# Installation and Performance Testing of a High Frequency 6-DOF Shaker Table

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*Abstract:* **A CUBE$^{TM}$ High Frequency 6-DOF Shaker has recently been installed at the KULeuven Noise and Vibration Laboratory. This paper describes the (pre-) installation phase and reports on the first results of an extensive performance testing program. Site preparation issues, like the lab layout and the isolation mass, performance issues, like the shaker concepts and the hydraulic supply, and results, for shock and vibration testing and the use of Time Wave Replication (TWR), are discussed. The results from the testing program are used to benchmark the performance of the CUBE$^{TM}$ shaker table for a small set of applications within the field of NVH, Durability and Certification testing.**

*Keywords:* **Hydraulic shaker, Shock & Vibration Testing, Multi-Axial testing**

## I. INTRODUCTION

Over the last two decades the development of hydraulic NVH, Durability and Certification testing devices has been marked by a very consistent and performance driven evolution. Modern devices feature more degrees of freedom, more power, more speed and more accurate response, and a broader frequency range of application. Parallel to this hardware evolution the software side has added revolutionary control performance and an exploding amount of analysis tools, making today's hydraulic shakers into highly dynamic, multifunctional integrated test systems. An illustrative example of this evolution is the CUBE$^{TM}$ High Frequency 6-DOF Shaker from Team Corporation. (Figure 1.)

## II. PREPARING THE LABORATORY

### A. Layout (Figure 2.)

Installation of a highly dynamic hydraulic testing device requires a complete site preparation. The laboratory needs to be adapted to ensure clean operation of the system and to allow full performance usage of the shaker. For the KULeuven installation these conditions needed to be fulfilled within the restrictions of the actual existing university building, already over 75 years old. Apart from this, the engineering side added some more specifications and restrictions.

One of the planned research topics, using the CUBE$^{TM}$ shaker table, is multi-axial testing of vehicle suspensions. For this, the configuration is made as such, that a full size vehicle can be mounted with one tire patch on the shaker, the other tires resting on custom made support structures, because the building construction does not allow the shaker table top to be placed at ground level. It is therefore residing at 1.4 meter above the laboratory floor. (Figure 3.)

Next to the testing of vehicle suspensions in a full car setup, quarter car and component testing configurations are planned. For this, two steel base plates with a grid of taped holes, one on each longitudinal side of the CUBE$^{TM}$ shaker table, will be installed. In addition, an overhead crane with transversal and longitudinal mobility is available for lifting loads up to 2 ton.

In order to reduce the noise levels, originating from the shaker system, the hydraulic power supply is located at some distance in a separate sound insulated room to permit undisturbed noise measurements when using the shaker table. To avoid the pipes from visibly running through the whole lab, they are installed in a canal below ground level, allowing additional noise reduction measures. (see paragraph C.) A large double circuit (oil – water primary – water secondary – air) cooling tower is installed right outside the room.

### B. Designing the reaction mass

The most important part of the laboratory preparation concerns the vibration isolation of the shaker table (with payloads up to 450 kg) from the surrounding building structure. In this case, with the uncertainty about the old buildings response, a very save and effective configuration was adopted.

The CUBE$^{TM}$ shaker table is mounted on a reaction mass (Figure 4.) of about 32 ton, suspended by 6 air springs with automatic mechanical leveling system, highly increasing the systems isolation capacity. Some more weight is added by a steel base plate and the non-dynamic mass of the Cube, about 3.5 ton in total. In general a seismic mass, without air springs, is dimensioned about 4 times the dynamic force [1], in this case 82 kN. Considering the presence of the air springs, a factor 4 is thought to be sufficient, yielding a specification of 33,5 ton. With the length and width of the mass restricted to 2,2 and 4,4 meters by regular car track and wheelbase dimensions, the height of the mass restricted to something more than 1,5 meter by the buildings foundation pillars, a maximum volume of 15 cubic meters was available. Cavities needed to be introduced for installing the air springs, reducing the available volume to about 10 cubic meters. To achieve the necessary weight, a special, characteristically 'red colored, hematite highly reinforced concrete with a density of 3,5 ton per cubic meter was used.

The CUBE$^{TM}$ shaker table needs to be firmly connected to the reaction mass. This is done using an intermediate steel base plate. On one side, the base plate is connected to the reaction mass, using 9 anchor rods of which the connection tubes where installed before pouring the concrete, thus forming a solid connection with the mass. On the other side of the base plate, the CUBE$^{TM}$ shaker table is mounted using 44 M16 bolts, also yielding a quite solid connection.

In addition to all above mentioned vibration isolation measures, the air springs are resting on a box-looking, highly reinforced concrete secondary mass, weighing another 15 ton. The floor part of this mass is resting on dedicated foundation pillars, completely separated from the buildings foundations, the upstanding walls of this mass are separated from the laboratory floor by means of a 3 cm foam spacer.

The necessity of this last measure has been confirmed, with observed vibration levels decreasing from primary reaction mass, over the air springs to the secondary mass, while no vibrations are felt on the laboratory floor or the building structure.

### C. *Noise reduction measures*

First of all, the usage of the CUBE$^{TM}$ shaker table for NVH testing demands significant attention to be paid to the noise levels generated by the system. Next to this, measures need to be taken to reduce the reverberation time in the laboratory, due to the ‘hard’ reflecting brick walls.

Three main noise sources can be identified. Firstly the 90 kW 280 bar hydraulic power pack, rated at 76 dBA (manufacturer rating; 1m open field measurement). A second sound source concerns the CUBE$^{TM}$ shaker itself. With a cubic volume of about one cubic meter, vibrating at velocity’s up to 1 m/s and accelerations up to 10g in a frequency range from 0 to 250 Hz the CUBE$^{TM}$ is a serious sound source. A third sound source concerns the hydraulic circuit and components. The noise emitted by the valves and hydraulic pipes can be disturbing for critical measurements.

Next to these three main noise sources, two unexpected additional noise sources where observed. The first one is a quite noisy ventilator on the IST 19” control rack. In general applications, with the controller installed in a separate room, this is not a problem. In the KUL lab however, the controller is in the same room as the shaker. A second unexpected phenomenon is the air release and pumping noise of the air springs leveling system, when the shaker is driven with high accelerations at low frequency (below 5 Hz), yielding a relatively large seismic mass response.

The noise control measures to reduce the noise and to reduce the lab reverberation time are split up in three phases.

A first phase is mainly focused on the power pack. This was placed in a separate room with a sound insulating wall and sound absorbing panels attached to the ceiling.

A second phase, with installation of a second sound insulating wall, partitioning of the hydraulic piping canal with sound absorption panels and adding panels to the back of the steel plates covering this canal, is to follow shortly.

The third phase, tackling the remaining noise problems, will be executed later, after evaluation of the first two noise control phases and with priorities established by the testing requirements.

## III. PERFORMANCE ISSUES

### A. *Shaker Concepts*

The CUBE$^{TM}$ shaker table is quite revolutionary by its fully integrated concept. In contrast with a classic hydraulic test system, the actuators, with separated pistons for each direction of motion, are located on the inside, with the magnesium thick walled outer shell connecting the pistons and thus closing the kinematical chain. (Figure 5)

Hydrostatic bearings are used for the pistons and the piston head, allowing almost frictionless operation. The high performance three-stage hydraulic valves are located in between both piston ends, reducing the hydraulic path to a strict minimum and thus increasing the bandwidth of the shaker system.

The 6 degrees of freedom are realized by jointly or oppositely driving the three actuator pairs, one pair for each orthogonal direction, two degrees of freedom for each direction and actuator pair.

The CUBE$^{TM}$ shaker table specification promises stunning NVH testing performance [2,3]. The dynamic force rates at 82 kN, the table dynamic mass being 590 kg, manufacturer rates for a 450 kg payload are given in Table I.

**Table I : Performance Specification**

| DOF 0-250 Hz | Stroke (mm) | Velocity (m/s) | Acceleration (g) |
|---|---|---|---|
| Longitudinal | 50,8 | 0,96 | 6,8 |
| Transversal | 50,8 | 0,96 | 4,4 |
| Vertical | 101,6 | 0,96 | 5,3 |
| Pitch | 5° | n/a | n/a |
| Roll | 4,5° | n/a | n/a |
| Yaw | 6° | n/a | n/a |

Driving the CUBE$^{TM}$ shaker table up to this high performance is not a straightforward story. First of all, because it is a dynamic system with at the base control level a software - hardware PID combination, accurate reproduction of a demand signal in the range from 0 to 250 Hz is not immediately achieved. The dynamics of both the PID control, not further discussed here, and the shaker dynamics, mainly the oil column resonance, are of influence.

The mathematics for calculating the oil column resonance can be found in [1, 4, 5, 6]. Performing this exercise for the CUBE$^{TM}$ shaker table yields a lowest (middle of stroke) resonance of about 62 Hz for the vertical DOF. (Figure 6.) The roll off after this resonance can clearly be observed on the response of a simple drive signal, but is fully compensated by using TWR [7]. This issue is further discussed in paragraph D.

### B. *Hydraulic Power*

The hydraulic power for the shaker table is provided by a 280 bar, 165 l/min hydraulic power pack. With the operating pressure at 280 bar, the flow rate of 165 l/min theoretically leads to full SDOF performance in the whole frequency range from 0 to 250 Hz. MDOF full performance with this flow rate only becomes available above 25 Hz.

In the current configuration the inertia of the hydraulic power pack response to sudden flow demands was observed to be too large, restricting the dynamics of the system in the maximum flow frequency range from 7 - 17 Hz (450 kg payload). A note must be made that 5g half sine shocks (Figure 7.) were executed for a payload of about 400 kg without any problems; the above restriction clearly concerns the performance range from 75 to 100% full power. A large additional accumulator near the shaker table is currently investigated as solution to this problem.

A smaller modification of the hydraulic circuit currently under investigation is to increase the operating pressure of the return pipe. The current back pressure is too low (75 psi), due to the very unrestricted return flow. The return flow is thereby not using the return accumulator (100-150 psi) of the shaker table and cavitation occurs at the closing of the valves at higher excitation levels. This is heard as a “hammer” knock in the return pipes.

## IV. PERFORMANCE TESTING : FIRST RESULTS

### A. *Vibration isolation*

Further testing is conducted to verify the expected performance of the reaction mass. The resonance frequencies for the 6 rigid body modes are calculated from the spring stiffness and inertia estimates. (Table II.) A first experiment, using sinesweep and step displacement of the shaker as input and reaction mass displacement, measured with a laser vibrometer, as output, was conducted. The force levels of the sinesweep were too low to yield good measurements. The first step experiment, on the Yaw DOF, showed a quite nice sinusoidal decay, but at a frequency of about 2 Hz. Because the output was a single point measurement and differed considerably from the calculated resonances, no DOF could clearly be identified. New experiments will be conducted using 6 DOF position measurements on the reaction mass.

**Table II : Reaction Mass Resonances**

| DOF | Calculated (Hz) |
|---|---|
| Vertical | 1.4 |
| Longitudinal | 1 |
| Lateral | 1 |
| Pitch | 1.67 |
| Roll | 1.47 |
| Yaw | 1.36 |

### B. *Noise levels*

Some noise level measurements were performed before and after phase 1 of the noise control program. Before, with the power pack running, noise levels at the pack were at 96 dB(A) and some 85 dB(A) at the CUBE$^{TM}$ shaker table. After installation of the wall and ceiling treatment in the separate room, levels are at 85 dB(A) inside the room and 63 dB(A) at the CUBE$^{TM}$. Depending on the driving signal and type of specimen this level can increase up to 80 dB(A) and more. Further reduction of the level is expected with phase 2 and 3 of the noise control program.

### C. *SDOF performance curves*

An extensive testing program to verify the standard performance features of the shaker table is planned. Peak velocities are rated to be no less than 0.96 m/s and peak accelerations are ranging from 2 to 9 g depending on the payload mass and centre of gravity, with a minimum vertical peak value of 5 g.

Currently the power pack is operating at something more than 200 bar, to be raised to 280 bar after the testing program and some necessary system modifications, some already mentioned above. On the performance graph (Figure 8.) for the vertical axis, both the 200 and 280 bar curve are plotted.

At present, experiments are conducted to verify the curves. As mentioned before, the dynamics of both the mechanical and the control system play a role in how this is achieved. While it is quite easy to achieve the 10g acceleration for a single sine wave by gradually increasing the demand amplitude until the level is reached, this is not of practical use for experiments. Time Wave Replication is used to estimate the system dynamics, including the specimen under test, and reproduce the target excitation levels. This is discussed in the next paragraph.

### D. *SDOF accuracy*

In this paragraph some first steps are taken in the exploration of the controller, performance and accuracy, and the use of the Time Wave Replication (TWR) toolbox, provided by LMS International.

As previously described the reproduction of target excitation signals (displacement, acceleration or other) on the shaker table is not straightforward. TWR uses FRF models of the system dynamics, identified in a first step, to generate the required drive signals for the shaker, yielding the targeted response.

In practice, for the CUBE$^{TM}$ shaker table with 6 exciter DOF and (in the KULeuven configuration) 12 measurement channels a 6 x 12 MIMO system can be used.

The TWR technique has been applied to a case study: vibration testing for railway equipment according to the NF EN 61373 norm. All excitations specified in the norm are uni-axial acceleration spectra and half sine acceleration shocks. Three different system configurations were compared in the preparation phase: 6 x 6, 1 x 1, 1 x 3.

It was found that the 6 x 6 system, after a successful identification step, became unusable in the target simulation step, with every iteration increasing the excitation level of the DOFs with zero target spectra.

The second system, a SISO implementation, yielded very good control performance and achieved the target spectra with accuracy within 3 dB and less. The measured channel in this configuration however was the displacement of the shaker, measured by the internal LVDT of the shaker table.

Because the target acceleration spectra are specified at the shaker-device interface, a third implementation with two additional accelerometers on the shaker table outer surface was used. This configuration resulted in a stable and very fast converging use of the TWR technique and was adopted for the actual testing. A typical result spectrum graph is shown in Figure 9.

On the drive signal side, a clear amplification is observed above the oil column resonance frequency. Performance is limited by the maximum valve flow, in the effort of the valves to compensate the response roll off by increasing the flow. In all conducted TWR tests, no performance limitation due to this effect was observed. The valves are thereby assumed to have sufficient flow capacity to fully compensate the roll off in the whole working range up to 250 Hz.

## V. APPLICATIONS

The results from the testing program are used to benchmark the performance of the CUBE$^{TM}$ shaker table for a small set of applications within the field of NVH, Durability and Certification testing.

A first application, as previously shown (Figure 3.) is the use of the shaker table for vehicle suspension testing. Performance specifications of the shaker table show a vertical stroke of 10 cm and velocities up to 1 m/s. Both specifications are about one third of the performance of a standard fourposter and damper testing machine (25 cm / 3 m/s), but these last two only provide uni-axial excitation.

Next to this, the shaker table provides excitations up to 250 Hz and beyond, which makes it very useful for structure-borne road noise experiments. In contrast with classic fourposter or dynaroll excitation, it now becomes possible, not only to control the cross-axis (pre-) loading, but to provide controlled multi-axial force excitation at the tire patch. A first experiment, with a full size car mounted with one tire patch on the shaker and the use of multi-axial tire patch force input, is being conducted at present, as part of a larger road-noise project.

As described in the previous paragraph, a series of certification and durability tests for railway applications was performed. The great advantage of a multi-axial testing machine is that excitation directions are switched at the push of a button and no additional setup time is needed, shortening the test time to a very minimum. A full test sequence, containing 9 different tests in 3 excitation directions, with a total runtime of 16.5h was accomplished within 24h, including all setup and TWR identification time. No comparable performance of any other testing device is known to the authors.

## VI. CONCLUSIONS

This paper describes the (pre-) installation phase and reports on the first results of an extensive performance testing program. Site preparation issues, like the lab layout and the isolation mass, performance issues, like the shaker concepts and the hydraulic supply, and results, for shock and vibration testing and the use of Time Wave Replication (TWR), are discussed. The results from the testing program are used to benchmark the performance of the CUBE$^{TM}$ shaker table for a small set of applications within the field of NVH, Durability and Certification testing.

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## REFERENCES

[1] F. De Coninck and S.Tollens. "Design and Construction of a Testrig for Brake Judder Analysis." Diss. KULeuven, 2001.
[2] Team Corporation. Team Cube : Operation & Maintenance Manual. 2002.
[3] John Davis. Estimating CUBE System Performance. Application Note #CUBE-001. 2000. Team Corporation.
[4] Team Corporation. "Oil Column Resonance." Team News 2002: 2-4.
[5] Findeisen D&F. Ölhydraulik. 4, voll. neubearb. ed. Berlin: Springer, 1994.
[6] Anthonis J., Kennes P., and Ramon H. "Development of a Test-Rig with Six Degrees of Freedom to Study Vibrations on Mobile Agricultural Machinery." Journal of Agricultural Engineering Research 77(2) (2000): 155-69.
[7] J.DE CUYPER and M.VERHAEGEN. "State Space Modeling and Stable Dynamic Inversion for Trajectory Tracking on an Industrial Seat Test Rig." Journal of Vibration and Control.8 (2003): 1033-50.

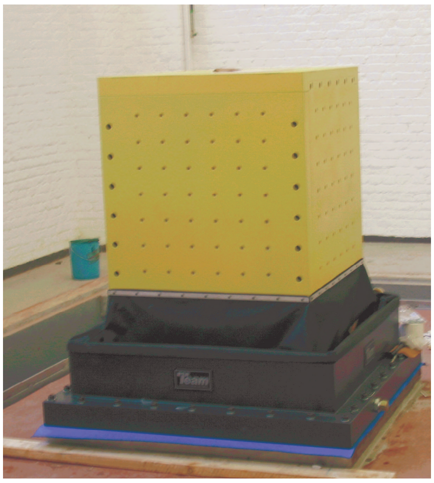

**Figure 1 : Cube Shaker Table**

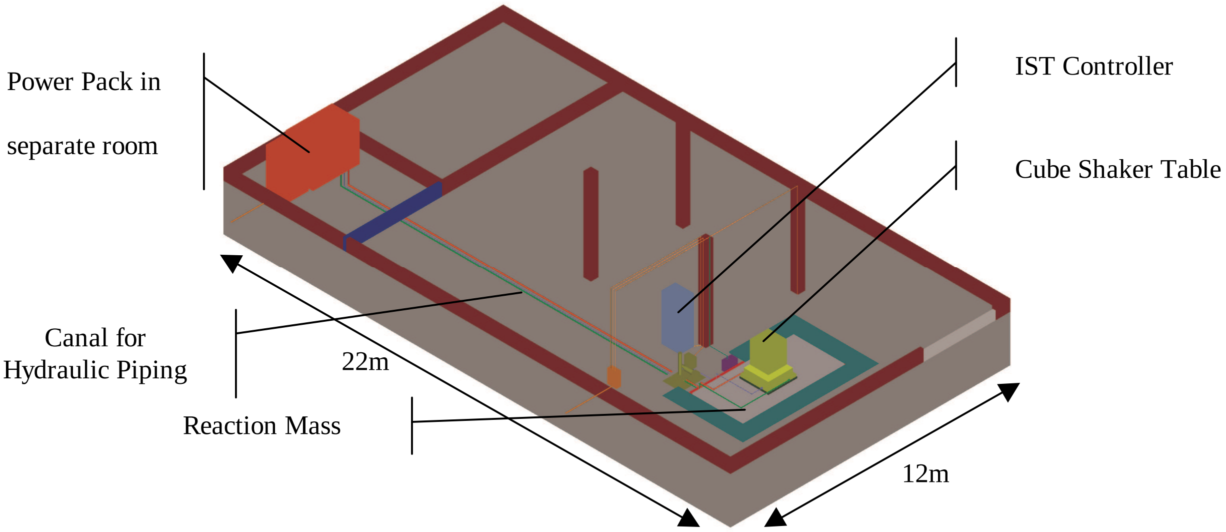

**Figure 2 : Laboratory Layout**

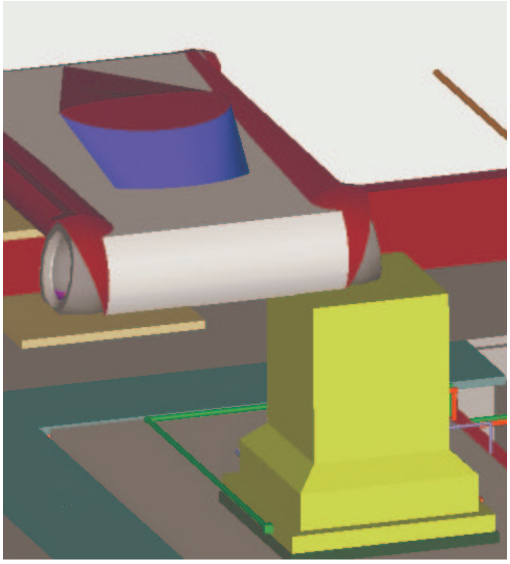

**Figure 3 : Full car setup**

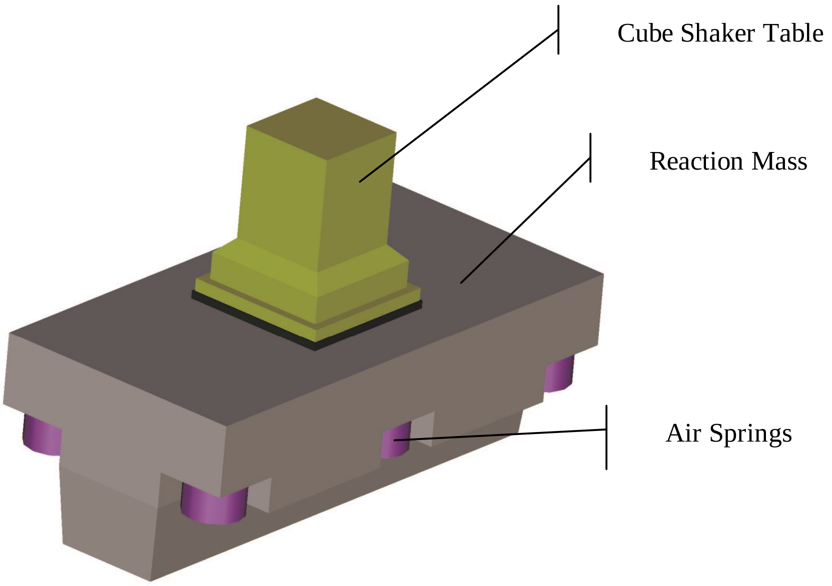

Figure 4 : Reaction Mass

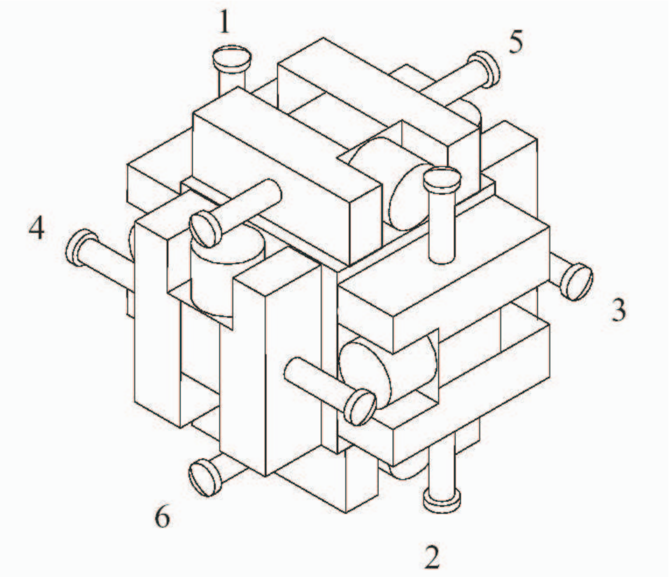

Figure 5 : Shaker Integrated Concept

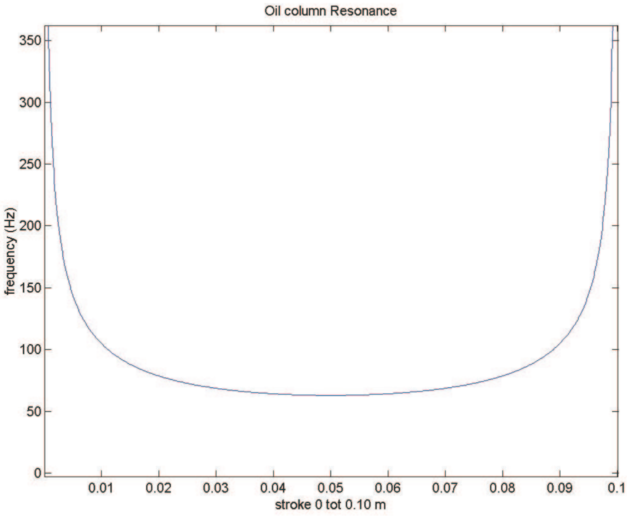

Figure 6 : Oil Column Resonance

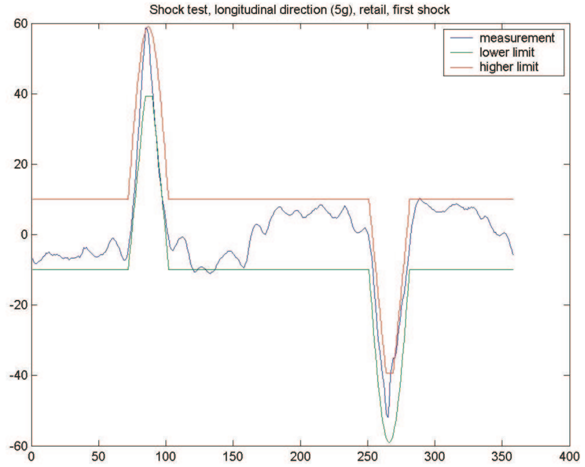

**Figure 7 : Shock Test, Acceleration (m/s2) vs. time (ms)**

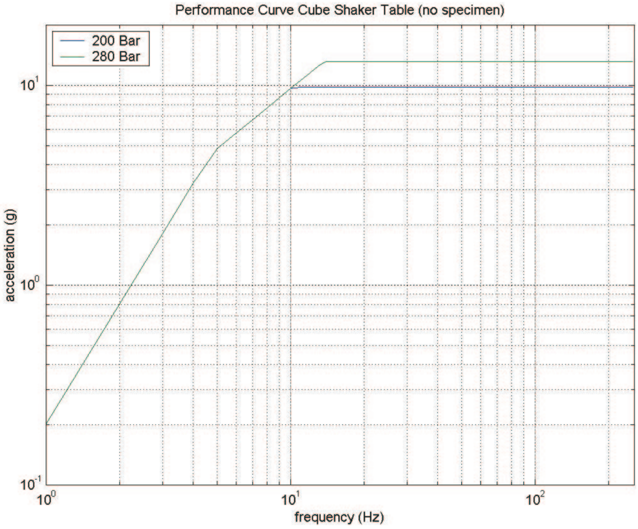

**Figure 8 : Performance Curves Vertical DOF**

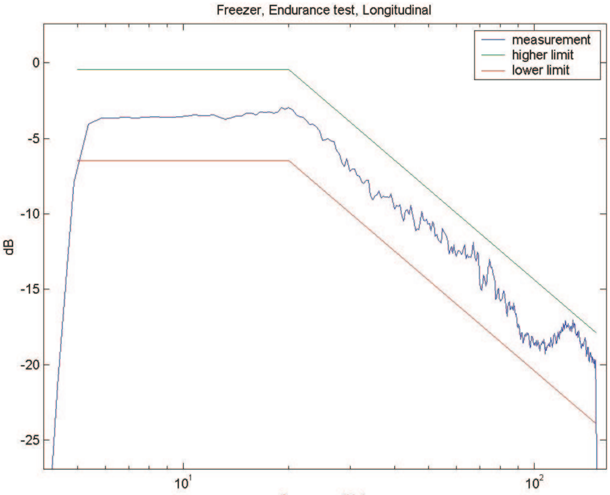

**Figure 9 : Typical Endurance Test Result Spectrum (acc vs. freq)**