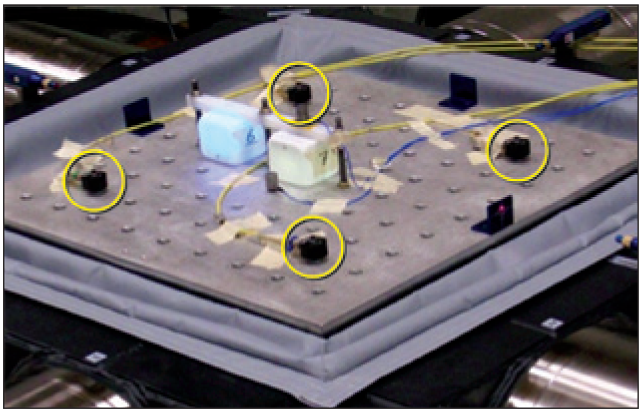

*Figure 4. Tri-axial control accelerometer placement.*

There are three common MIMO control schemes that can be used to control the Tensor 18kN. These are commonly referred to as:

- Square control – equal number of drive and control points.
- Over-determined control – unequal number of drives and control points, with more control than drive points.
- Signal transformation – linear mapping of the drive and/or control points to some pre-determined DOFs used for control.

The results presented here were produced using the signal transformation control scheme. The details of this transformation method are outlined in MIL-STD 810G, Method 527, Annex B.[6] Signal transformation is typically used to map the linear acceleration input measurements to the unconstrained rigid body DOFs of the table.

The number of DOFs that this algorithm can control is limited in theory by the number of independent drive points, but a test can be configured to control fewer. In the case of the Tensor this limit is 12 DOFs. Typically, the mapped DOFs of a "virtual point" are chosen to be the rigid body DOF of interest. The location of the virtual point is defined by the user. Additional DOFs can be defined, up to the number of drives. These additional DOFs may be flexible body modes of the table. The intent of adding flexible-body DOFs to the control is to give the system more control authority over the table so it can work to suppress particular vibration modes.

Four tri-axial accelerometers were used as the control inputs of the Tensor 18kN and were placed directly in line with the shakers, as shown in Figure 4. The signal transformation method was used to map the response of the 12 control accelerometers to six rigid body degrees of freedom, with rotations defined about the virtual control point, which was chosen to be the center of the table's top surface. The calculated linear and angular accelerations of the virtual control point were used as the control signals for the six-DOF tests. The signal transformation is given in Equation1 (below), where the left-hand vector contains the six DOFs of the virtual point to be controlled, and the right-hand vector is the individual accelerometer responses. The geometry of the accelerometer locations is used to convert the measured linear accelerations to three translations (X, Y, Z) and three rotations ($R_x$, $R_y$, $R_z$) about the three orthogonal axes assuming rigid body motion.

$$\begin{Bmatrix} X \\ Y \\ Z \\ R_x \\ R_y \\ R_z \end{Bmatrix} = \begin{bmatrix} 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 \\ 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 \\ 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 \\ 0 & 0 & 1.23 & 0 & 0 & 1.23 & 0 & 0 & -1.23 & 0 & 0 & -1.23 \\ 0 & 0 & -1.23 & 0 & 0 & 1.23 & 0 & 0 & 1.23 & 0 & 0 & -1.23 \\ -0.62 & 0.62 & 0 & -0.62 & -0.62 & 0 & 0.62 & -0.62 & 0 & 0.62 & 0.62 & 0 \end{bmatrix} * \begin{Bmatrix} x_1 \\ y_1 \\ z_1 \\ x_2 \\ y_2 \\ z_2 \\ x_3 \\ y_3 \\ z_3 \\ x_4 \\ y_4 \\ z_4 \end{Bmatrix}$$

The signal transformation scheme, in a certain sense, is a MIMO control-averaging method. For a single-axis vibration test, in general, it is considered acceptable practice to average the response of multiple control accelerometers to achieve proper response of a given system and control through vibration modes. This single-axis averaging is done in the frequency domain on the magnitude of the response power spectral density (PSD) at each frequency. Since the PSD contains only the magnitude of the spectrum, no consideration is given to the phase of the response signals. In a MIMO vibration test, however, the signal transformation scheme must consider both the magnitude and phase of the responses in the time domain. Below the first resonance of the table, the PSD average of the control signals and the signal transformation will produce nearly the same result for translation in a given axis. At resonance frequencies, it is possible that control points will be out of phase with each other, resulting in cancellation in the signal transformations and yielding a lower value for signal transformation than would be measured using the PSD averaging techniques used in single shaker control.

## Vibration Tests

Two specific tests were run on the Tensor 18kN to demonstrate the system's capabilities.

**Test 1**. Modified Version of MIL-STD 810g Composite Wheeled Vehicle Test (Method 514.6, Annex C, Table 514.6C-VI).[7]

- Simultaneous excitation of each linear DOF
- Case 1: Active suppression of three rotary DOFs
- Case 2: Active excitation of three rotary DOFs at a low level
- Bandwidth: 15-500 Hz each axis
- X-axis (longitudinal)
  - Acceleration: 3.2 g-rms
  - Velocity: 10.0 in/sec peak
  - Displacement: 0.08 inches peak
- Y-axis (transverse)
  - Acceleration: 3.2 g-rms
  - Velocity: 16.9 in/sec peak
  - Displacement: 0.15 inches peak
- Z-axis (vertical)
  - Acceleration: 3.2 g-rms
  - Velocity: 15.2 in/sec peak
  - Displacement: 0.13 inches peak

**Test 2**. 5-2,000 Hz Broadband Random Profile

- Simultaneous excitation of each linear DOF
- Active excitation of three rotary DOFs at a low level
- Configured to suppress the first torsion mode of the table
- Bandwidth: 5-2,000 Hz flat profile each axis
- Acceleration: 1.0 g rms each axis
- Velocity: 1.8 in/sec peak each axis
- Displacement: 0.03 inches peak each axis

## Composite Wheeled Vehicle Test Results, Case 1

As noted, this test is a modified version of the profiles detailed in MIL-STD 810g, Method 514.6 Annex C.[7] To remain within the displacement limits of the shakers, the breakpoints below 15 Hz on all three profiles were removed. Two separate cases were conducted for this particular test. Case 1 excited all three linear DOFs (X, Y, Z) simultaneously and actively worked to suppress the rotatary DOFs, roll ($R_x$), pitch ($R_y$), yaw ($R_z$), to a null RMS reference level of 0.70 rad/sec$^2$ (three-DOF excitation). Case 2 controlled the linear DOFs in the same manner; however, now all rotary DOFs were simultaneously excited (six-DOF excitation). The profile for the rotary DOF excitation, in this case, was defined to be a flat profile, with a RMS level of 8.50 rad/sec$^2$ from 15-500 Hz. Note that both cases are full six-DOF tests, because all 12 shakers are being driven to control (excite or suppress) rotations as well as translations.

Figures 5 through 7 show the linear response of each axis, and Figure 8 shows the response of the rotary DOF for Case 1. The graphs of the linear DOF plots the individual acclerometer responses in addition to the response of the respective virtual point DOF. The graph of rotations shows all three rotary DOFs of the virtual point relative to the reference level.

Overall, the system performed very well in the linear directions, with some minor deviation around 35 Hz in the X and Y axes of the

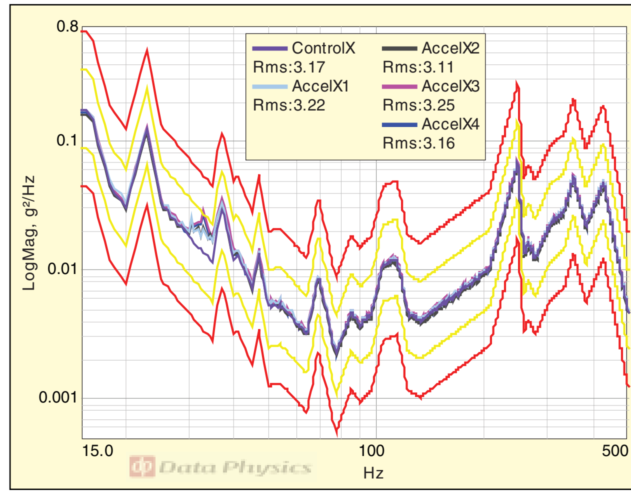


Figure 5. Test 1, Case 1, X-axis response.

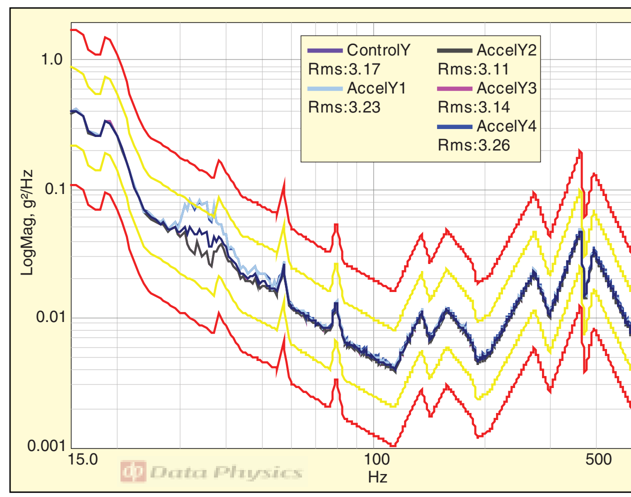


Figure 6. Test 1, Case 1, Y-axis response.

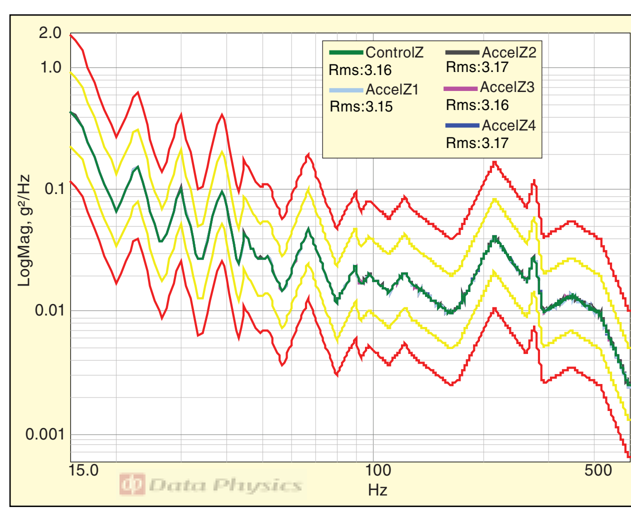


Figure 7. Test 1, Case 1, Z-axis response.

individual accelerometers. Considering the rotary DOF, the control cannot be suppressed down to the reference level, and it begins to diverge below 100 Hz, with a peak at 35 Hz also. This frequency is the shaker preload resonance, and it is difficult to suppress the response with the rotation reference levels set so low. One possible explanation for this is that the reference level for rotation is set below the noise floor of the accelerometers, making it difficult for the controller to resolve the signals well enough for control. Another possible explanation is that the reference level is so low that it is below the physical limits for producing the translational motion without exceeding the small amount of allowable rotation,

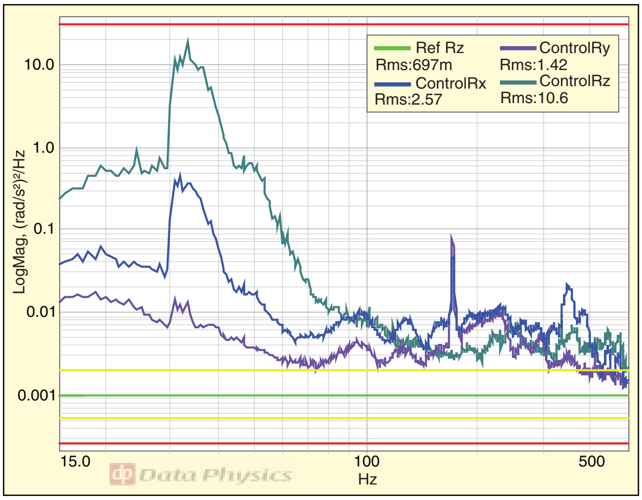


*Figure 8. Test 1, Case 1, rotary DOF response.*

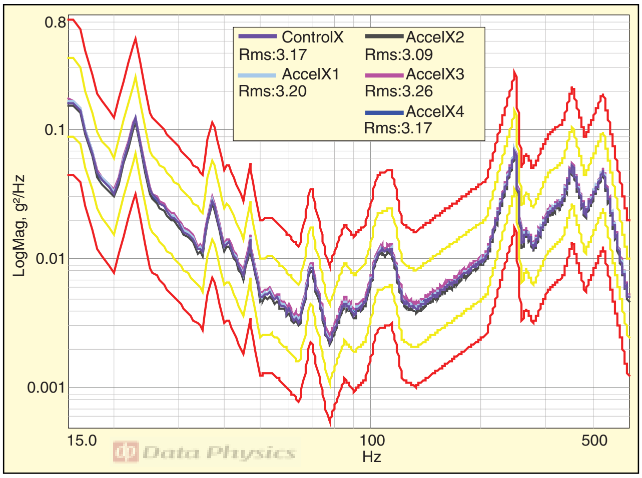


*Figure 9. Test 1, Case 2, X-axis response.*

especially near the preload resonance. Case 2 provides a solution to this problem.

## Composite Wheeled Vehicle Test Results, Case 2

The control of the rotary DOF was altered in Case 2 of the composite wheeled vehicle test so that the rotations were excited to a slightly higher level relative to Case 1. The difference is that now the controller attempts to excite the rotations to a level slightly above the noise floor, rather than suppress them to a null level. It is a subtle difference and can be thought of as a six-DOF excitation test compared to a three-DOF excitation test.

The results for the linear DOF of Case 2 are given in Figures 9 through 11 and the rotary DOF in Figure 12. Exciting the rotations to this level created a significant improvement in the overall control of the system. Now the controller is able to maintain outstanding control over the rotations across the full bandwidth, illustrated by all three rotary responses overlaying the reference level at nearly all frequencies. Note that the yaw ($R_z$) preload resonance at 35 Hz has been suppressed by approximately two orders of magnitude due to this minor change. When comparing Figure 12 to Figure 8, the plots have the same Y-axis scale to highlight the improved rotary response of the system. The change to the rotary reference level also improved the response of the system in the X and Y axes at 35 Hz. The preload resonance in these axes is now very well controlled, and the virtual point and individual accelerometer responses for each axis matches the reference levels extremely well over the entire test bandwidth.

This test provides a clear example of how important it is to properly define a MIMO vibration test and to understand the capabilities of the system. Each shaker input of the Tensor 18kN has an effect on all accelerometer measurements (per the geometric definition of a Tensor – linear mapping), resulting in a closely coupled system. If

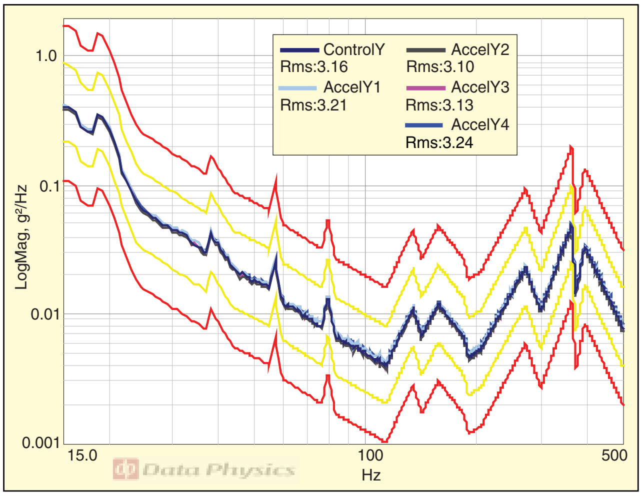


*Figure 10. Test 1, Case 2, Y-axis response.*

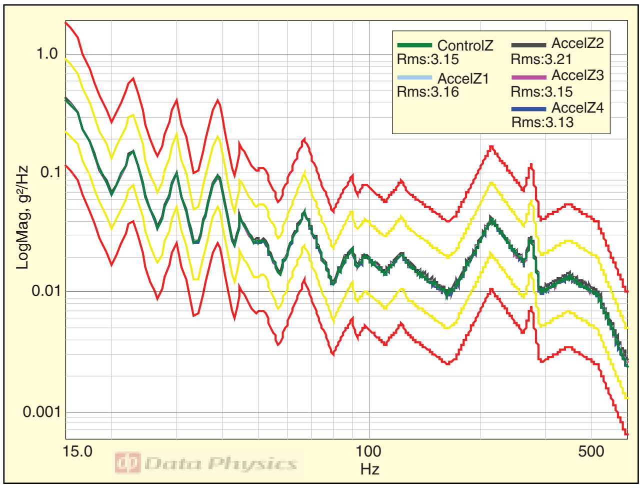


*Figure 11. Test 1, Case 2, Z-axis response.*

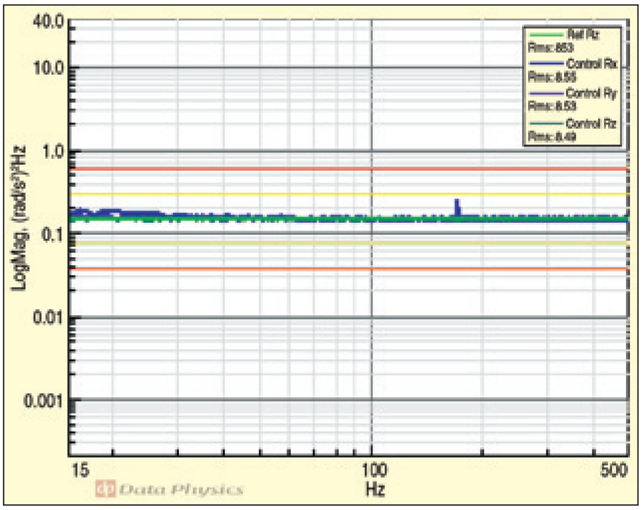


*Figure 12. Test 1, Case 2, rotary DOF response.*

a reference level is set outside of the system's limits (high or low), most likely, the response of other DOFs will degrade as the system is unable to control the DOF with the unreasonable reference. This test demonstrates how a very minor increase in one parameter can significantly improve the control of the overall system.

## 5-2,000 Hz Broadband Random Test Results

The second test under consideration presents the broadband performance of the Tensor 18kN and also highlights its ability to control particular table vibration modes. This broadband test was set up to excite all three linear DOFs simultaneously from 5-2,000

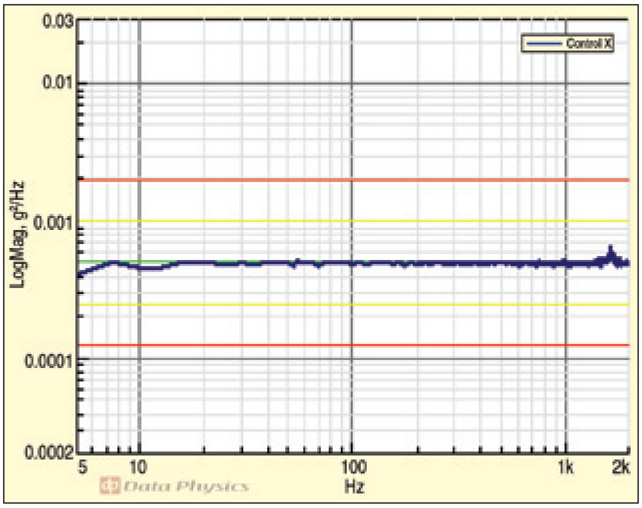


*Figure 13. Test 2, virtual point X-axis response.*

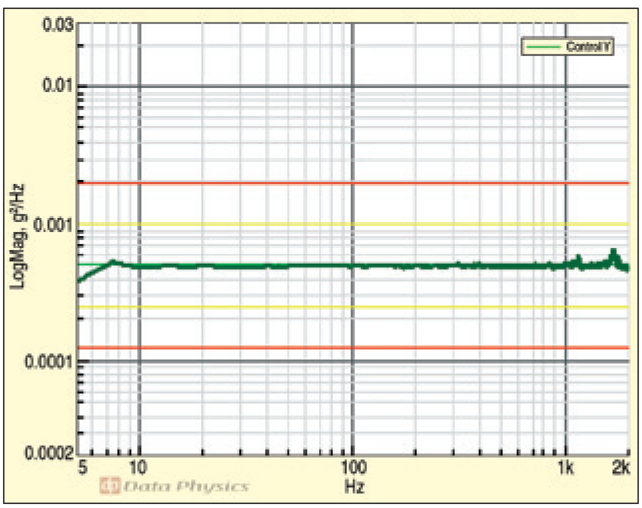


*Figure 14. Test 2, virtual point Y-axis response.*

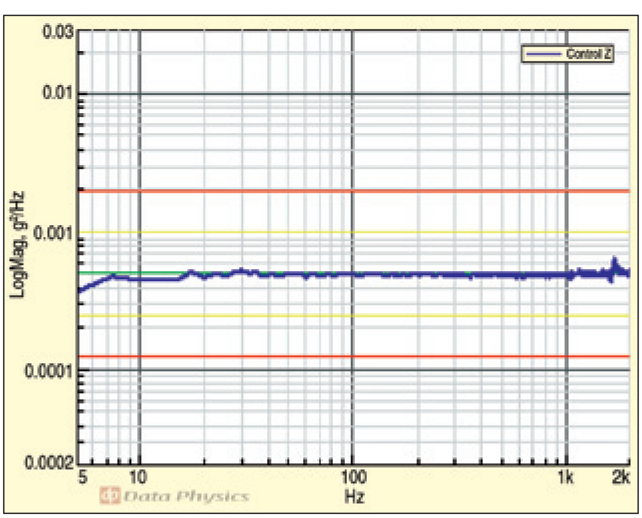


*Figure 15. Test 2, virtual point Z-axis response.*

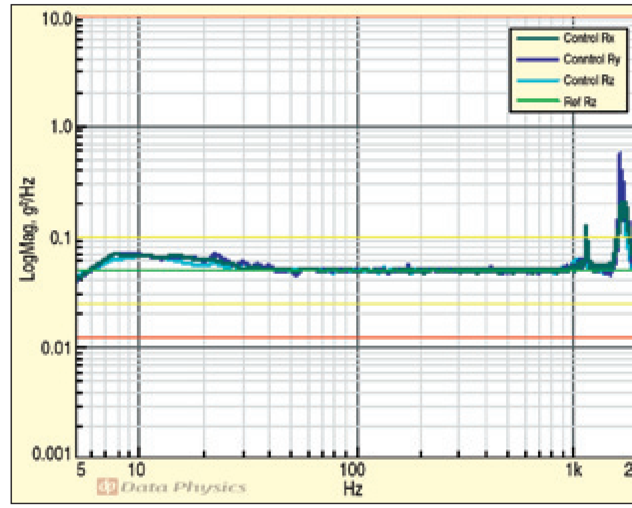


*Figure 16. Test 2, virtual point rotary DOF response.*

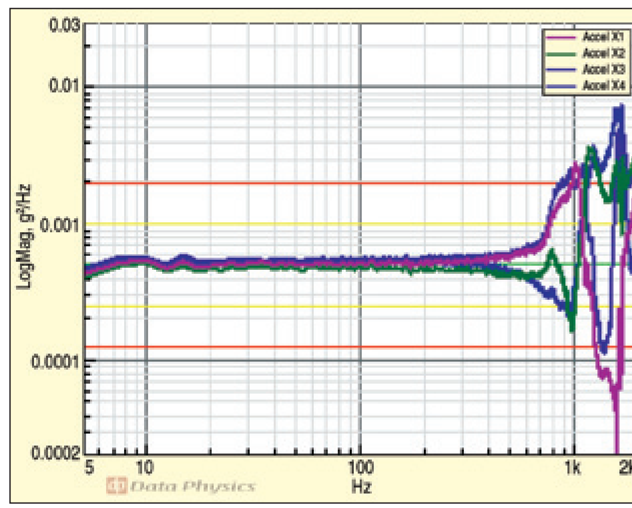


*Figure 17. Test 2, X-axis response.*

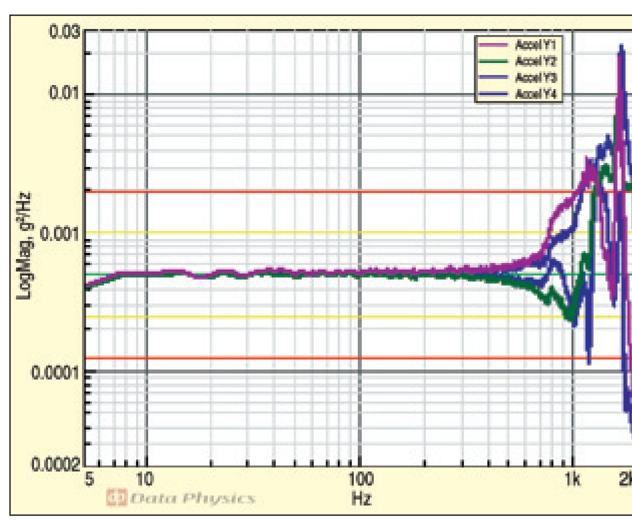


*Figure 18. Test 2, Y-axis response.*

Hz and, similar to Case 2 of the previous test, excite the rotations to a low reference level. The profile of each linear axis was flat, from 5-2000 Hz, with an RMS acceleration level of 1.0 g and an angular acceleration RMS level of 10.0 rad/sec$^2$. The signal transformation method was used for MIMO control.

Figures 13 through 15 plot the response of the virtual point's linear DOF for each axis of the six-DOF test. These plots show that the controller is able to maintain excellent control of the virtual point across the full bandwidth. Figure 16 gives the virtual point's response for all of the rotary DOFs. All three rotary DOFs match the reference level very well over the full bandwidth, except for two higher order modes at 1170 and 1680 Hz, respectively.

For rigid-body motion, it is expected that the individual accelerometer responses will closely match the virtual-point response. However, when the system hits a resonant frequency, the signal transformation method works to control the virtual point to be the average of the accelerometer responses in both magnitude and phase. Using an averaging scheme to control through a vibration mode is commonly done in single-axis testing and is generally accepted because it is extremely difficult and costly to develop a

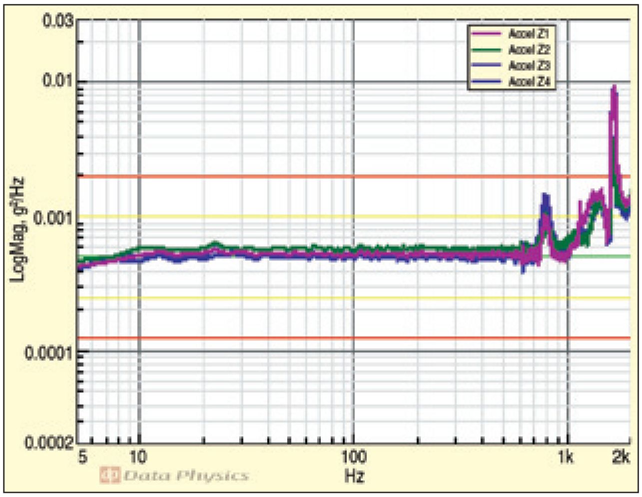


*Figure 19. Test 2, Z-axis response.*

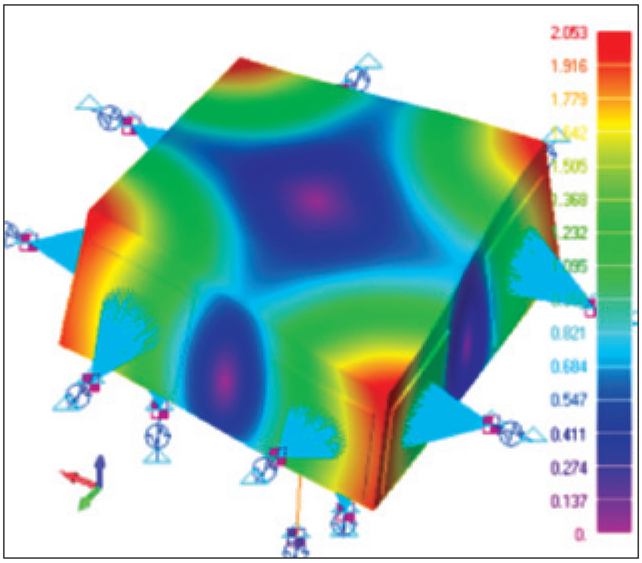


*Figure 20. First flexible body vibration mode shape of table.*

large structure that is resonant free through the full bandwidth of high-frequency tests.

Figures 17 through 19 plot the individual accelerometer responses for each axis, which are used to calculate the virtual-point response in the signal transformation. These plots show nice control up to around 800 Hz. This indicates that the system is resonant free to this point, since the accelerometers closely match the virtual point. The first torsion mode is shown in the Z-axis response at 800 Hz, followed by two other modes at 1,350 and 1,680 Hz. The X and Y responses begin to diverge in this region also, and there are modes dominant primarily in these axes. This is due to the vibration table having significant depth and no longer behaving as a thin plate. The finite-element model of the system shows several in-plane "shearing" and "breathing" modes that support this. Overall, this test produced excellent average control of the defined virtual-point six-DOFs. There are deviations from the reference level of the individual accelerometers at high frequency vibration modes, but this, to a certain extent, is to be expected for this size of a structure. One method for dealing with these resonances is presented in the following section.

## First Torsion Mode Control

From the inception of the Tensor vibration test system, one of Team Corporation's fundamental design goals was to use the redundant shakers to minimize the response of table vibration modes, providing better high-frequency response of the system. On the smaller Tensor 900, this methodology was not required, because the first table mode was above 2,000 Hz. However, this advanced control scheme can be applied to the Tensor 18kN, because the first bare table mode is at 800 Hz due to the significantly larger size. In addition, the Tensor 18kN has considerably more force to react the modes, which makes it a prime candidate to experiment with this complex control. The following section presents the results from an experiment that was conducted to determine the feasibility of controlling the first flexible-body mode of the table.

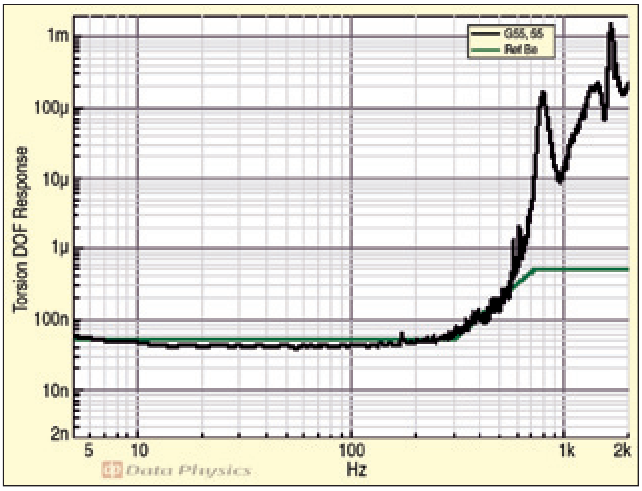


*Figure 21. Torsion mode DOF uncontrolled response and reference level.*

To perform control on the first mode of the table, a seventh DOF that defined this particular mode shape was added to the virtual point. This was implemented in the controller's signal transformation matrix by adding another row to the transformation matrix and virtual point vector as shown in Equation 2 (below). The shape of the table's first vibration mode is shown in Figure 20. For this torsional shape, the transformation can be estimated using only the Z-axis accelerometers. Each accelerometer was given equal weight, but the phasing was chosen so that accelerometers on the diagonals of the table had the same phase and adjacent accelerometers had the opposite phase, which corresponds to the torsion shape. Note that the weighting of the torsion DOF was defined to be less than that of the Z-DOF. This was done to give the Z-DOF more control authority than the torsion-mode DOF.

$$\begin{Bmatrix} X \\ Y \\ Z \\ R_x \\ R_y \\ R_z \\ \varphi_b \end{Bmatrix} = \begin{bmatrix} 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 \\ 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 \\ 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 & 0 & 0 & 0.25 \\ 0 & 0 & 1.23 & 0 & 0 & 1.23 & 0 & 0 & -1.23 & 0 & 0 & -1.23 \\ 0 & 0 & -1.23 & 0 & 0 & 1.23 & 0 & 0 & 1.23 & 0 & 0 & -1.23 \\ -0.62 & 0.62 & 0 & -0.62 & -0.62 & 0 & 0.62 & -0.62 & 0 & 0.62 & 0.62 & 0 \\ 0 & 0 & 0.13 & 0 & 0 & -0.13 & 0 & 0 & 0.13 & 0 & 0 & -0.13 \end{bmatrix} * \begin{Bmatrix} x_1 \\ y_1 \\ z_1 \\ x_2 \\ y_2 \\ z_2 \\ x_3 \\ y_3 \\ z_3 \\ x_4 \\ y_4 \\ z_4 \end{Bmatrix}$$

With the transformation defined, a reference level for the torsion DOF was needed. It was important to define this properly within the system limits (as shown in the previous case study), so the test was run first as a six-DOF test (same as the previous section results), and the response of the torsion DOF was measured. Additionally, running first as a six-DOF test gave a baseline for the control to compare with the results using control of the torsion mode. The uncontrolled torsion DOF response is given in Figure 21 as the black line. Based on this plot a profile was chosen that closely matched the measured response below the first mode, since control for this DOF is not required below the resonance. The first mode is shown in the torsion DOF response as the first peak at 800 Hz. The reference profile was set to ramp up at nominally the same slope as the response and be flat in the high-frequency region above the first mode. The level of the high-frequency plateau of the reference profile was set to be roughly an order of magnitude below the lowest point of the response. The reference level, based on the measured torsion DOF, is shown as the dark green line in the same figure.

With the baseline level known and the reference level defined

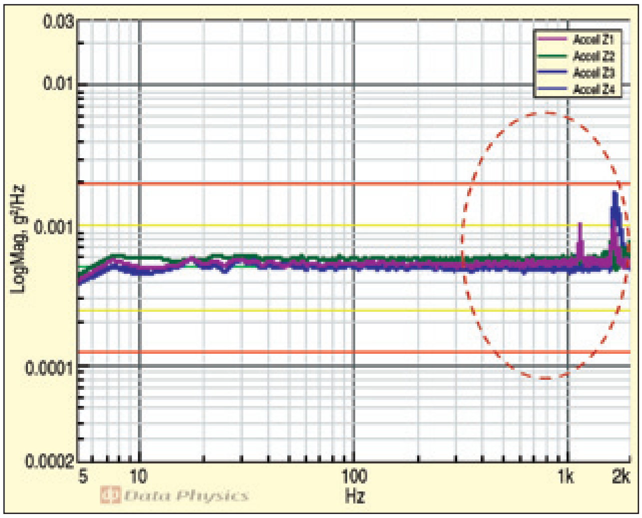


*Figure 22. Test 2, Z-axis response with torsion mode control.*

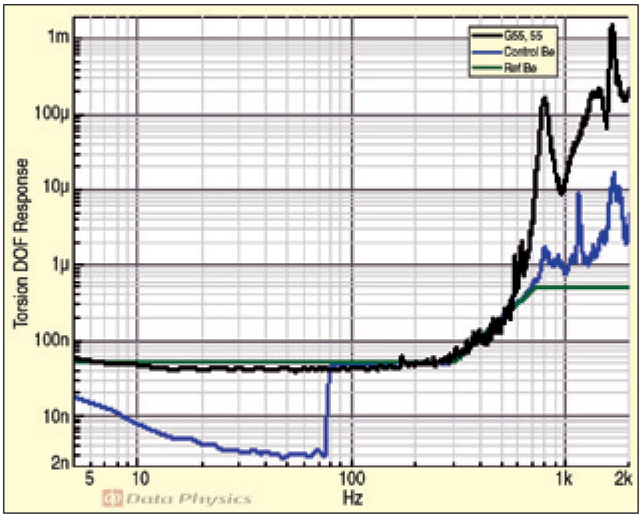


*Figure 23. Uncontrolled and controlled response of torsion DOF.*

for the torsion DOF, the test was run again, this time implementing the torsion mode control. The six-DOF response of the virtual point and the X and Y response of the individual accelerometers (Figures 13 through 18) was nominally the same as the test without torsion mode control, so the plots are not shown again.

The major improvements of torsion mode control were realized in the Z-axis accelerometer responses. Figure 22 plots the Z-axis response using torsion-mode control. Comparing Figure 22 to Figure 19 shows that the 800-Hz peak of the first mode is completely gone, highlighted by the red oval. The vertical ED shakers controlled out the response of this vibration mode completely. This is a remarkable result. Further comparison of the Z-axis response above 1,000 Hz reveals an unexpected result. Implementing the torsion mode control also removed the peak at 1,350 Hz and reduced the peak at 1,680 Hz by roughly 80%. Figure 23 plots the response of the torsion DOF with and without torsion mode control implemented and further illustrates the significant improvements that were attained in this test. The peaks of this DOF at 800, 1,350, and 1,680 Hz were all reduced by approximately two orders of magnitude when comparing the uncontrolled to the controlled response, black and blue lines respectively. These plots clearly illustrate the capability of the Tensor 18kN to minimize vibration modes when implementing torsion mode control to the signal transformation method.

## Future Testing Plans

The Tensor 18kN is a new system capable of performing advanced high-frequency vibration testing. This is demonstrated by the results presented here. However, due to the over-constrained design, it is inherently a complex system to control, and there is still much to learn regarding the nuances that can be applied to advance the system performance. Further testing is being planned to implement additional advances to the control. Considerations for changes to the control are:

- Define additional flexible-mode DOFs – The results presented using torsion mode control were the first attempt at performing this type of control on the Tensor 18kN. In this test, only one additional DOF was added to the signal transformation matrix, with the most significant improvement realized in the Z-axis. The X and Y accelerometer responses did not change in the test presented because they were not affected by the added DOF. The plots of this test showed several high-frequency modes causing peaks in the response of the X and Y axes. Further testing is planned to investigate the application of control to the flexible body modes that affect these two DOFs.
- Apply narrow-band notching – This technique could be applied to decrease the response of the higher frequency modes by notching the reference profiles, reducing excitation at these frequencies.

The testing will hopefully be conducted in the near future on one of the two Tensor 18kN systems currently installed. The intent is to publish the results in a future article.

## Conclusions

The Tensor 18kN is the newest multi-axis vibration test system available from Team Corporation. It expands a novel concept of using 12 electro-dynamic shakers to excite all six degrees of freedom of a vibration table to 2,000 Hz. Team first applied this design in a proof-of-concept system referred to as the Tensor 900. The Tensor 18kN now brings this proof of concept to a size more practical for typical vibration testing applications.

The results presented show that this system produces excellent control of the vibration table using the signal transformation control methodology. Using the over-determined mechanical design, it is possible to control out the first vibration mode of the table and significantly reduce the response of higher order modes. Various subtleties were discussed regarding the control of the system, with suggestions for possible ways to increase performance. This research will continue as the state of the art for vibration testing advances.

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