Sound & Vibration
DOI:10.32604/sv.2020.08909
Article

Fixture Modifications for Effective Control of an Electrodynamic 3D-Shaker System

Kurthan Kersch[1,*] and Elmar Woschke[2]

[1]Robert Bosch GmbH, Powertrain Solutions, Center of Competence for Vibration, Schwieberdingen, 71701, Germany
[2]IFME Institute of Mechanics, Otto von Guericke University Magdeburg, Magdeburg, 39106, Germany
*Corresponding Author: Kurthan Kersch. Email: Kurthan.Kersch@de.bosch.com
Received: 23 October 2019; Accepted: 18 February 2020

Abstract: This work aims to improve the setup of an electrodynamic triaxial shaker prototype with respect to its usability for the automotive industry. Triaxial shakers being capable of meeting the corresponding requirements are not available as standard test equipment. Modifications on the fixture have to be conducted in order to ensure an effective control. The first part of the work is the qualitative description of the system behavior. Therefore, the shaker is treated as a black box. The second part is the modification of the test fixture in order to handle the resonances of the shaker, which is elementary for its usage. A setup is found, that improves testing within the desired frequency range. Thereby, acceleration levels are considered as well as excitation phases and coherences. The proposed setup is used for an exemplary specimen with two different control scenarios. Conclusions are then drawn about the usage of triaxial shakers.

Keywords: Multi-axis testing; electrodynamic shaker; vibration control; fixture design

1 Introduction

Multi-axis testing is widely known as more realistic than performing sequential uniaxial tests. Early investigations of triaxial excitation on aerospace hardware showed nearly twice the fatigue damage from sequential uniaxial excitation [1]. Further, it has been shown experimentally and in simulation, that different maximum stresses, locations of maximum stress and modal participations occur with multiaxial testing [2]. The simultaneous excitation of plate structures results in either higher or lower response energy than uniaxial tests, depending on the excitation level [3]. Nonlinear effects occurring with simultaneous excitation have also been investigated on real electronic components and emphasize the need for multi-axis testing [4]. Different failure modes have been experimentally proven on test specimens [5]. Furthermore, inadequacies in sequential uniaxial testing regarding sequence effects have been investigated [6].

Within the automotive industry, the validation of components is usually conducted on uniaxial shakers. Gradually, the need to perform more realistic testing has arisen. On opposite to the aerospace industry, partially higher acceleration levels and frequency ranges up to 2000 Hz are common for automotive testing profiles. Multiaxial shaker systems with the respective requirements are not available as standard test equipment. A big challenge is the construction of the bearing unit that couples the different shaker axes. For large industrial shakers, such as needed by the automotive industry, low stiffness and thus resonances of the bearing unit are unavoidable which may violate testing results [2]. A six-degree-of-freedom shaker has been investigated, where both occurring system modes were left unconsidered for testing [7]. For every shaker model a qualification test has to be undertaken.

This paper aims to modify the fixture for a given triaxial shaker prototype in a way that vibration profiles can be replicated accurately. For automotive testing, commonly a frequency range of 10 Hz to 2000 Hz is considered. Besides the acceleration levels, also phase and coherence shall be replicated sufficiently. The input profile can be specified as a fully populated Spectral Density Matrix (SDM) [8]. The shaker specifications are not fully known and thus it is considered as a black box. Also the vibration controller can be treated as a black box. Therefore, an initial investigation of the system has to be performed including accelerometer and laser measurements. As a next step, different simple structures are used and adjusted towards an optimum configuration. The effectiveness of the control is then presented with an exemplary automotive component.

2 System Description

The system to be investigated is the IMV TS-3000-3.2H-CE, shown in Fig. 1. It consists of five independent shakers which are coupled through a bearing unit. The horizontal X- and Y-axis are realized with two 15 kN shakers each, working in a push-pull configuration. The vertical Z-axis is excited with a single 30 kN shaker. The maximum acceleration for sine excitation is specified as 200 m/s² and for random as 120 m/s² Root Mean Square (RMS). The maximum displacement is given as 50 mm peak to peak. Furthermore, the shaker table is designed quadratically with an edge length of 320 mm.

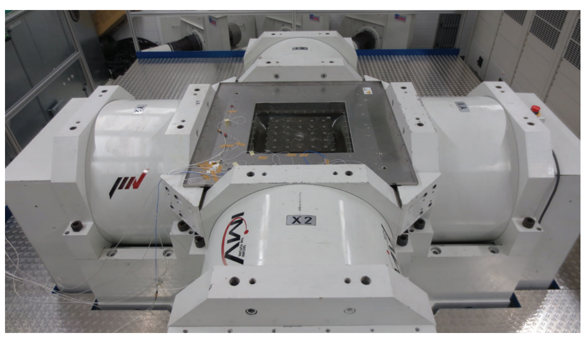

Figure 1: 3D-Shaker system

For the Multi-Input-Multi-Output (MIMO) vibration control, both the IMV K2 controller and the Dataphysics Signal Star Matrix controller were used. The response was measured with triaxial Dytran 3133 series accelerometers and the control was realized using a triaxial PCB 356A61 accelerometer. The data acquisition was performed with an external measurement system and initially, five measurement points were considered according to Fig. 2.

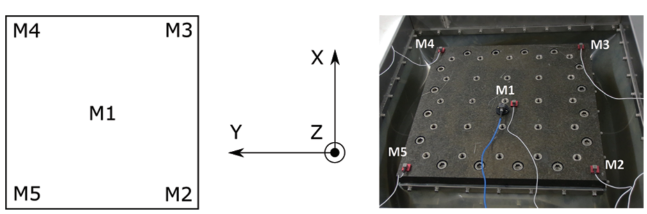

Figure 2: Shaker plate

3 Preliminary Investigation

The most inherent difference of the 3D-shaker compared to ordinary uniaxial shakers is the bearing unit between the shaker axes. Usually, uniaxial shakers only have a low suspension resonance below 10 Hz and an armature resonance around 5000 Hz. Preliminary sine sweep tests show roughly 14 resonances of the bearing unit within a frequency range of 10 Hz to 2000 Hz. The lower modes are mostly a combination of rigid body motion and rotation, between 1000 Hz and 2000 Hz combinations of rigid body rotation and in-plane bending occur. In this frequency range the two first bending modes of the plate can be detected.

Due to the nature of the bearing unit, the shaker axes are not perfectly decoupled and thus excitation always results in cross acceleration. The strongest correlation can be detected between the Y- and Z-axis. Furthermore, the accuracy of a XYZ-excitation is similar to a YZ-excitation, while the other combinations show higher accuracies. It is assumed, that the connection between the Y- and Z-axis is least stiff. Moreover, the 3D-shaker has a non-linear behavior. Fig. 3 shows the responses of a simultaneous excitation of all three axes and the sum of the sequentially uniaxial applied excitation. For the test, a random profile with an RMS value of 15 m/s² from 10 Hz to 2000 Hz was conducted. The table was controlled in the center using the IMV K2 vibration controller. Furthermore, the system slightly changes over time, which will be shown during further procedure. The Z-axis shows minor disturbances caused by current noise probably due to insufficient grounding of its amplifier.

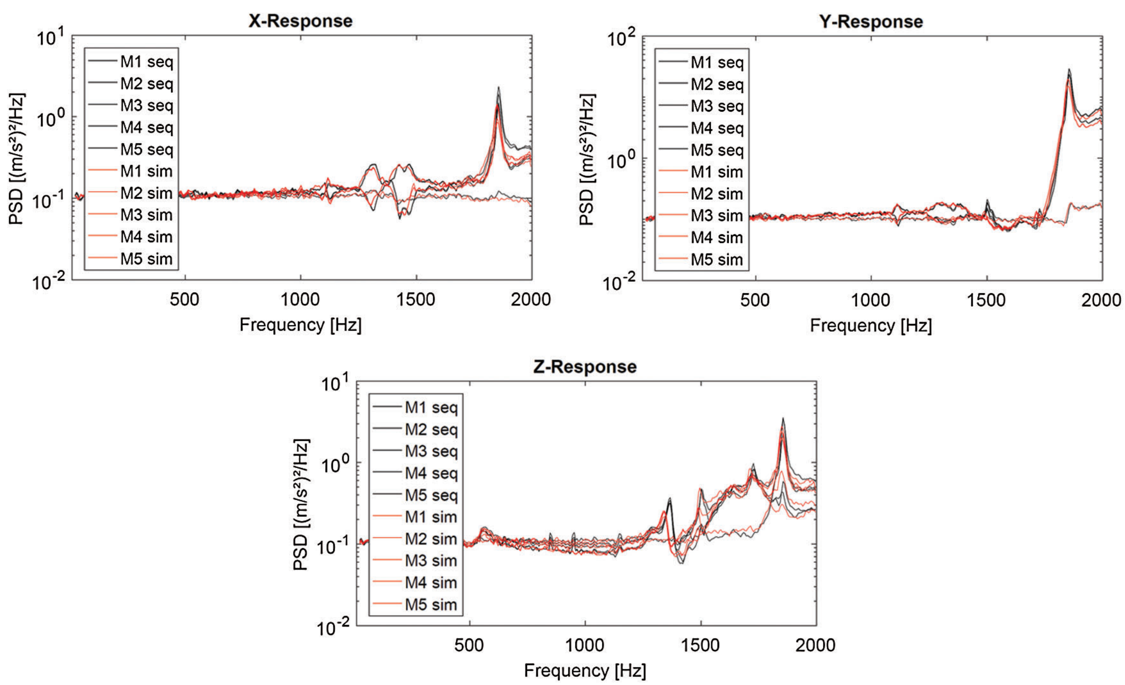

Figure 3: Simultaneous and sequential responses in X-,Y- and Z-direction

As indicated in Fig. 3, the most critical modeshape occurs in the vicinity of 1850 Hz when exciting in Y-direction. For the better understanding, an Operational Deflection Shape (ODS) analysis using a laser scan of the table was performed with a Polytech PSV-400 3D-laser scanning system. Fig. 4 shows clearly that the mode can be described as the second bending mode of the plate in combination with a motion in Y-direction. The reason for the bad controllability is the modal node in which the sensor is placed. It should be mentioned, that the mode was not able to be excited with an impact hammer. Thus, an operational analysis has to be

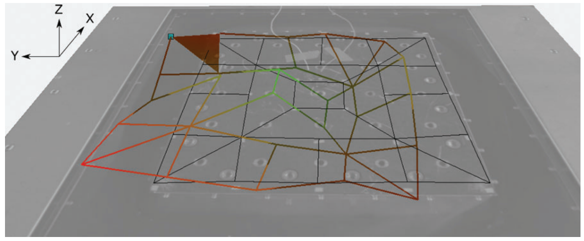

Figure 4: Laser vibrometry of second bending mode at around 1850 Hz

accomplished for its detection. Furthermore, the bare table is hardly controllable along the edges due to other modes occuring at around 1500 Hz.

In general, asymmetrically mounted masses cause disturbances in the system. The resonances get larger amplitudes and may damage the bearings. Thus, the specimen shall be mounted in the center. Since the response power at arbitrary points on a structure usually deviates from the responses at the control point, the specimen shall also be controlled at a single-point in the center. There can also be a change of phase and coherence when considering different locations, especially influenced by the modeshapes of the shaker.

Due to the system complexity, a characterization of the shaker in terms of Experimental Modal Analysis (EMA) or system identification is not meaningful. Therefore, this paper focussed on the modification of the shaker setup in order to perform optimum measurements up to 2000 Hz.

4 Development of a Fixture

In order to control a central mounted specimen, the shaker plate needs to be modified. An inherent limitation is the number of 25 threaded holes in the table. Different setups shall be investigated and evaluated with respect to their accuracy, which is determined as the mean and maximum deviation of the RMS value of the most imprecise axis at each frequency. It shall be noticed that every setup changes mass, stiffness and inertia of the system at a time. All subsequent tests are performed with a simultaneous (XYZ) RMS 15 m/s^2 random profile. Every test is repeated after a total service time of 100 operating hours. For every setup, each screw joint is always applied with a 20 Nm torque. The first measurements were performed with the IMV K2 vibration controller and are more or less corrupted by current noise mainly occurring in the shape of peaks at 850 Hz and 950 Hz. The second measurements were performed with the Dataphysics Signal Star Matrix and are not corrupted by the noise, since the controller was able to compensate it.

As a reference, the accuracy of the bare table controlled in the center is considered. The results are given in Tabs. 1 and 2.

Table 1: Deviation from reference profile for bare table setup

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	9.0% (noisy)	11.4%	299.3%
Max deviation	21.1% (noisy)	44.4%	2319.3%

Table 2: Deviation from reference profile for bare table setup after approx. 100 operating hours

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	6.0%	8.9%	348.9%
Max deviation	14.0%	22.6%	2126.4%

As a next step, a homogeneous plate of 16.8 kg with a height of 24 mm is mounted with 25 screws, as shown in Fig. 5. The accuracies are listed in Tabs. 3 and 4. For the medium frequency range, especially a mode around 1500 Hz causes a worse accuracy in Z-direction. The mode can be described as a rigid body rotation around the X-axis combined with a motion in Y-direction. The plate and the screws lower the response amplitude at the center, where the particular node is located, and thus lead to a worse controllability. The decrease of the Z-response has been verified with a test controlled on the edge for both bare table and plate. At the second bending mode around 1850 Hz the response amplitude is higher than on the bare table. The reason is the increased stiffness of the plate and thus the increased rigid body motion in Y-direction at the center. The behavior has also been verified with a test controlled on the edge for both bare table and plate. Using two mounted plates or plane structures with a comparable mass through the insertion of cavities counters the increase of stiffness and cannot be controlled.

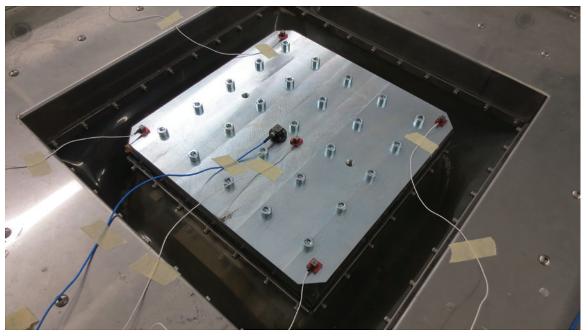

Figure 5: Setup with plate

Table 3: Deviation from reference profile for plate setup

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	9.4% (noisy)	32.8%	206.3%
Max deviation	63.7% (noisy)	310.6%	794.0%

Table 4: Deviation from reference profile for plate setup after approx. 100 operating hours

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	6.3%	33.6%	249.4%
Max deviation	26.4%	152.3%	991.5%

Since a plane setup is not appropriate for a precise control, a block setup is considered according to Fig. 6. The block is assembled out of two pieces, connected with bolts. It has a weight of 8.3 kg, a height of 120 mm and is mounted on the plate with 9 screws with a nominal length of 90 mm and a thread length of 30 mm. The accuracies are listed in Tabs. 5 and 6. The raised position of the control sensor yields an increased response amplitude for the modes, whose nodes are located around the center. Furthermore, the block shifts the second bending mode to the vicinity of 2000 Hz while the modal participation on the table is lowered. The slight improvement of the accuracy in the higher frequency range can be explained by a minor relaxation of the bearing unit, which leads to a higher response in the particular frequency range and thus better controllability.

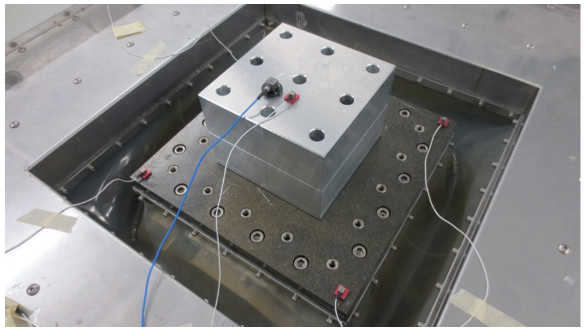

Figure 6: Setup with block

Table 5: Deviation from reference profile for block setup

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	10.3% (noisy)	16.4%	24.0%
Max deviation	35.2% (noisy)	30.7%	37.9%

Table 6: Deviation from reference profile for block setup after approx. 100 operating hours

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	8.1%	21.7%	14.8%
Max deviation	21.7%	33.2%	32.2%

The setup can be enhanced by adding two frames around the block in order to further stiffen the plate. It is shown in Fig. 7. Thus, the response amplitudes along the edge can be reduced for several modes. Each frame has a weight of 2.5 kg and a height of 25 mm. For the mounting, 16 screws with a nominal length of 60 mm and a thread length of 30 mm are used. The accuracies are slightly better than given in Tabs. 5 and 6.

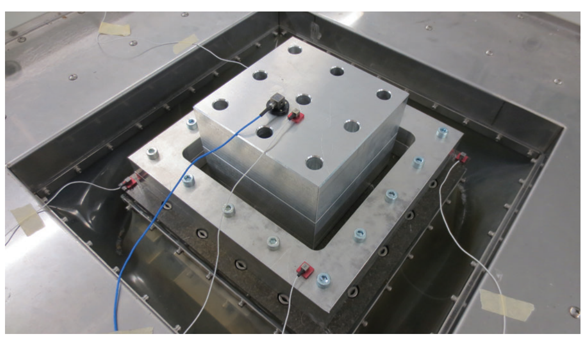

Figure 7: Setup with block and frame

5 Test with Exemplary Structure

The difference in test accuracy shall be demonstrated on an exemplary structure, a hot-film air mass meter (HFM) together with its fixture. The setup has a total weight of 1.7 kg. The main fixture plus the HFM has a weight of 0.7 kg, the adapterplate a weight of 1.0 kg. Generally, it depends on the fixture and its mounting point to the specimen where the control sensor position is most appropriate. For this demonstration two scenarios shall be considered, a control position on bottom of the fixture (see Fig. 8) and a control position on top (see Fig. 9). The vibration control is accomplished with the Dataphysics Signal Star Matrix. The results in Tabs. 7 and 8 show a sufficient accuracy for the control on top, but an absolute imprecise accuracy for the control at the bottom.

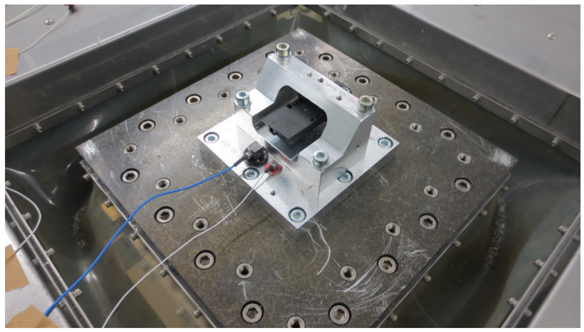

Figure 8: Specimen mounted directly on shaker plate, control at bottom

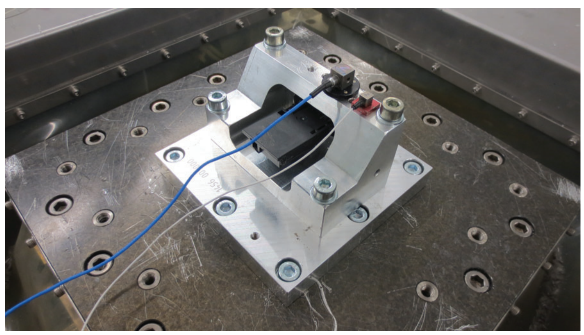

Figure 9: Specimen mounted directly on shaker plate, control on top

Table 7: Deviation from reference profile for specimen mounted directly on shaker plate, control on bottom

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	4.2%	10.0%	665.3%
Max deviation	13.4%	20.7%	3698.6%

Table 8: Deviation from reference profile for specimen mounted directly on shaker plate, control on top

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	4.2%	7.2%	16.4%
Max deviation	16.6%	15.2%	35.5%

If the control shall be conducted at the bottom, a further modification of setup has to be used. Therefore, the HFM main fixture is mounted on the composition out of block and frames, shown in Fig. 10. The results are given in Tab. 9.

Also in case of control on top of the fixture, slight improvement can be achieved with using the proposed composition out of block and frames (see Fig. 11). The results are given in Tab. 10 and can be explained by the further raised position of the control point, where higher responses occur.

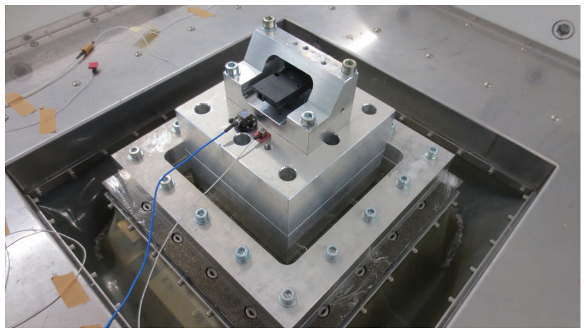

Figure 10: Specimen mounted on block with frame setup, control at bottom

Table 9: Deviation from reference profile for specimen mounted on block with frame setup, control at bottom

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	5.7%	16.3%	15.3%
Max deviation	18.1%	23.6%	27.7%

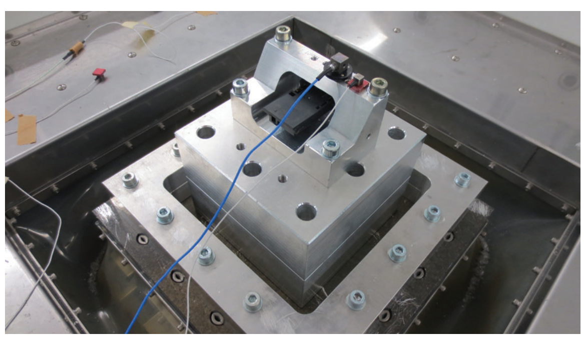

Figure 11: Specimen mounted on block with frame setup, control on top

Table 10: Deviation from reference profile for specimen mounted on block with frame setup, control on top

Frequency range	10–1000 Hz	1000–1500 Hz	1500–2000 Hz
Mean deviation	5.3%	9.5%	7.6%
Max deviation	15.4%	19.4%	14.6%

When using the proposed setup (Fig. 11), a phase control without disturbances by the modes can be achieved, shown in Fig. 12. The input phases −120°, −120°, 0° between the axes are each applied with a coherence of 0.5. Thereby, the SDM gets nearly negative definite which pushes the controller to its limits as can be detected by the deviation of the respective phases from −120°. Constellations with a coherence higher than 0.5 are not able to be reproduced with the given phases. However, the achievable phase and coherence control is sufficient to reproduce typically measured field data.

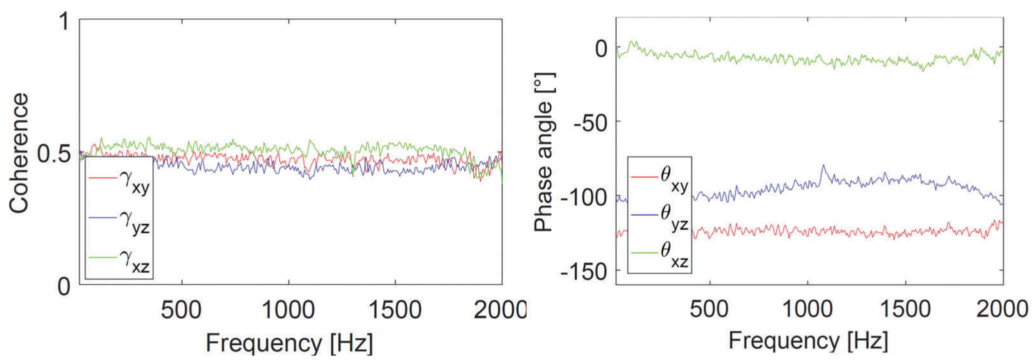

Figure 12: Phase and coherence output for input phases −120°, −120°, 0° and input coherences of 0.5

6 Conclusion

This paper aimed to improve the setup of an electrodynamic triaxial shaker prototype. Preliminary investigations of the system showed a complex and changing behavior. Thus, the main focus was put on the modification of the test setup in order to ensure an effective control. Different setups and control positions were examined sequentially. A test setup was proposed that enables more accurate testing conditions than on the bare shaker table. Within the scope of the investigation, the shaker has been monitored over a time period of 100 operating hours in total.

Especially for large shakers, such as used within the automotive industry, a modification of the test setup might be necessary. Bearing resonances lead to worse controllability and can be severe to the system. Preliminary considerations should be accomplished when designing a test fixture. Different configurations should then be tested consecutively, evaluating their influence on the system behavior. During the fixture design process, the particular specifications of the shaker, such as maximum total mass, have to be considered. The investigated system showed a high amount of modes, especially above 1000 Hz. Generally, the system should be excited carefully and with awareness of the particular modes. Sine sweeps over system eigenfrequencies are rather not appropriate for testing and must be avoided.

Further investigations can be conducted on multipoint control strategies for both single and several specimens, that are distributed on the table.

Funding Statement: The author(s) received no specific funding for this study.

Conflicts of Interest: The authors declare that they have no conflicts of interest to report regarding the present study.

References

1. Himmelblau, H., Hine, M. J. (1995). Effects of triaxial and uniaxial random excitation on the vibration response and fatigue damage of typical spacecraft hardware. *Proceedings of the 66th Shock and Vibration Symposium*, *Biloxi, MS*.

2. Gregory, D., Bitsie, F., Smallwood, D. O. (2008). Comparison of the response of a simple structure to single axis and multiple axis random vibration inputs. *Proceedings of the 79th Shock and Vibration Symposium*, *Orlando, FL*.
3. Nelson, G., Jacobs-O.'Malley, L. (2014). Comparison of multi-axis and single axis testing on plate structures. *Proceedings of the 85th Shock and Vibration Symposium*, *Reston, VA*.
4. Ernst , M., Habtour, E., Dasgupta, A., Pohland, M., Robeson, M. et al. (2015). Comparison of electronic component durability under uniaxial and multiaxial random vibrations. *Journal of Electronic Packaging, 137,* 011009.
5. French, R. M., Handy, R., Cooper, H. L. (2006). A comparison of simultaneous and sequential single-axis durability testing. *Experimental Techniques, 30(5),* 32–37. DOI 10.1111/j.1747-1567.2006.00083.x.
6. Whiteman, W. E., Berman, M. (2001). Inadequacies in uniaxial stress screen vibration testing. *Journal of the IEST, 44(4),* 20–23. DOI 10.17764/jiet.44.4.f72822w825r1156j.
7. Smallwood, D. O., Gregory, D. (2008). Evaluation of a six-DOF electrodynamic shaker system. *Proceedings of the 79th Shock and Vibration Symposium*, *Orlando, FL*.
8. Underwood, M., Ayres, R., Keller, T. (2010). Some aspects of using measured data as the basis of a multi-exciter vibration test. *Proceedings of the 28th IMAC, Jacksonville, FL*, 939–954.