Equipment for Accelerated Vibration Testing

Utrustning för Accelererad Vibrationstestning

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Abstract

The increasing complexity with the decrease in size of EEE – components (Electronic, electric and electromagnetic) raises the question on how higher energy frequencies will affect the components and their continuous development. The most common vibration testing equipment currently in use within the automotive industry and SCANIA CV AB are the electrodynamic shaker (ED system). This thesis covers the characteristics of different vibration testing equipment while analysing their strengths and weaknesses, not only for the automotive industry but also including equipment more commonly handled within the aero and space industry. The project aims to find a complement for the ED system and study the possibility for its replacement in the automotive industry.

In particular, experiments are carried out and documented on a so-called "Repeatable shaker system" (RS system) for the purpose to get a better understanding on the functions of the equipment and its overall differences compared to the electrodynamic system when it comes to random vibration testing.

It became clear that complementing or replacing the ED system is difficult and that the RS system work fundamentally different in comparison. Accordingly, the RS system is not a potential replacement for our purpose and it cannot perform at the same level of precision but instead is able to achieve higher energy frequencies overall, making it still ideal for its intended purposes, but not as a replacement of the ED system.

Sammanfattning

Den ökande komplexiteten vid samtida förminskning i storlek av EEE-komponenter (Elektronisk, elektrisk och elektromagnetisk) skapar frågan om hur mer energirika frekvenser kommer att påverka dessa komponenter samt deras fortsatta utveckling. Den vanligaste typen av redskap för vibrationstestning inom fordonsindustrin och SCANIA CV AB är den så kallade elektrodynamiska skakaren (ED system). Denna avhandling täcker de karaktärsdrag som präglar olika typer av vibrationstest-verktyg, samtidigt som en analys inom deras olika styrkor och svagheter genomförs, inte bara från fordonsindustrin utan inkluderat verktyg som är mer vanligt att se inom flyg- och rymdindustrin. Detta med förhoppningen att hitta ett komplement för ED systemet eller något som kan ersätta den helt.

Senare i rapporten finns det dokumenterat från praktiska experiment från ett så kallat ”Upprepat skakar system” (RS system) av anledningen för att få en bättre förståelse i verktygets funktioner och generella skillnader i jämförelse med elektrodynamiska systemet när det kommer till slumpartade vibrationstest.

Det kunde fastställas att komplettera eller att ersätta ED systemet är svårt och at RS systemet fungerar fundamentalt olikt i jämförelse. Detta betyder att RS systemet inte är en potentiell ersättare för vårat syfte och att systemet inte kan prestera på samma nivå av precision, men når istället högre frekvensnivåer generellt, vilket fortfarande gör RS systemet idealt för sitt menade syfte, men inte som en ersättare för ED systemet.

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Nomenclature

ED	Electrodynamic
RS	Repetitive/Repeatable shock
DUT	device under test
RMS	root mean square
Grms	root mean square of an acceleration level
PSD	Power spectral density
ESD	Energy spectral density
SRS	Shock response spectrum
PPDF	Peak probability spectrum
FDS	Fatigue damage spectrum
AFDF	Accumulated fatigue damage factor
FFT	Fast Fourier transform
HALT	Highly accelerated life length testing
HASS	Highly accelerated stress screening
PCB	Printed circuit board
σ	*Stress*
n	*Number of cycles*
b	*Material constant (Basquin law)*
F	*Force*
F_{max}	*Maximum amount of force*
m	*Mass*
Y	*displacement*
v	*Velocity*
a	*Acceleration*
W_{sum}	*Total weight of shaker head, slider and specimen*
P	*Pressure*
P_{ref}	*Pressure reference*
P_s	*Sound pressure*
P_{RMS}	*Sound pressure response, RMS value*
P_{PSD}	*Power spectral response*
f	*Frequency*

f_n	*Natural frequency*
Δf	*Frequency span*
Q	*Transmissibility*
G	*Acceleration in relation to gravitational constant*
G_s	*RMS acceleration due to sound*
G_{out}	*RMS acceleration response*
A	*Surface area*
A_{be}	*Break-even surface area*
ρ	*Density*
h	*Height*
G^2/Hz	*Acceleration spectral density*

1 Introduction

1.1 Background

All types of vehicles and structures are in some way subjected to dynamic forces which can have different effects related to induced vibrations. It is not only the magnitude of the forces which may cause the harm, but the specific ranges of frequencies exciting the structure due to the structure's natural frequencies. As of today, the general frequency spectrum for vibration testing within the automotive industry and SCANIA CV AB roughly ranges from 2 – 2000Hz. This raises some questions when it comes to smaller electrical components. The complexity of electronics are increasing due to the requirement of high efficiency with the decrease in size of the components. With the decrease in its size and increase in reliability, it is crucial to understand how the EEE-components (Electronic, electric and electromagnetic) are affected by higher frequencies. In the field of automotive industry, the induced vibrations come from the common usage of the vehicle and these are in form of random vibration, generated from the movement over the road, then transmitted throughout the truck affecting its parts and components in different ways, depending on the severity and type of the vibrations. These vibration can induce cyclic stresses that affect the EEE-components depending on a number of different factors. Aerospace vehicles are exposed to one of the harshest environments while looking at different types of vibrations compared to other vehicles, but it can be harsh for trucks as well, and even here induced vibrations can have a severe impact on the whole vehicle down to its smallest component.

Widely within the automotive industry, it is accepted to have the upper limit for reliability vibration testing around 2000Hz, and this seems to be, in part, due to the mechanical limitations of most general Electrodynamic shaker. Even though there are electrodynamic shakers that are able to test for frequencies up to 4500Hz, but the drawback is sacrifices to armature size or payload capacity for it to handle a wider frequency spectrum. Adding to that, these types of shakers tend to be very expensive because of extensive cooling systems and are often not as versatile as those with a smaller frequency spectrum because of these types of sacrifices. Therefore, before investing into these specialised equipment, a thorough investigation and comparison of different shaker needs to be made to clarify the shaker can cover all the different requirement for a specific vibration reliability test.

Other methods like acoustic vibration testing are more known within aerodynamic and space industry, but for some specific purposes it may have some uses within the automotive industry, specifically within the high energy frequency spectrum. There are some clear drawbacks already in the fundamentals of acoustic vibration testing for its use in the automotive industry. During impact or shock testing, high energy frequencies are easily achieved, but some concerns raise when it comes to how practical it is for repeatable testing of the method. There are also some obscure and not so well known methods that reach high energy frequency during testing as well, but are often highly specified for a specific area and not suitable for most types of vibration testing within the automotive industry and the applications SCANIA are trying to cover related to vibration testing and fatigue accumulation.

1.2 Objectives

This study will cover different types of vibration testing equipment, and compare these different equipment, looking at their strengths and weaknesses depending on what kind of end result one would want to achieve. The motivation for our cause is to find the best method to analyse high frequencies without sacrificing too much compared to load capacity, fatigue accumulation, aperture size or test configuration complexity and control in test accuracy. This will be achieved by analysing existing vibration testing equipment through literature within the automotive, aero, and space industry followed up by actual physical tests, comparing their strengths and weaknesses against each other. This in all for the conclusion if there are some type of equipment that can replace or complement the electrodynamic shaker.

2 Characteristics of signals

During vibration analysis, there are three types of different signals or oscillations which all have different characteristics and must be treated differently for processing. These three types are:

- Periodic signals
- Random/noise signals
- Transient signal

When working with all types of vibrations, something that is crucial in almost all aspects in vibration analysis are resonance frequency. The resonance frequency is where an object or system will oscillate/vibrate at a greater amplitude compared to other frequencies. When talking about vibrations it is known that they can be divided and characterized dependent on their sources of propagation:

- Acoustic
- Aerodynamic
- Mechanical

The predominance of the different sources is highly dependent on the design of the vehicle and other factors such as launch pad design, phase of flight and the total mission of the vehicle [1].

2.1 Periodic signals

The periodic signals, Figure 2-1, are deterministic in character which mean that no randomness is involved during the timespan of the signal. In other words, every point over its timespan is known. A periodic signal can be, stated from the Fourier theorem, divided into infinite different sinusoidal waves. Describing a periodic signal is done with the use of a linear or effect spectrum [2, p. 189].

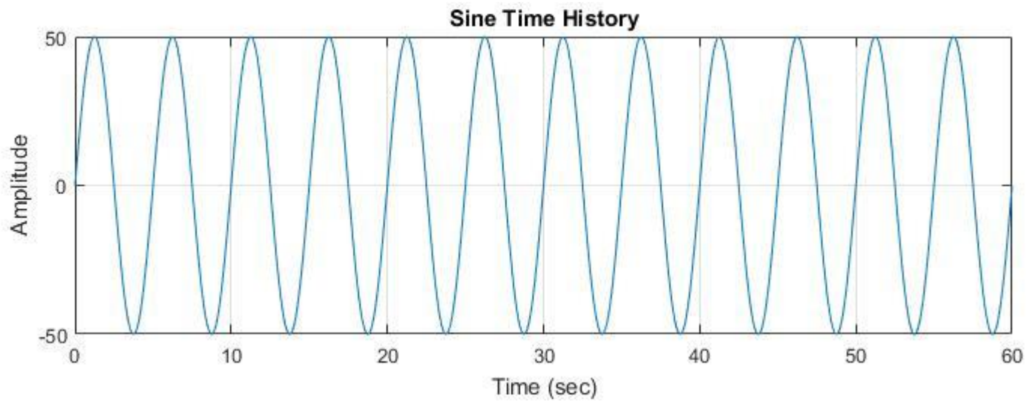

Figure 2-1. A periodic time signal in form of a sine wave at 0.02Hz

2.2 Random Signals

The characteristics of random vibrations are that they are non-periodic, i.e. non-deterministic, Figure 2-2. This means that statistics has to be applied and used for the prediction of a random vibration signal and its continuous motion. Even if it is possible to predict the probability of occurrence of various displacement or acceleration, it is not accurate enough to predict a precise magnitude at a specific time, only its probability.

One of the characteristics of random vibrations is that, when analysing over a given frequency bandwidth all frequencies are present at all times over the time span. This means that when a component is exposed to random vibrations, all its resonance frequencies will be excited at the same time, compared to sinusoidal vibrations where they are excited individually. But most random/noise signals have some consistency, which may be its mean acceleration value or root mean square (RMS) value. A random signal can also be called a stochastic process [2, p. 190].

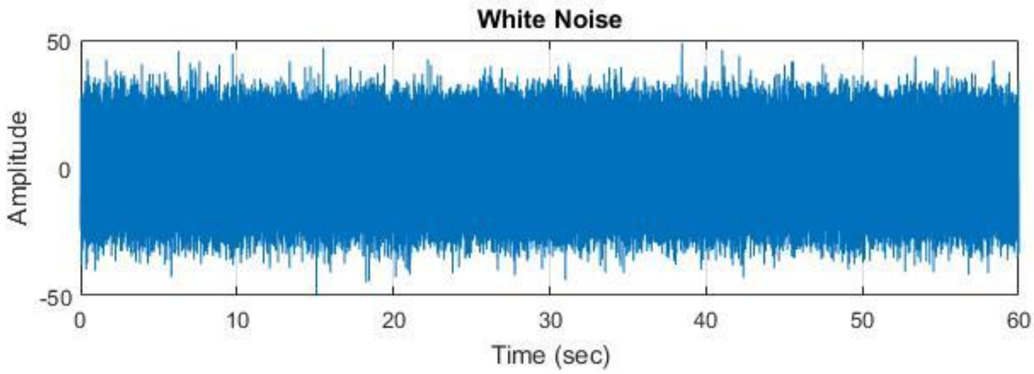

Figure 2-2. Random/white noise **signal generated in MATLAB**

2.3 Transient signals

Transient signals, Figure 2-3, as random/noise signals have a continuous spectra. However, what differentiate a transient from a random signal is that the signal amplitude decreases over time from initial impact, it is not constant over the time interval [3, p. 59].

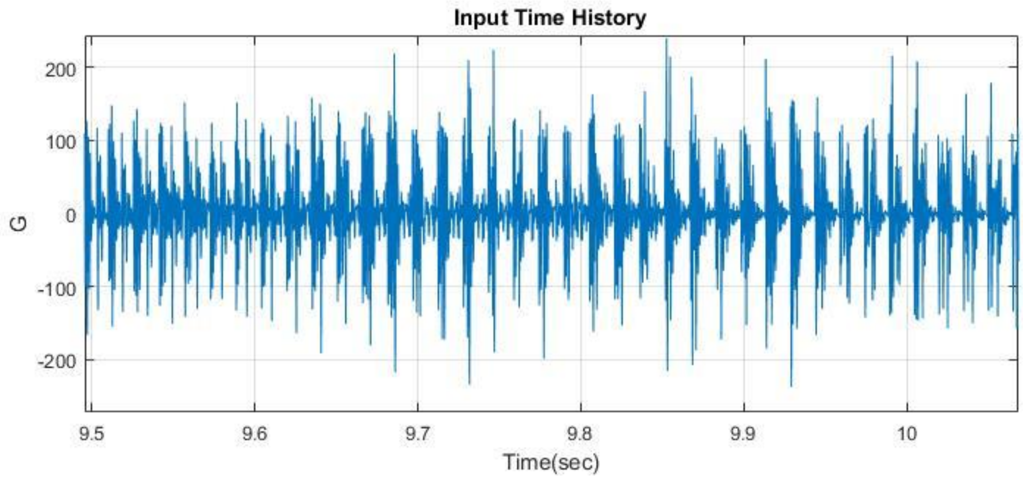

Figure 2-3. Example of a repeated transient signal response from a repeated shock vibration test

3 Theory on vibration signal processing

Depending on the wanted end result is it crucial to know how processing tools work, where they work and the information they mediate. This chapter contains a summary on vibration signal processing and how to analyse a random signal with different processing tools.

3.1 Analysis of random signal

In the analysis of the random vibration input or output signals the root mean squared (RMS) level matters, but it does not mediate all the necessary information for a complete analysis. This is where power spectral density (PSD) is implemented. The magnitude of the PSD in the region of the resonance frequency correlated to the structure is more important. There can be an infinite number of PSD curves with the equal RMS acceleration input, meaning the area under the PSD curve, this means that for a correct analysis of a PSD the acceleration level must be known at its correlating frequency [2, p. 222].

3.2 PSD – Power Spectral Density

Power spectral density is a tool and the most widely used for analysing random vibration signals [4]. This type of analysis results in a plot of average acceleration power per cycle of the analysed bandwidth and is achieved by normalizing the amplitude over the frequency resolution. A PSD shows which frequency variations are strong and which are weak. By integrating within a specific frequency range of the PSD, it is possible to determine the power for that range [5, 6].

3.3 ESD – Energy Spectral Density

The ESD is used to decompose a transient time signal over the signal's energy. In short, the EDS is the correlating PSD times the time duration and is to be interpreted as the area under the curve [3].

3.4 SRS – Shock Response Spectrum

SRS is the maximum response of a single degree of freedom (SDOF) system against a given transient signal. One of the problems with SRS is that different types of excitations responses under a different time interval can produce the same SRS, which means that the fatigue accumulation estimate may not be accurate [4].

3.5 Rainflow counting

Rainflow counting or rainflow cycle counting is a traditional fatigue analysis to decompose a time signal and turn into fatigue cycles [7]. In combination with the rainflow analysis and Miner's rule it is possible to access the fatigue life of a component or structure when even though the loading might be complex [8].

3.6 PPDF – Peak Probability Distribution Function

When analysing a non-Gaussian excitation, those for example generated by a RS system, the characterization of the stress potential is achieved through the use of a rainflow analysis. Then by plotting the data from the rainflow analysis and that plot is called peak probability distribution function. This illustrates the data as bars in a diagram and their respective height represents how probable it is for that stress potential to show during the excitation [4].

3.7 FDS – Fatigue Damage Spectrum

The fatigue damage spectrum is derived from the SRS, for a linear single degree of freedom system (SDOF). Instead of implementing the acceleration response, the displacement or stress response is used. In combination with the number of repeated cycles and the material parameter b, then the fatigue damage can be derived from the material's S-N-curve, the correlation between fatigue, number of cycles and stress is depicted in equation (1). If the signal response is in the frequency domain the FDS is then calculated through a probabilistic approach instead of a deterministic [9].

$$\boldsymbol{FDS} = \boldsymbol{n\sigma^b} \tag{2}$$

Accumulated Fatigue Damage Factor reads as a summation of a number of FDS results:

$$\boldsymbol{AFDF} = \sum(\boldsymbol{n_1\sigma_1^b} + \cdots + \boldsymbol{n_x\sigma_x^b}) \tag{3}$$

3.8 Fast Fourier Transform

The Fast Fourier transform process when used on a time signal, transform it from the time domain into its equivalent representation in the frequency domain. Which then can be used and derive its correlating energy spectrum [10].

3.9 Spectral Resolution

Spectral resolution, commonly called "lines", is the amount of calculated points under the test frequency spectrum. Higher resolution is achieved with higher number of points. The time it takes to collect data from a random vibration profile is directly proportional to the number of lines. It is most common for experts within the field of vibration testing to use 800 lines, doubling this amount doubles the calculation time [11].

3.10 Accelerated life length testing

The expected life length of the EEE-components are sometimes over 1000 hours. This is not ideal for reliability life length testing purposes. Instead, accelerated life length testing is used as an approximation of the real working component environment, only taking the fraction of the time without sacrificing reliability of the end result. The conditions in which a specific material should be tested can be derived from a S – N diagram [12].

3.11 Summary of vibration signal processing

The different processing techniques and analysing methods explained in this section are well known in the area of vibrations and are used depending on specific applications. The ones that will be used most frequently in this paper will be Power spectral density (PSD) for the processing of random vibrations.

4 Different vibration testing equipment

4.1 Electrodynamic shaker - ED

One of the most commonly used equipment for accelerated vibration testing is the Electrodynamic shaker. Basically, the electrodynamic shaker fundamentally works like an ordinary loudspeaker, but the electrodynamic shaker is more robust in comparison. The main working part in the middle of the structure is the coil of wire, going around the ferrous inner cylindrical core along its length in its radial direction, which is fixed to an armature as illustrated in Figure 4-1. When a current travels through the coil it induces a magnetic field making the coil move in the axial direction of the core, which is connected to the shaker head, making it oscillate in the desired harmonic or random motion. In order to accommodate heavier currents the shaker coils use heavier conductors compared to loudspeakers [13].

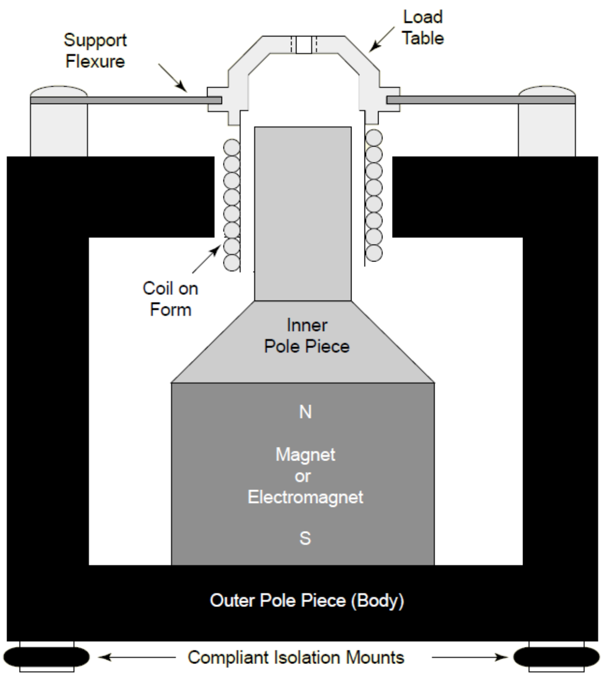

Figure 4-1. The fundamental parts of a typical electrodynamic shaker [13]

When doing a vibration test, it is essential for the machine to be able to analyse and move in the three mutually perpendicular axes of the system. Because the armature only oscillates along one axis a solution is applied. This is achieved through mounting the head on a trunnion that permits the head to be rotated around an axis 90 degrees and locked in position. Where to mount the shaker system itself depends on the general G forces and mass of the DUTs that will be tested. If the machine is to be moved around frequently then the machine should be mounted on vibration isolators. However, this is not ideal with for ED system that are going to handle large masses and high G forces. In these cases, a stationary fixation of the machine is recommended. The ED should be mounted on a large concrete block, at least 10 times the shaker's mass for it not to dissipate its vibration energy by shaking itself. This large mass should be isolated from the rest of the building structure to avoid low-frequency resonance which can lead to damage to the building.

For the fixture, the shaker head often has threaded holes in a pattern where it is possible to mount the test object/specimen by using some kind of jig and machine screws. Large electronic boxes need a type of mechanical adapter, one that is able to transfer the vibrational energy to the box. This type of adapter is often called a vibration test fixture and should have its lower natural frequency about 50% higher than the highest forcing frequency [2, p. 347].

Common types of test on the electrodynamic shaker, as stated before are different types of sinusoidal, random and shock testing [14]. It is easy to justify why one should use the electrodynamic system for standardised or customized vibration testing because of the ability to control during testing, i.e. real time control. The level customisation and level of control during a test becomes more clear when comparing to a repeatable shock system (RS system), sometimes also called a shaker table [15]. Details on RS systems are given in section 4.2.

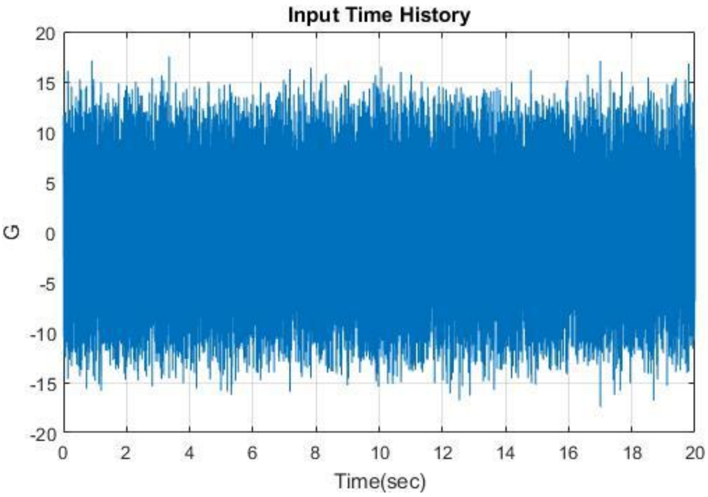

Figure 4-2. Random noise time input signal for an ED system

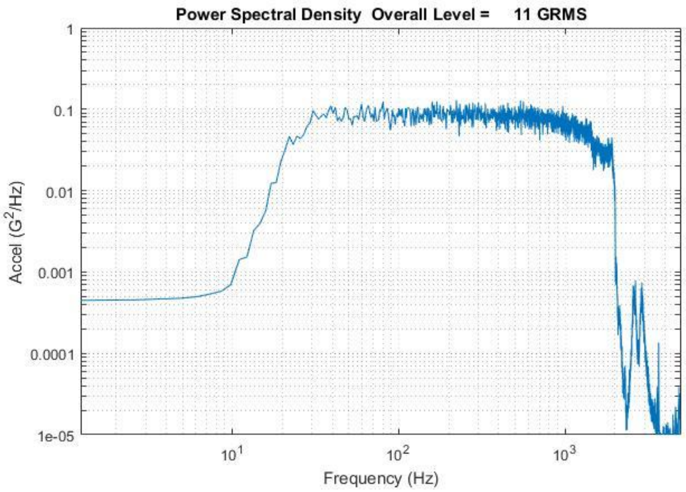

Figure 4-3. PSD response from ED system with the frequency span 10 - 2000 Hz and input at a constant Acceleration at 0.1 G²/Hz

An electrodynamic shaker's function is limited by; displacement, velocity and acceleration. At low frequencies the displacement is the limiting factor, while at high frequencies, the acceleration and velocity are the dominant factors. Most shaker systems have a limit of displacement at 50 mm peak-to-peak distance. Once the machine is to over travel, a safety precaution can be to utilizing sensors, so if disaster strikes the system then shuts down before physical limits are reached [11].

It is typical for an electrodynamic shaker to be operated under frequency ranges between 4 to 2000 Hz while the shaker head resonance is above 3000 Hz. But some types of electrodynamic shakers can operate frequencies up to 3000 Hz. These types of shaker are generally rated in terms of peak force, based on its sinusoidal motion with peak forces between 100 to 100,000 N depending on the weight of the system. However for a higher peak forces a dual shaker system can be used. Peak velocity can be up to 2.5 m/s for an ED and the velocity determines how wide its range are for shock pulses [2, p. 346].

4.1.1 Acceleration force capability of ED shaker with mathematical formulas

The limitations of the shaker system can be derived from Newton's law of motion

$$\boldsymbol{F = ma} \tag{4}$$

The G capability for a specific shaker reads:

$$\boldsymbol{G_{peak} = \frac{F_{max}}{W_{sum}}} \tag{5}$$

The displacement, velocity and acceleration are also given by:

$$\boldsymbol{Y = Y_o \, sin \, \omega t} \tag{6}$$

$$\boldsymbol{v = \dot{Y} = \omega Y_o \, cos \, \omega t} \tag{7}$$

$$\boldsymbol{a = \ddot{Y} = -\omega^2 Y_o \, sin \, \omega t} \tag{8}$$

How to calculate the Grms from correlating PSD:

$$\boldsymbol{G_{RMS} = \sqrt{\frac{\pi}{2} P_{PSD} f_n Q}} \tag{9}$$

4.2 Repetitive shock system – RS

The Repetitive vibration system is designed for the specific purpose, to as fast as possible accumulate stress through fatigue and make the weakest parts of the design to fail [4]. This means that by decreasing the testing time by faster fatigue accumulation leads to lower costs and higher customer satisfaction.

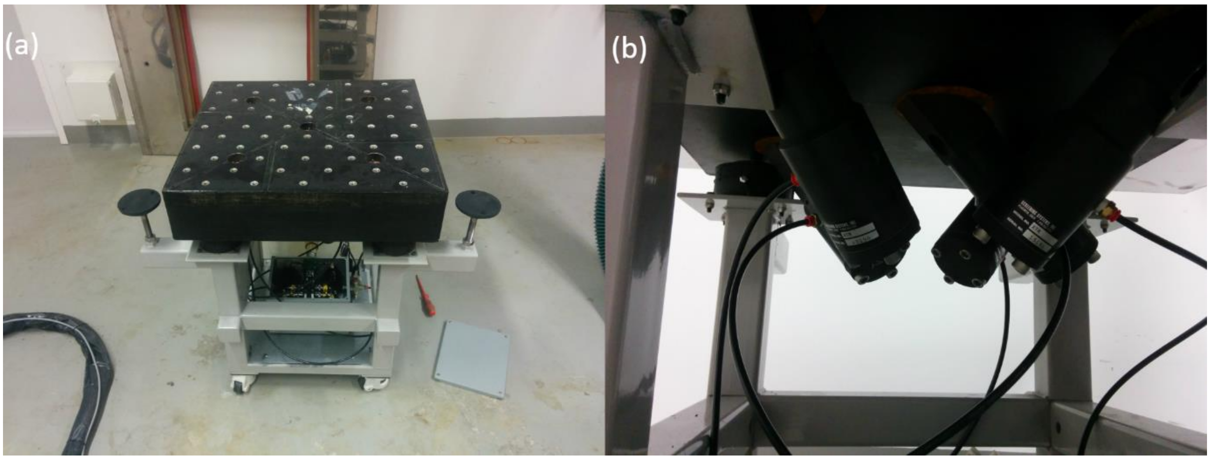

Figure 4-4. (a) A RS - System with showing the table top, (b) Pneumatic actuators shown attached under the table

This type of vibration testing equipment employ air pressure driven impact actuators that are attached to the bottom of a table. These attached pneumatic driven impact hammers are then oriented so that they are able to transmit energy through vibration in the x, y and z direction, and rotational leading to a “six degrees of freedom” system (6-DOF) [16]. As stated before, RS systems is generally used for its ability for reducing defect detection time and precipitation time, and are in many cases in combination with a temperature chamber in a HALT/HASS system. More detailed description about HALT/HASS can be found in section 4.5.

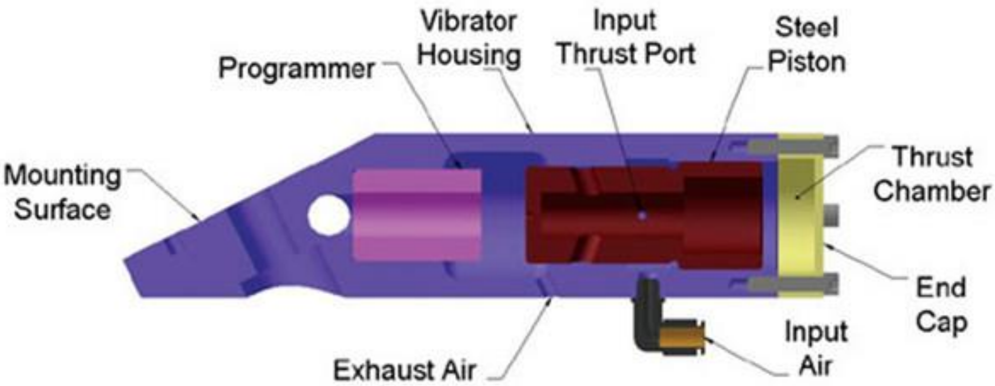

Figure 4-5. RS Actuator design [17]

An ED vibration system during standard random vibration testing can be controlled very precisely from a custom shaped random PSD to meet specific stress goals. The spectral profile generated from a repetitive shock table however is referred to “pseudo-random” and unlike the ED system, the RS system’s spectral profile is not controlled in real time, but is dependent on the table, its fixture and the product response to the impactor hammers [16]. The RS system is controlled with a closed feedback loop strategy that controls the max value of the Grms. The accelerator measuring the response from the vibration table feed the information to a DSP controller, whereas the controller then adjust the voltage, which in turn is converted in the convector from input voltage to output air pressure. This means that each pneumatic hammer works at the expected value for a given input to the system [18]. By analysing these excitations from Electrodynamic machines and Repetitive shock machines, it can be stated that the excitations of an ED system are of Gaussian character while RS vibration system excitations are non-Gaussian. This means that the assumptions (about RMS and PSD analysis) used for ED systems cannot be applied for an RS machine when analysing the ability for fatigue of a component.

The RS is able to attain higher G levels and handle more mass than most ED systems. The main disadvantage is that the frequency distribution of the vibration spectrum cannot be controlled to the same extent or precision as an ED system [4]. As stated in literature [19], the RS are effective in screening small objects due to its high power output at high frequencies, which is totally opposite of traditional shakers. Example of objects which are susceptible to high frequencies are small electrical components.

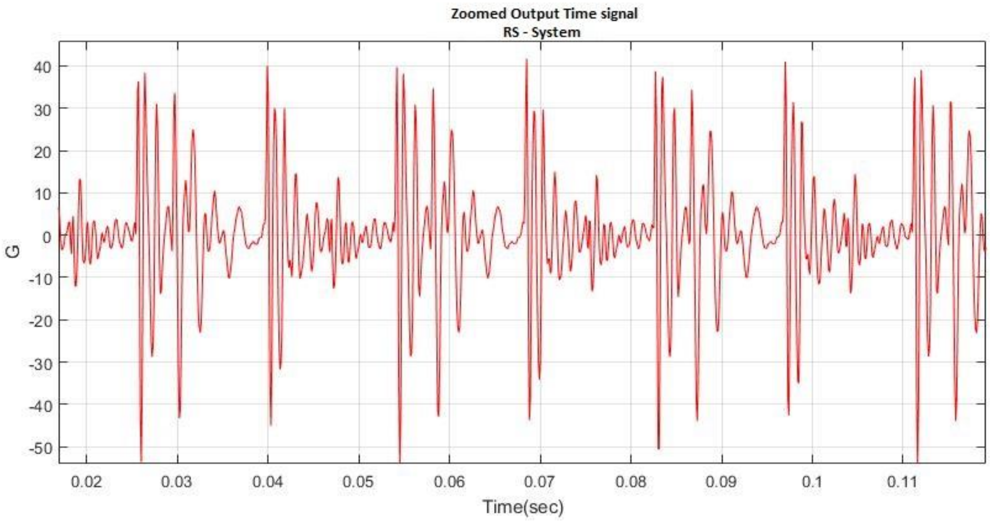

Figure 4-6. Acceleration time signal response by the RS system

A Study by Pei and Jianjun, [18], compares a conventional ED shaker with a new RS table, and concludes that the RS table outclasses the ED shaker at almost every aspect in frequency range, peak amplitude distribution, and stresses loading rate. Additional, the RS is also more effective at accumulating fatigue damage at a faster failure rate. As seen in Figure 4-7, there is a clear difference in the frequency range for the respective methods. These quick returning acceleration peaks will accumulate fatigue faster than the more continuous and controlled response from the ED system [16].

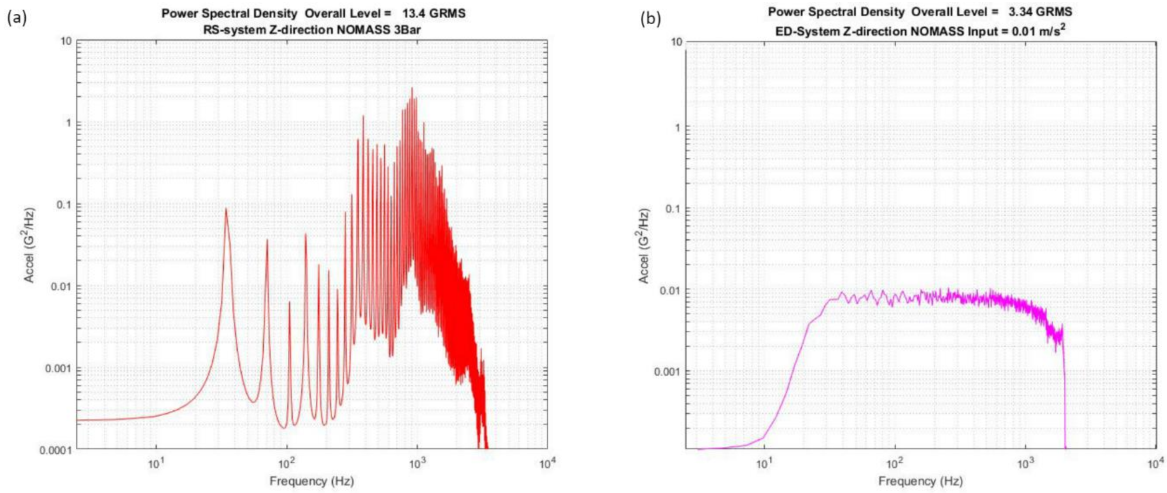

Figure 4-7. Acceleration PSD response for (a) RS and (b) ED system respectively

For the analysis of the PSD generated by the RS system, it is important to take into consideration that the RS system excites 3 axes simultaneously. This means during analysis of the outgoing signal that an evaluation of all axes is needed for a complete comparable conclusion [16].

The problem that arise with the RS system is the relative difference of G levels over the table. During response measurements, when testing multiple products at the same time, it is possible for the different products to have a Grms variation up to 100% relatively to each other. For better control a new type of software program is being developed by Thermotron® called Multi-Zone vibration control [20]. The software divides the table into one, two or four zones and controls each zone so that the relative Grms value is the same over the table [21, 22].

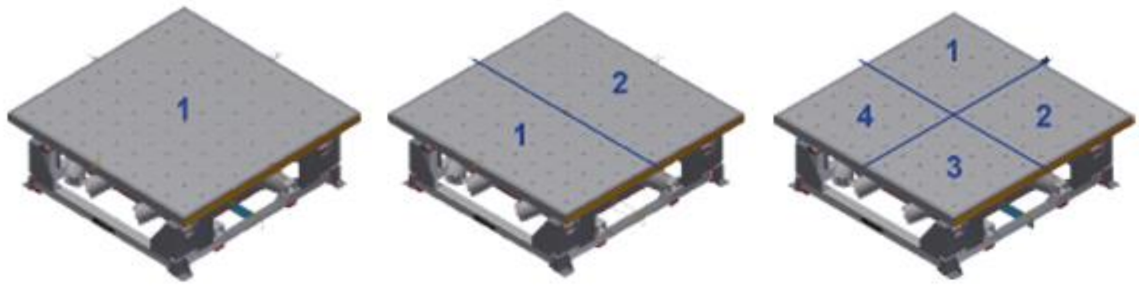

Figure 4-8. Multi-Zone vibration control patented by Thermotron® [23]

4.3 Acoustic vibration testing

Acoustic vibration analysis for different components and structures are most common during vibration testing of aerospace and spacecraft equipment due to stresses caused by pressure fluctuations during lift-off and flight. These type of vibrations can then be transferred throughout the structure into mounted components, which may result in failures if not accounted for during the design and analysis of the structure. The fundamental difference between random vibrations and acoustic induced vibrations is the manner of how the forces are applied. In random vibrations, the vibrational movement of the component is in direct correlation with the spring rate of the mounting and its mass distribution. In acoustic vibration, the input is often distributed over the whole components and its surface area, while its mount often has a relatively low spring constant and not connected to any structure.

Acoustic vibration testing can reach frequencies from 20 to 10 000 Hz, where the acoustic pressure environment generates dynamic loads on the testing specimen in two ways; either by direct impact on the surface of the specimen or by the acoustic pressure impact on the specimen's mount components, making the mount vibrate, transmitting the energy mechanically to the component.

To be able to analyse a specimen under acoustic pressure, it is put into an acoustic chamber. This enclosed space has thick walls with smooth interior surfaces so that high reverberations can occur, affecting the specimen. With this it is possible to verify the specimen's ability to withstand random vibrations generated from acoustic pressure. The specimen is excited by one or more electro-pneumatic driver with horns mounted in the walls of the reverberant chamber. An alternative configuration is to use electro-dynamic speakers arranged close and in a circle around the DUT in an acoustic chamber or an open space. The second configure mentioned is called a "direct acoustic test", where the majority of the soundwaves hit the DUT directly, not reflecting off another surface prior to that [24, 25].

Figure 4-9. Example of a speaker Cabinet Installation for aerospace testing [25]

It is typical to use this test method on lightweight structures with large surface areas, and with acoustic sensitive structures. The acoustic testing chamber in general is of a large volume where the specimen is places, often strung up by cords or put on stands in the middle of the chamber. Microphones are place for control measurements around in the chamber, as well as above and below the specimen.

The response of the structure is measured by sensors. The different types of sensors which can be used during acoustic vibration testing are, accelerometers, strain gages, microphones, surface microphones and flow sensors. The response feedback are then presented as power spectral densities, PSD for short, in a desired frequency range within a certain bandwidth.

For a successful acoustic test there need to following procedure and the test should include:

- A description of the specimen; structure type, volume, mass and the position of the specimen in the reverberant chamber.
- The test sequence, sound pressure levels, deviations and durations

- Map out sensor placement and the number of sensors in the chamber
- Details of sensor measurement presentation, resolution of time and frequency domain
- Success criteria

Before conducting the acoustic test, the chamber will be needed to be run without the specimen to establish a reference and correct control equipment settings. For each run the deviations and duration of the sound pressure level should be defined and are dependent on the specimen [2, pp. 41, 233].

Determining the correct setup for acoustic vibration testing comes down to the area/mass ratio of the component. If the component has a high ratio, which means large area relatively to its low mass, then it can be suspended in the air by bungie cords inside the reverberant acoustic chamber, directly hitting it with the sound pressure waves [26].

It is more problematic if this area/mass ratio is low, which then the input vibration energy should come through the mounting structure of the component. This is used in aerospace because the acoustic vibrations may excite a spacecraft structure, where they are then transferred via the structure surface to the smaller mounted components resulting in fatigue damage. The "break even" area/mass ratio is where the acoustic and the random vibration test induce an equal stochastic acceleration response. To determine when this happens a single degree of freedom system can be solved for analytically [27].

Figure 4-10. Acoustic test with a plate strung up with a component attached [26]

4.3.1 Acoustic effects on electronic components with mathematical formulas

An acoustic noise generated by any source may affect its surrounding. Everything from a siren to a jet engine, all creates vibrations which travels through a medium through radiation or conduction depending if it is through a gas, a liquid or a solid medium. As stated before, large thin structures acoustic vibrations can lead to a rapid failure of the component.

The acoustic noise generated is measured in decibels (dB), and correlates to the ratio of the mean square sound pressure to a reference mean square pressure. This reference is the threshold of hearing [2, pp. 233–245].

How to convert sound pressure to number of decibels:

$$\boldsymbol{Number\ of\ decibels\ (dB) = 20log_{10}\frac{P}{P_{ref}}} \quad (10)$$

The sound pressure spectral density is given by:

$$\boldsymbol{P_s} = \frac{\boldsymbol{P}^2}{\Delta \boldsymbol{f}} \tag{11}$$

Sound pressure response to acoustic noise excitation:

$$\boldsymbol{P_{RMS}} = \sqrt{\frac{\pi}{2}\boldsymbol{P_s f_n Q}} \tag{12}$$

Or

$$\boldsymbol{G_{out}} = \sqrt{\frac{\pi}{2}\boldsymbol{A_s f_n Q}} \tag{13}$$

The component surface area is:

$$\boldsymbol{A_s} = \frac{\boldsymbol{G_s^2}}{\Delta \boldsymbol{f}} \tag{14}$$

and

$$\boldsymbol{G_s} = \frac{\boldsymbol{P}}{\boldsymbol{w}} = \frac{\boldsymbol{P}}{\boldsymbol{\rho h}} \tag{15}$$

<u>Calculating the break-even area (Acoustic testing)</u>

Break-even area is given by

$$\boldsymbol{A_{be}} = \frac{\boldsymbol{G_{RMS} m}}{\boldsymbol{P_{RMS}}} \tag{16}$$

Break-even area/mass ratio equals

$$\frac{\boldsymbol{area}}{\boldsymbol{mass}}\boldsymbol{ratio} = \frac{\boldsymbol{A_{be}}}{\boldsymbol{m}} \tag{17}$$

More detail on how to handle and calculate the acoustic effects on electronics and PCBs are provided by Steinberg [2].

4.4 Shock vibration test

Vibrations caused by shock are a transient mechanical loading which last for a short duration. They also have a high frequency content and high acceleration amplitudes with rapid rising times. In other words, a significant increase in stress, velocity, acceleration or displacement occurs within the system. When a complex structure is exposed to shock, it is often excited over its resonant frequencies which can result in different types of failures; through high stresses, high acceleration levels, high displacements and/or electrical malfunctions. Depending on the material and geometries of the structure, it may react differently to shock. The structural response to shock depends on its impact sensitivity, notch sensitivity, and brittleness. For structures with brittle character, shock may cause problems in regard to the dynamic stresses with respect to the anticipated environments. For most structures, the ductility of the structural elements are above 5%, which means that the limiting factor is then related to the yield strength of the material.

There are a lot of different methods to specify the impact from shock vibrations. The most popular methods are pulse shock, velocity shock and shock response spectrum.

Pulse shock is a simplification of a shock impact in reality and deals with accelerations or displacements through simple waves in forms of squares, half sine, and various triangular waves. This makes pulse shock easy to work with, but is not good at representation for realistic shock vibrations. However, it can come in handy in revealing weak spots of different structures.

Velocity shock handles when a system goes through a sudden change in velocity, which can be an falling object which hits the ground, resulting its velocity go to zero in an instant. These types of impacts it is often called a drop shock. Another example may be to slam some type of object on to the fixture of system of the analysis at high velocities and can, for example represent an explosion [2, pp. 41, 248–253].

In a drop test, the DUT comes to a sudden stop from striking the ground and its velocity goes to zero generating high accelerations and stresses on the DUT.

In an impact test, also called a pendular test, the DUT is fixed on to a plate and a hammer is lifted up in a circular motion to hit a base plate. This in turn induce vibrations in the baseplate which are transferred onto the DUT [28].

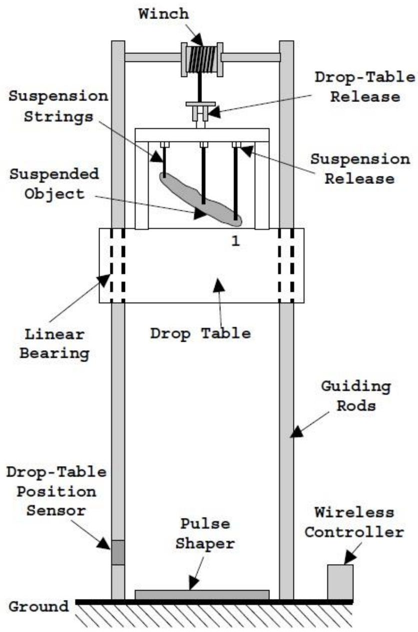

Figure 4-11. Illustration of how to setup a reliable drop/impact test [19]

Electrodynamic shaker and Repeatable shock/Pneumatic shock testing can be configured for vibration shock tests to replicate a reliable impact test and achieve high level shock pulses. The Electrodynamic shaker has some physical limitations for shock testing due to its limited armature displacement and shaker/amplifier power rating [29].

4.5 HALT and HASS

HALT (Highly Accelerated Life Test) and HASS (Highly Accelerated Stress Screening) are two processes which are used for accelerated testing to efficiently expose the weak points during different phases of production. In the transition between the development phase and manufacturing, there are often problems that arise concerning the product. This can be that the design safety margin is either too small, which means that the functionality during manufacturing may be compromised, or too high, which is not cost effective, leading to unnecessary expenses. This is where HALT comes in handy. With

HALT it means that the product can easily be re-designed during the design phase and therefore avoid the need for changes in a later stage, which may be more difficult to solve. HASS is used during the manufacturing stage to expose the weak point which may have occurred during this process and to assure that the improvements done after HALT have the desired results of the product [15].

The HALT procedure is where a product or component is subjected to continuously increased stresses through thermal dwells, temperature cycling, vibrations and combined environment. During HALT, the operating conditions for the chosen product are not taken into account during this procedure, however it will be under operation while the continuous stresses are applied. This process alone is not enough to improve the reliability if the product, but it will hopefully be able to determine the root cause for failure and then preventable by other means during design [30].

HASS procedure is during the beginning of manufacturing to look for systematic or single failures. During HASS the highest stress is used for the highest possible fatigue accumulation or lifetime degradation. The stress levels that are present during HASS are not as high as during HALT but still well beyond the qualification level. There are different types of procedures depending what kind of fatigue environment it is supposed to reflect [31, 32].

4.6 Piezoelectric shakers

Piezoelectric shakers use the inverse piezo effect to generate vibrations. Because of the use of the piezo electrical effect there are some advantages compared to an electrodynamic shaker. Comparing the different testing methods it is clear that the vibration amplitude relative achievable forces are completely different. While Electrodynamic shakers have a large amplitude with comparatively small forces, the piezoelectric shaker has small amplitudes with comparatively high forces as illustrated in Figure 4-12.

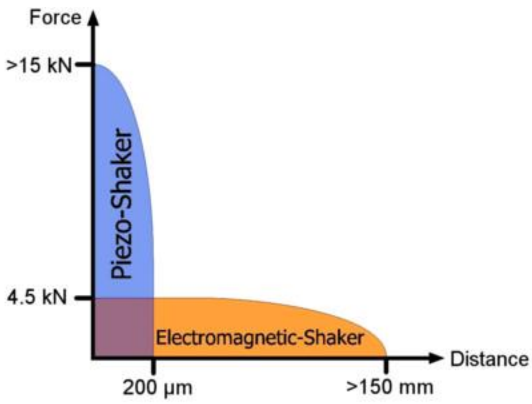

Figure 4-12. Comparison between Electromagnetic/dynamic and piezoelectric shakers [33]

It is possible, even at these high forces to test devices up to 2000 N at high frequencies. However it happens that the amplitude will decline quadratic with an increased frequency. Electrodynamic shakers are suited for tests at the lower frequencies while the piezoelectric shaker are designed for applications in the higher frequency region, particularly over 100kHz, but have a disadvantages at lower frequencies and non-designated high amplitudes [33]. But in comparison, the piezoelectric shaker's ability to achieve really high accelerations is much better compared to an ED system. Small piezoelectric shakers can reach up to 3000 m/s^2, but is then highly specialised for a specific area of testing [34].

4.7 Summary of various testing equipment

With research on different vibration testing systems and their methods, and looking at how practical it would be to implement these systems, a conclusion will be stated in this section.

Getting a closer look at the Electrodynamic shaker system shows its great versatility in the field of vibration testing and it is understandable why it is one of the most used types of equipment within the field of vibration reliability testing. Because of this the electrodynamic shaker system is often used as a template when comparing other vibration testing methods.

The electrodynamic shaker is versatile in that of vibration testing because of its ability to simulate most of vibration environment requirements concerning industries handling vibration testing, especially compared to most other types of vibration testing equipment is the level of control which is achievable with an ED system higher. Of course the ED system has its limitations and it is expensive to have the best on the market, the one that push the technology close to its physical limitations. Buying a more powerful machine mostly comes at the cost and reduction of some other parameter, meaning for example, a reduction in armature size for higher energy frequency testing and/or loss in peak acceleration as well. It is possible to upgrade and use a double shaker system and use an extensive cooling system, but this solution may not be economically viable.

When analysing for the acoustic vibration testing has great potential for customizability on vibration testing of electrical components. It is used frequently within the aerospace industry, and is effective at screening large, but light objects. However, more research is needed to conclude if acoustic vibration testing is applicable for SCANIA and/or the automotive industry at all.

While the electrodynamic shaker may be the most common equipment within industries for vibration testing for its versatility, other methods are better within some areas of testing. The RS system, often in combination with a HALT/HASS system with a temperature chamber, may appear as a standalone for vibration fatigue testing related purposes. In particular, the repeatable shaker system is more efficient at accumulating fatigue compared to ED systems.

Shock vibration test it is hard to justify for its inability to exercise large amounts of iterations in a short time. When performing an accelerated life length test, the equipment should be able to represent a specific environment to reach a specific stress level at a set number of iterations, and with the current shock equipment is it hard to fill that criteria or replace the ED system.

Looking at the piezo-shaker and it is hard to find an application area in the automotive industry for this type of equipment. It has potential but falls short when it comes to peak-to-peak displacement and having its most effective frequency range around 100 kHz, which for our purpose is to high.

5 Electrodynamic and Repeatable shaker equipment response testing

Understanding how the ED system respond when tested at its limits will give the insight at were the max operational frequency really is at an arbitrary acceleration level. Knowing the limits of the ED system will set the level where the RS system needs to be implemented to complement and cover the frequency spectrum up to around 5000 Hz.

In this chapter, the response of the two different system are presented and their results are compared.

5.1 Electrodynamic shaker setup

The electrodynamic shakers that are in use today at Scania CV AB are LDS V8 and V875 vibration test systems. These systems are able to cover the criteria specified by SCANIA's vibration testing standard.

As stated before, the ED system have its resonance frequency above its recommended operational frequency, this operation limit is often set as a restriction in the software. When the resonance frequency is reached the machine will not give a reliable response for testing and in extreme cases overextend its peak-to-peak movement of the armature resulting in breaking of the machine. The ED system only operates and excites along one axis, perpendicular to the armature. If the response is out of the ordinary along the other axes, parallel to the armature, then it is hard to justify a standard test as reliable.

For the practical test, the V875 Vibration test system was used with a 440 mm armature size, SPA40K amplifier, see Table 5-1.

Table 5-1. Specification on Electrodynamic shaker V875 with amplifier SPA40K.

Armature Resonance (f_n)	**2650 Hz**
Usable frequency range	**3000 Hz**
Random force, RMS	**35.6 kN**
Acceleration, random, RMS	**735.5 m/s^2**
Displacement peak-peak	**50.8 mm**

With known specifications on the recommended operational range for the machine, the test specifications was set at recommended max operational range and the other at the max operational range of the control system and software, see Table 5-2.

Table 5-2. Equipment setup parameters for electrodynamic shaker

Equipment model	**LDS V875**
Input acceleration for both tests (constant)	**0.01 G^2/Hz**
1st test (recommended frequency range shaker)	**20 – 2000 Hz**
2nd test (limit of controller and software)	**20 – 4000 Hz**

5.2 Electrodynamic shaker system results

Figure 5-1, Figure 5-2 and Figure 5-3 shows the PSD processed from the random vibration time signal response in z, x and y direction respectively, where the blue curve is for 2000 Hz and red curve is 4000 Hz.

When testing in the z-direction, Figure 5-1, at an input acceleration level around 0.01 G^2/Hz, an extreme peak can be seen around 4000 Hz. This peak is an indication that the response from the armature is higher than expected, meaning that the test is not reliable as the output should represent the output.

Figure 5-2 and Figure 5-3, further indicate that a 4000 Hz test on the ED system is unreliable, also there are high peak responses around 2000 Hz as well. This results in that both tests over-work and this might then result in unaccounted for stresses.

While analysing the output from the ED system and its corresponding PSD it shows extreme acceleration peaks in the x, y and z direction around 4000 Hz. This data also show that it is not optimal for this specific machine to operate frequencies at 2000 Hz either, even if the ED system is specified to be able to handle frequencies up to 2000 Hz.

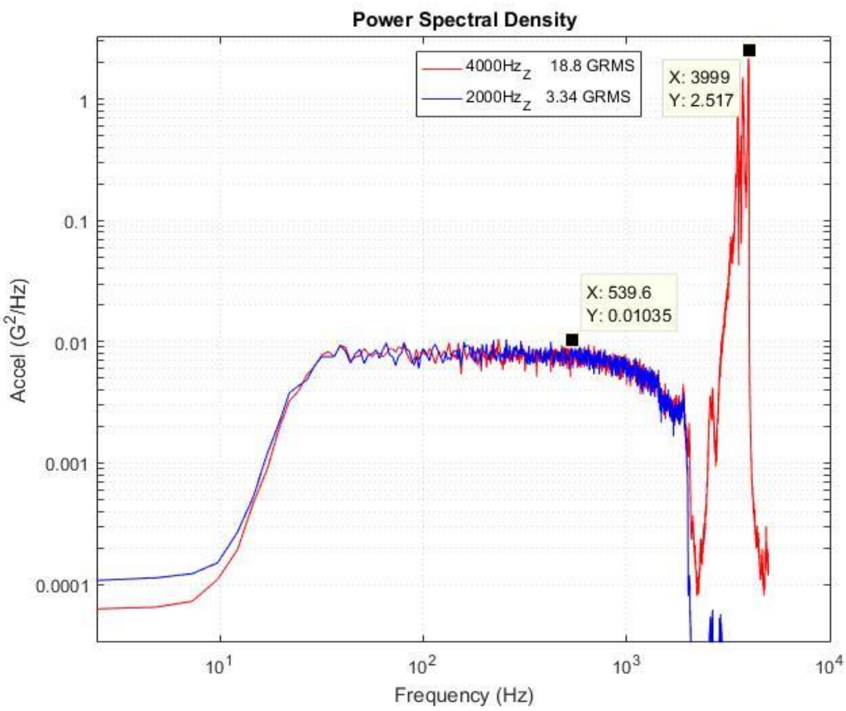

Figure 5-1. Response from ED system Z-direction, input up to 2000Hz (Blue), input up to 4000Hz (Red) with constant acceleration

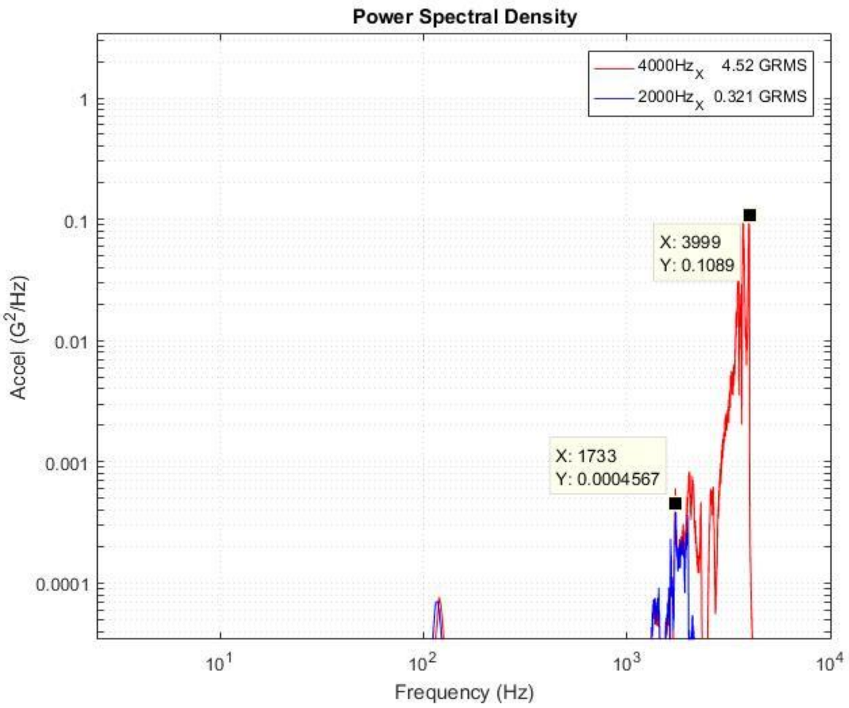

Figure 5-2. Response from ED system X-direction

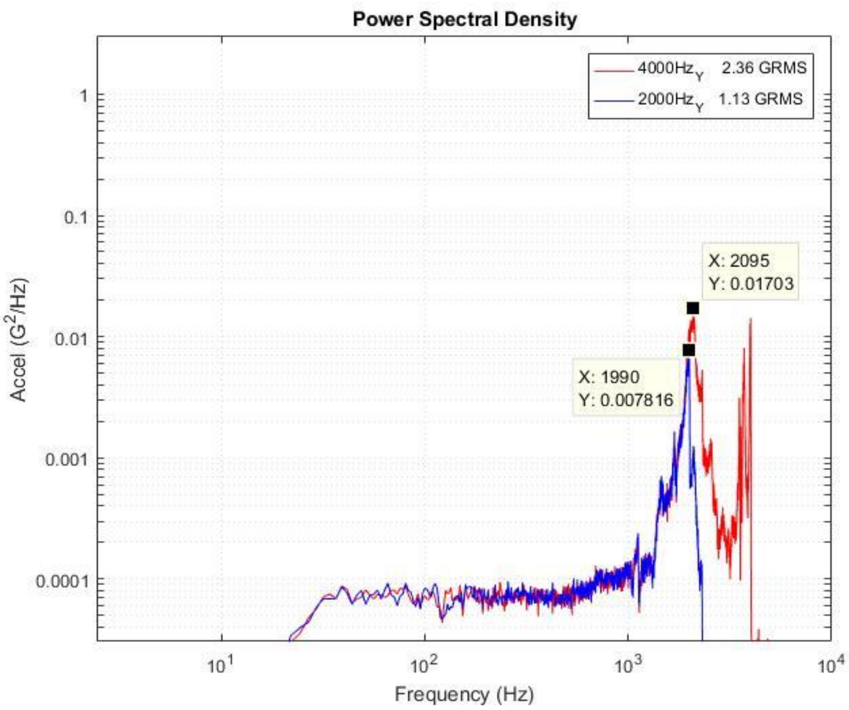

Figure 5-3. Response from ED system Y-direction

5.3 Repeatable shaker setup

The RS system at my disposal is a shaker table, which was a part of a HALT/HASS system, model FS750-60 VS with environmental chamber type FS340-60VS. The environmental part of the system was separated from the RS system, with that the corresponding control system as well. The core parts of the RS system were still functional, the only thing missing was a control system.

It was decided to construct a simple control system for the pneumatics actuators. The function of the controller was simply to open the pilot valves to the air flow to the actuators of the system. When air flows through it actuate the hammers, repeatedly hitting the table, in turn making the table vibrate. The fundamentals of a RS system can be found in section 4.2.

For the pneumatic system of the shaker table, a power supply 24 V, max 2A was attached and connected to the pilot valves controlling the airflow to the regulators. Every pilot valve were individually controlled by their own power switch, meaning that it was possible to control the activation of the hammers individually. The pressure was kept at a constant pressure for each individual test. This system setup was not a closed loop, leading to no real time control of the system.

In the first test the table was setup with four accelerometers in a square formation around the centre of the table top, as in Figure 5-5(b), measuring its response. Comparing different setups and configurations can show the characteristics of the table compared to different variables. For the first setup the air pressure was varied and its response measured in the x, y and z direction, Figure 5-5(a). For the second setup the test were performed at its max operational air pressure and adding weights, increasing the system's stiffness, Figure 5-5 (b, c, d).

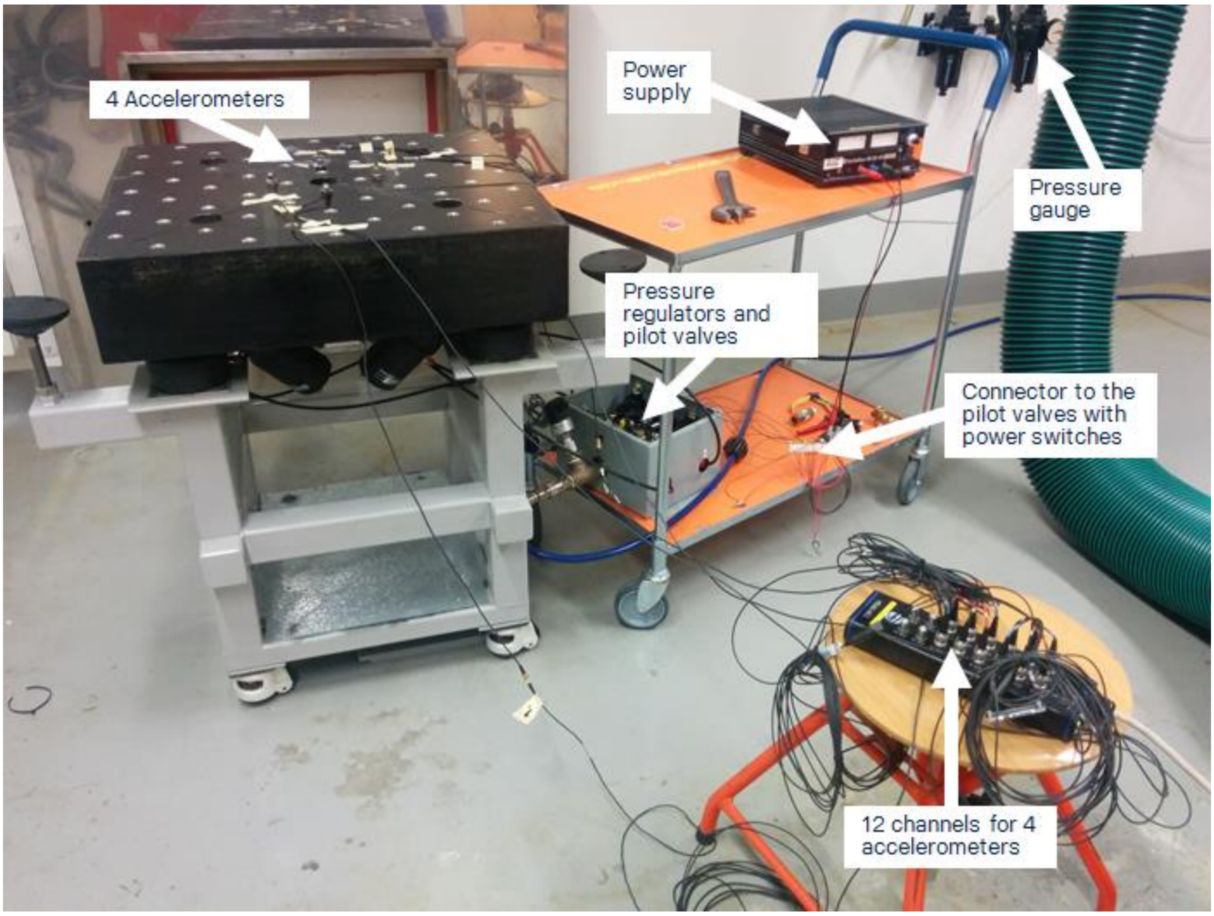

Figure 5-4. RS system setup with power supply and the pneumatic system

Figure 5-5. RS system for measuring the difference in response with different amount of mass (aluminium plates) (a) 1st test Empty table, (b) 2nd response test illustrating the position of the 4 accelerometers and their position, (c) 3rd response test, (d) 4th response test.

5.4 Repeatable shaker system response results

It is concluded from the analysis that the response from the RS system and the characteristics are different from the ED system comparing their corresponding PSD curves. In this section are the results from the practical experiments presented.

Analysing the different responses from the RS system under certain conditions will give an understanding of how it can be controlled and influences by external sources. For the first setup the table top was empty, as seen in Figure 5-5 (a), measuring the response. This setup was used, mostly as a reference for the following tests and compare the characteristics of the different responses. According to the literature [16, 35], it is stated that the RS system is not able to be controlled in real-time, that the time signal has these high acceleration burst and show "gaps" in the correlating PSD.

For the convenience of comparing data, only the response in the z – direction will be presented in this chapter, and some other results from the other directions can be found in chapter 7, appendix.

Table 5-3. Available setup parameters range

Air pressure	**3 – 7 Bar**
Number of active actuators	**2 – 5 pcs**
Number of aluminium plates	**0 – 5 pcs**

5.4.1 RS system experiment Empty table 3 - 7 bar result

The output signal from the RS system shown in Figure 5-5(a), show the response at 3 bar and it resulted in an irregular response pattern, Figure 5-6. It shows high irregular acceleration peaks at small instances in time. Zooming in on the time signal, Figure 5-8, makes it is easier to distinguish the characteristic pulses and peaks of the acceleration, then point out the fundamental different between the ED system response and the RS system response. If we still have the same setup as the as in Figure 5-5(a), but increase the system pressure provided to the actuators to 7 bar, then we get the response shown in Figure 5-7 and a more detailed look in Figure 5-9.

When analysing the corresponding PSD, the pulsating character of the RS system can be distinguished with its high peaks and low valleys with more or less pulsating through the entirety of the PSD as shown in Figure 5-10 and Figure 5-11. Comparing the response from the 3 bar test and the 7 bar there are clear differences. Not only are the acceleration levels higher with the increase of air pressure, but the signal is more predictable in its continuous pattern over the whole run time history. The time history from the 3 bar test is an indication that it is not reliable for repeated tests because of the unpredictable high acceleration peaks at points in the time spectrum while 7 bar response is what is expected from the equipment. In Figure 5-12 the PSD curves are super imposed for easier comparison and shows the PSD curves follow each other, only that the 7 bar test has a continuous higher amplitude over the frequency interval.

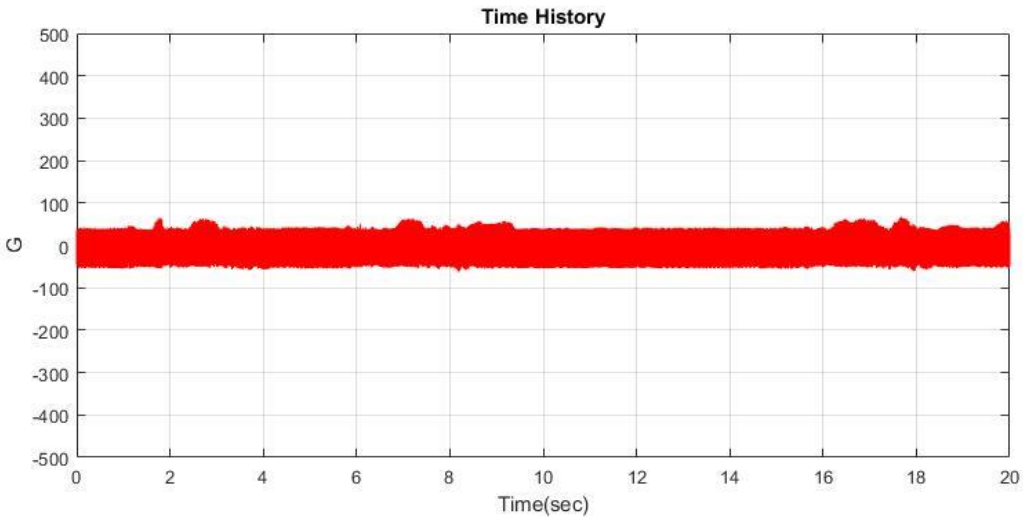

Figure 5-6. Output time history for 3 bar, empty table

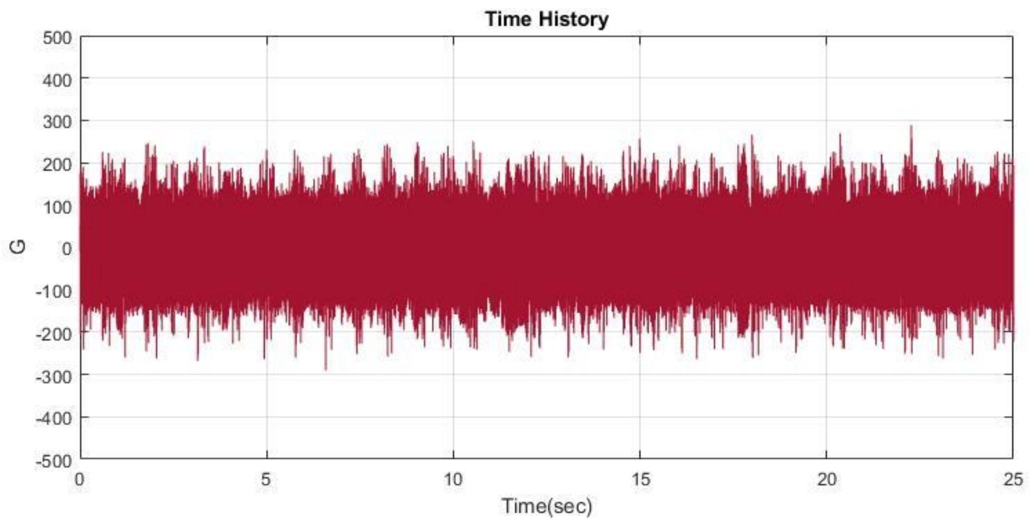

Figure 5-7. Output time history for 7 bar, empty table

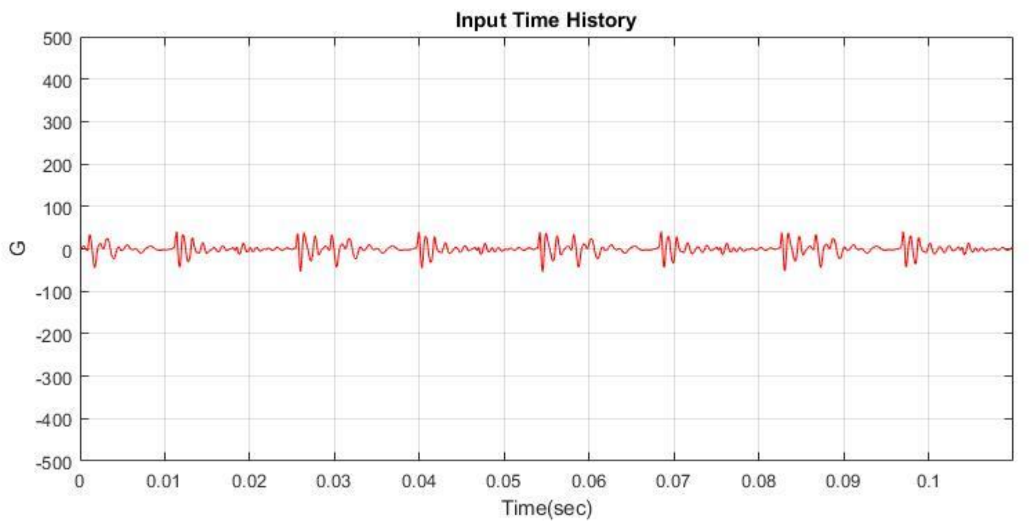

Figure 5-8. Zoomed output time history for 3 bar, empty table

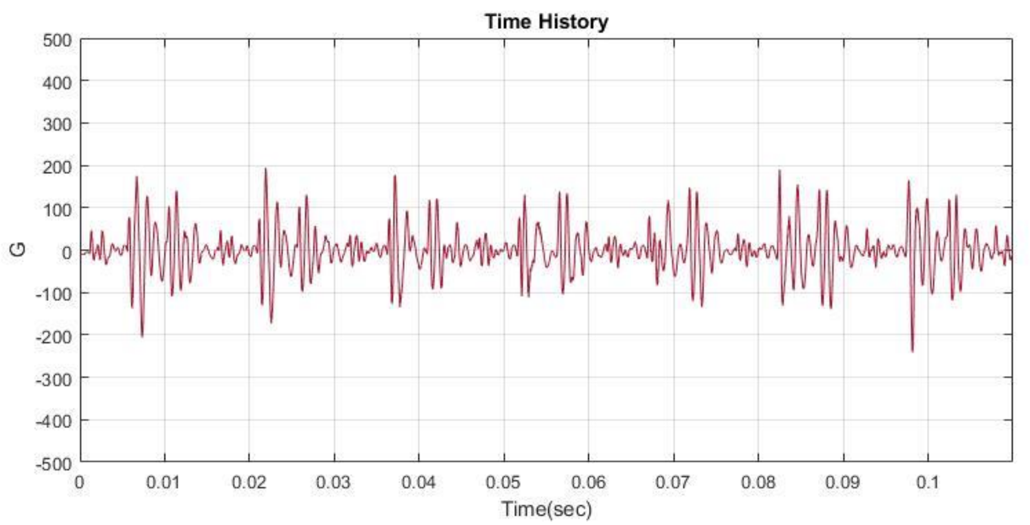

Figure 5-9. Zoomed output time history for 7 bar, empty table

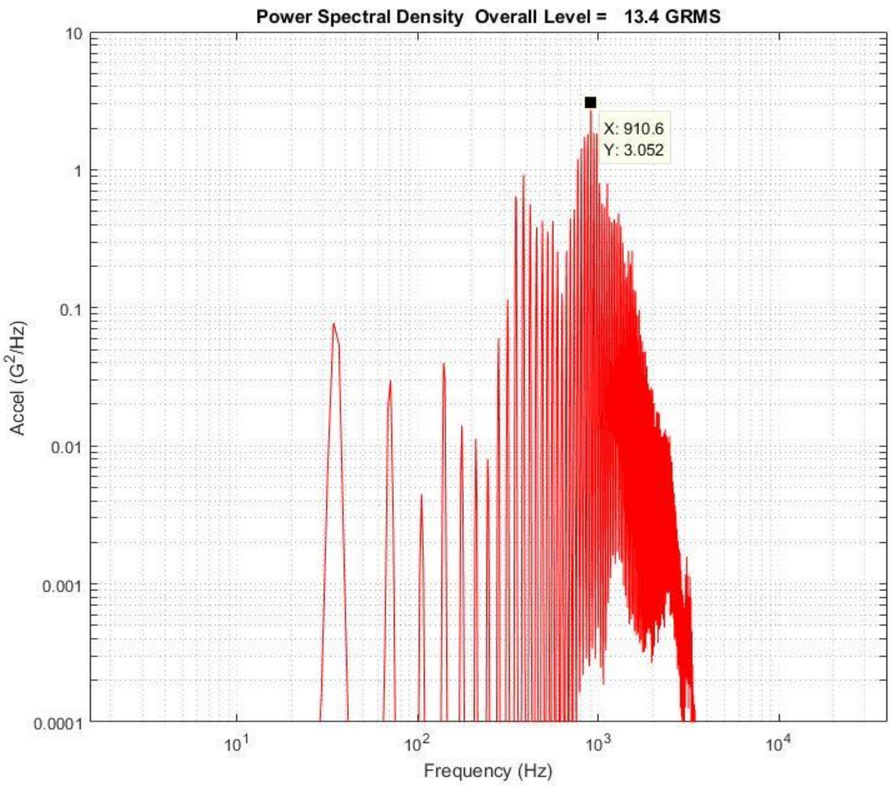

Figure 5-10. PSD output from empty table Z-direction with a 3 bar air pressure input empty table

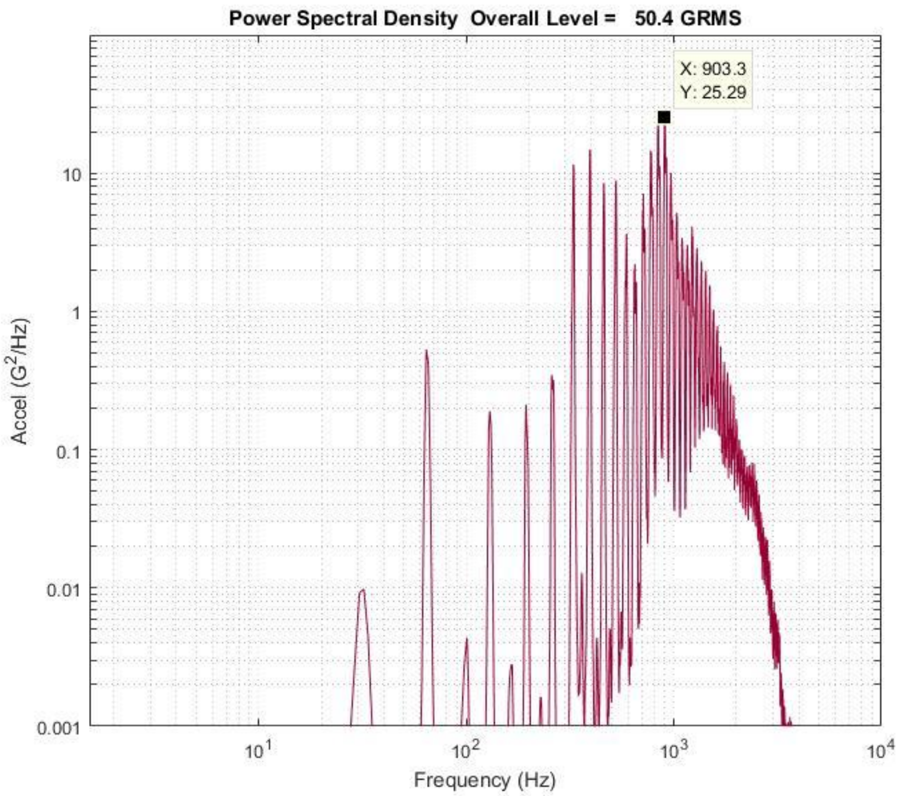

Figure 5-11. PSD output from empty table Z-direction with a 7 bar air pressure input empty table

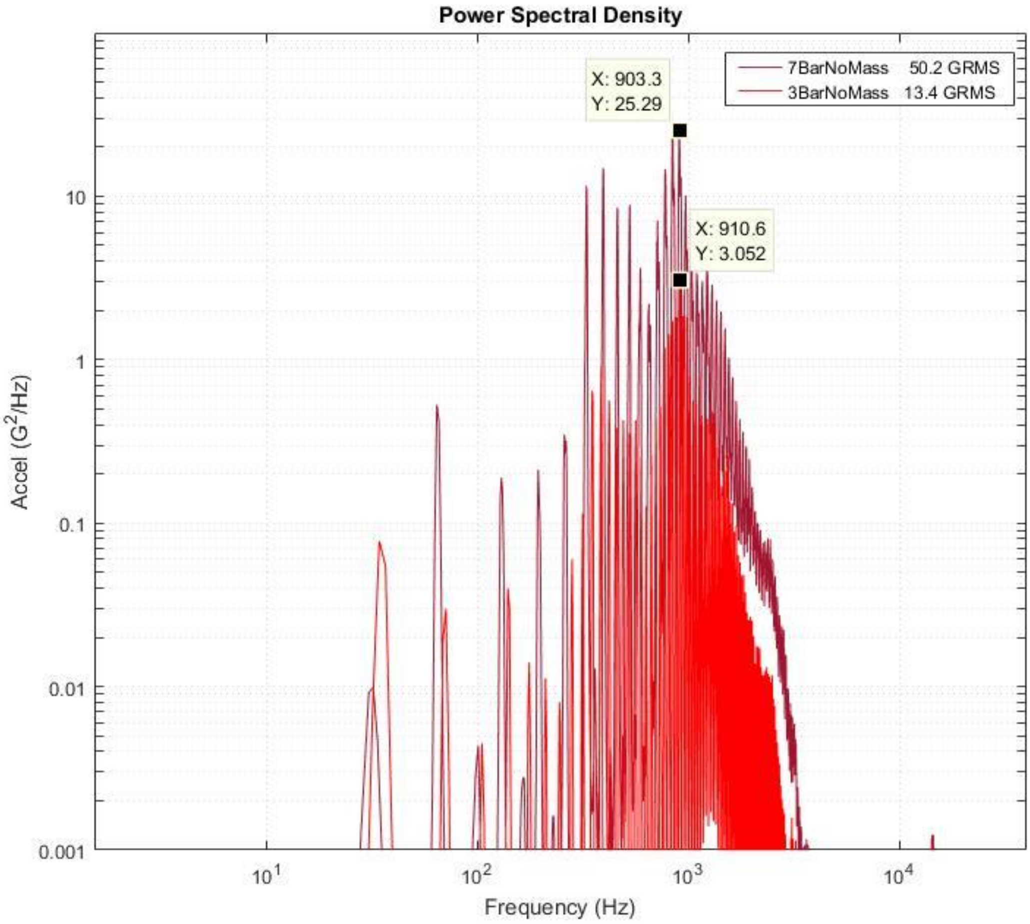

Figure 5-12 super imposed PSD curves from empty table Z-direction with 3 and 7 Bar air pressure input empty table

5.4.2 RS system experiment 1 and 5 plate(s) at 7 bar result

In the next setups, Figure 5-5(b, d), mass is added and fixed on top of the RS table at the constant pressure of 7 bar. The response measured by the accelerometers showed some differences when it comes to the time signals and its corresponding PSD. The first distinctive difference when analysing the time signal in Figure 5-13, which relates to the setup in Figure 5-5(b) with the one added aluminium plate, is the higher acceleration and more protruding acceleration peaks compared to the response in, Figure 5-14, from the setup in Figure 5-5(d) with five added aluminium plates. This means that the addition of more mass leads to a damping effect, lowering the extreme acceleration peaks. The different time signal responses are presented more in detail in Figure 5-15 and Figure 5-16.

By processing the time signals, analysing their respective PSD curves and the damping effect becomes clearer. The PSD curve from the setup with one aluminium plate, Figure 5-17, has an overall Grms value about 4 times higher than the PSD from the setup with 5 aluminium plates, Figure 5-18. But the energy that is apparent around 10 kHz is there in both of the PSDs, Figure 5-19.

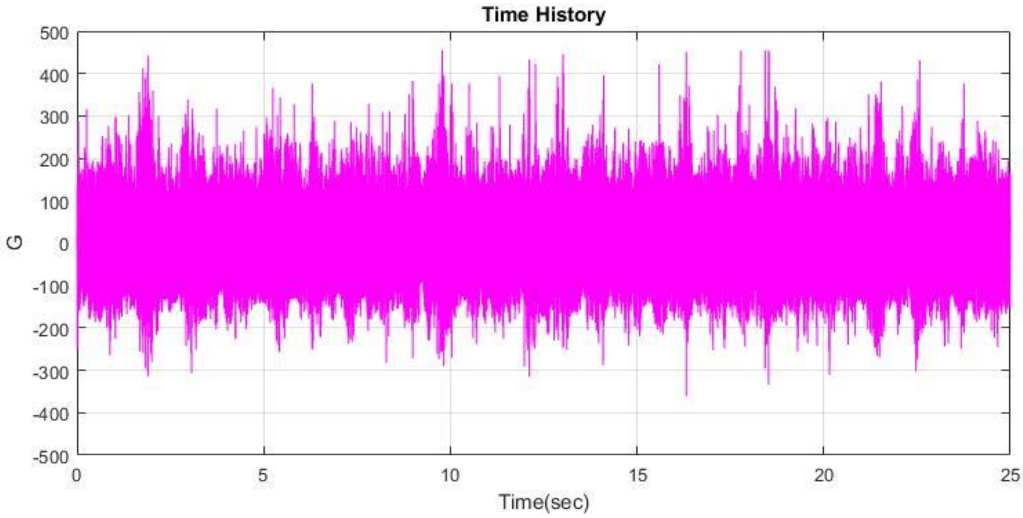

Figure 5-13. Time output signal from the 7 bar 1 plate test

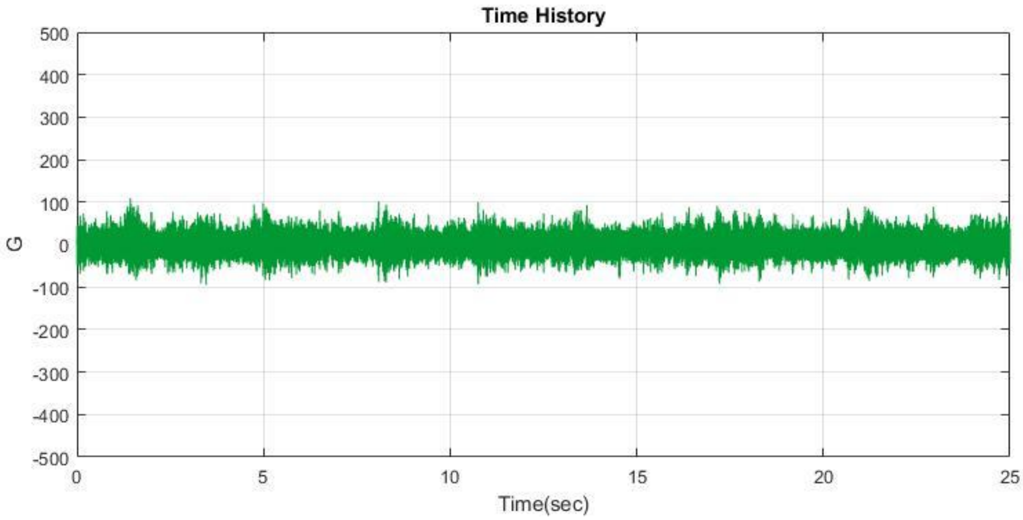

Figure 5-14. Time output signal from the 7 bar 5 plates test

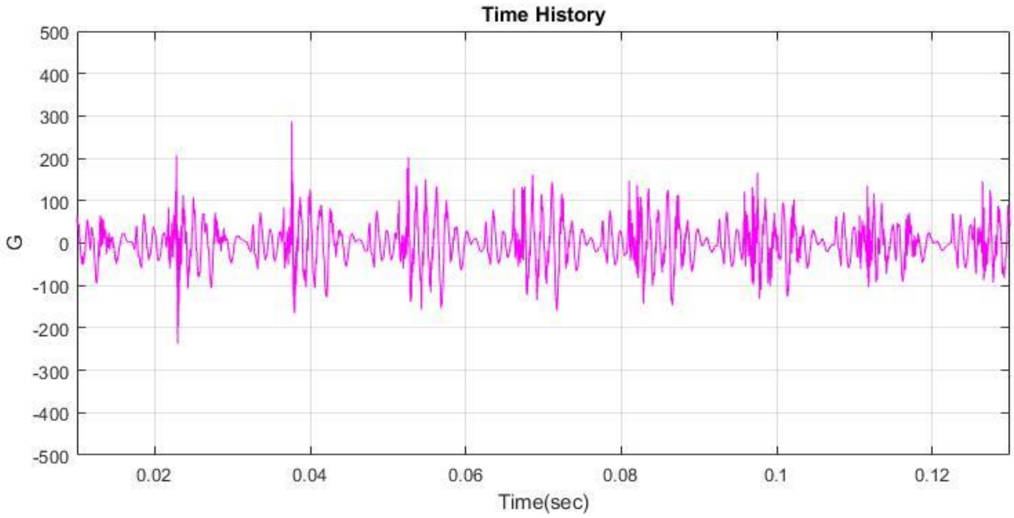

Figure 5-15 Zoomed time response signal of 7 bar 1 plate test

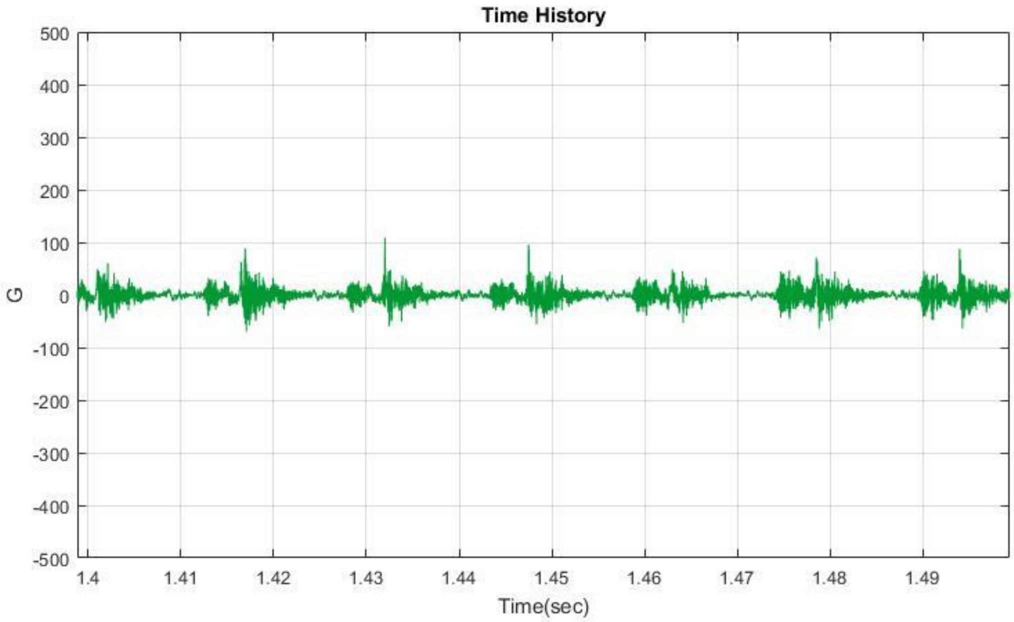

Figure 5-16 Zoomed time response signal of 7 bar 5 plates test

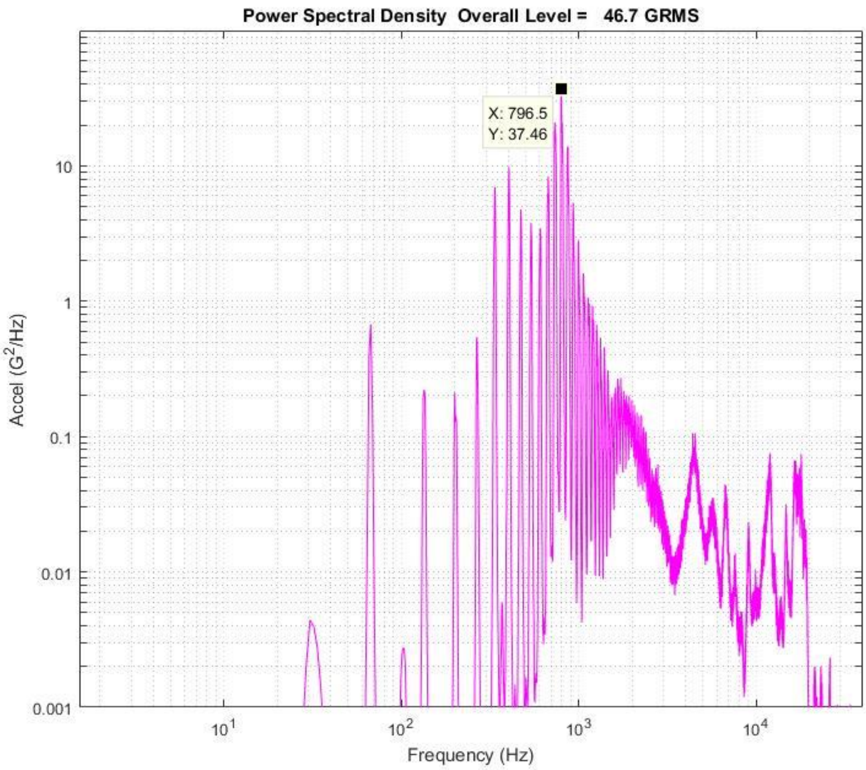

Figure 5-17 PSD curve of 7 bar 1 plate

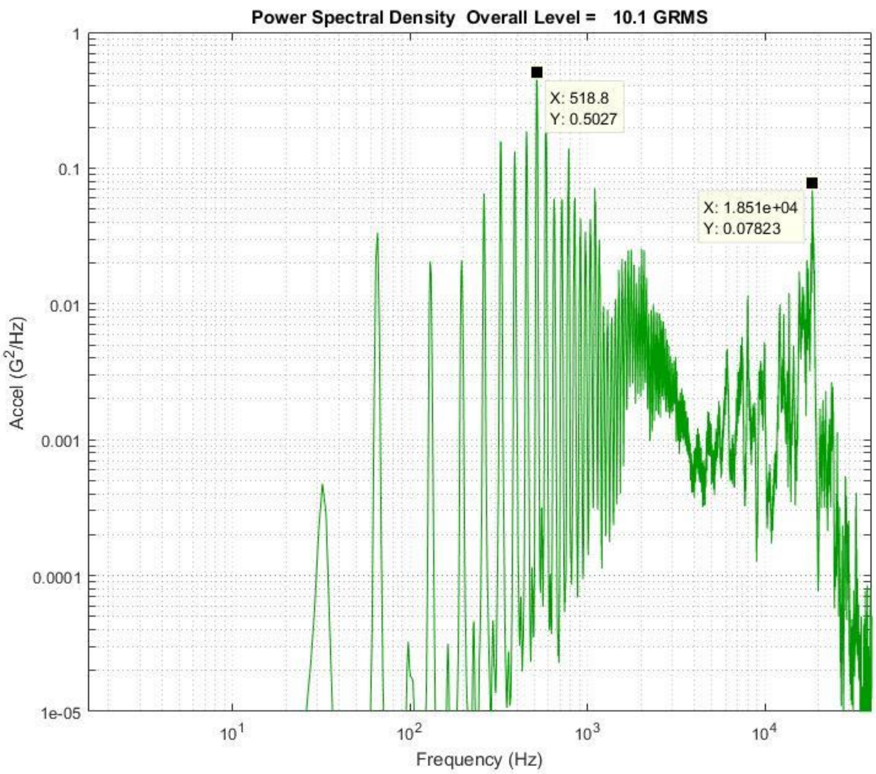

Figure 5-18 PSD curve of 7 bar 5 plates

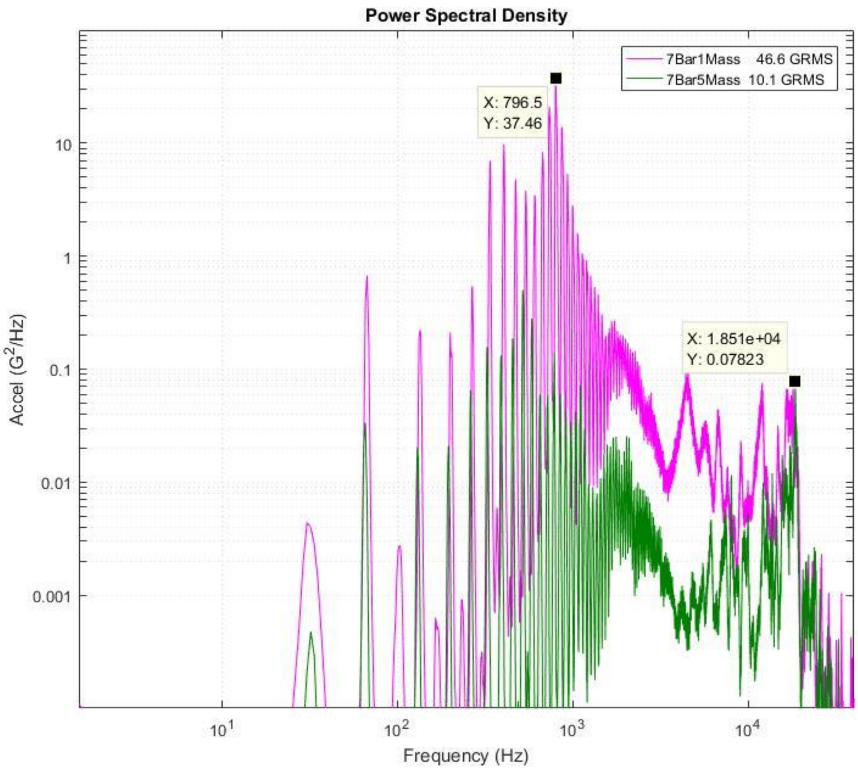

Figure 5-19. Superimposed PSDs curves 1 plate 7 bar (Purple), 5 plates 7Bar (Dark green)

6 Conclusion and prospect to future work

In this thesis different equipment for accelerated vibration testing were investigated due to the limitations of the current standard equipment used by SCANIA CV AB. It was intended to analyse if one type of equipment could replace or complement and if, would it be versatile enough to cover all areas without sacrificing performance or accuracy and still reaching higher energy frequencies. All the equipment that are investigated in this paper have potential and in some way covering our specifications more or less. The Electrodynamic shaker was able to achieve the criteria at the cost of a smaller armature size and/or lower max peak-to-peak acceleration. On the plus side is the well-developed control system that have the ability to control the acceleration profile in real time, also being able to perform sinusoidal, random and shock test. Because of the versatility, makes it difficult to replace this equipment with another equipment type. The second promising candidate were the repeatable shaker system, with its 6-DOF axial vibration. At first glance, analysing the results from the RS system showed that it had potential to cover our criteria for vibration testing, but the way it induces vibrations and their characteristics is not completely comparable with the results from the ED system. Therefore, the RS system cannot replace the ED system in its current developed state and cover all of the aspects. The main idea of RS system is ”test to failure”, not as the ED system which is “test to pass”. The general higher G peaks are the cause of this and the unavailability of real-time control. At this time the only available level of control is to set a Grms value with a closed feedback loop for the system, for it to work against, and through analysing the different responses in relation to system air pressure and the different amount of mass added, only a general relation to the these variables can be concluded. Analysing the literature gathered for the acoustic vibration testing, it concludes that it is a promising candidate which do not have the same restrictions that of the ED system. However, the acoustic vibration testing equipment has some drawbacks as well. The first to take into account is, to be efficient of transferring sound vibrations through the air it onto a DUT is only recommended for components or structures that have a large surface area compared to its mass. The second criteria that has to be filled for a versatile way of doing acoustic vibration testing is a large amount of space dedicated to the equipment. In most of the cases concerning acoustic vibration testing, it is clear that the setup takes up a great deal of space and this means that the investment cost is high. Other types of equipment like the typical shock or piezoelectric equipment does not fill the necessary requirement and cannot replace the ED system. In short, the shock system reaches high level of energy but is not efficiently repeatable enough, and the piezoelectric shaker is able to attain high energy levels at high frequencies but its peak-peak armature distance is minuscule and not enough for our applications.

In this thesis the RS system was analysed to see if it had some potential in replacing the ED system. But for that to happen, then the RS system would need a higher level of control, in its performance of inducing vibration i.e. be able to set a specific acceleration level like the ED system. When the practical response test on the RS were done the output energy could only be controlled to a certain level, and it went to establish a pattern which emerged relatively to the increase in air pressure and added mass. There is a correlation between the addition of more mass and the decrease in Grms, damping the most extreme peaks and making them not stand out as much from the other energy peaks over the frequency span, this correlation is seen in Figure 5-19. But as stated before, it would have been better with a manufactured control system, making the precision of control a little higher. But it does not change the fundamentals of the RS system and it will not at this point be able to replace the ED system and its high level of control during vibration testing.

For the future work within this area would be to look deeper into the acoustic vibration testing, trying to test if there are some ways to customize it so that it better fits for vibration testing standards within the automotive industry and SCANIA. It really has potential for covering these higher energy frequencies, with great potential for high precision during testing, because of the likeness between the ED system and a loudspeaker. This might be achieved by understanding how soundwaves behaves, its transmissibility and at what level these soundwaves transfer their energy into the DUT converting into mechanical vibrations. Hopefully this can be done with more research, using theoretical work in combination with practical experiments but concentrated on acoustic vibration testing.

7 References

[1] P. K. Aggarwal, “Dynamic (Vibration) Testing: Design Certification of Aerospace System,” *Encycl. Aerosp. Eng.*, 2010.
[2] D. S. Steinberg, Vibration Analysis For Electronic Equipment, 3rd ed.
[3] A. Brandt, “Ljud- och vibrationsanalys I”, Saven EduTech AB, 2000.
[4] G. R. Henderon “Dynamic Characteristics of Repetitive Shock Machines” GHI Systems, inc, 1993.
[5] “Power spectral density function.” [Online]. Available: http://www.cygres.com/OcnPageE/Glosry/SpecE.html. [Accessed: 23-Feb-2017].
[6] “What is a Power Spectral Density (PSD)?,” *Siemens PLM Community*, 17-Aug-2016. [Online]. Available: https://community.plm.automation.siemens.com/t5/Testing-Knowledge-Base/What-is-a-Power-Spectral-Density-PSD/ta-p/360969. [Accessed: 23-Feb-2017].
[7] A. Halfpenny and F. Kihm, “Rainflow cycle counting and acoustic fatigue analysis techniques for random loading,” in *10th International Conference on Recent Advances in Structural Dynamics RASD*, 2010.
[8] T. Irvine, “Rainflow cycle counting in fatigue analysis,” *Revis. Vib.*, 2010.
[9] M. Bonato and P. Goge, “Methods for analysis and comparison of automotive vibration tests,” in *2016 Annual Reliability and Maintainability Symposium (RAMS)*, 2016, pp. 1–6.
[10] J. Wægter “Note 5.1 - Stress range histories and Rain Flowcounting” June 2009.
[11] Thermotron Industries “Fundamentals of Electrodynamic Vibration Testing Handbook.” 2006.
[12] “S-N Fatigue.” [Online]. Available: https://www.nde-ed.org/EducationResources/CommunityCollege/Materials/Mechanical/S-NFatigue.htm. [Accessed: 19-Apr-2017].
[13] G. F. Lang, D. Snyder, and others, “Understanding the physics of electrodynamic shaker performance,” *Sound Vib.*, vol. 35, no. 10, pp. 24–33, 2001.
[14] T. Sarafin, P, Doukas, L. Demchak, and M. Browning, Instar Engineering and Consulting, Inc “Vibration Testing of Small Satellites” [April 11, 2014]
[15] M. Silverman, “Summary of HALT and HASS results at an accelerated reliability test center,” in *Annual Reliability and Maintainability Symposium. 1998 Proceedings. International Symposium on Product Quality and Integrity*, 1998, pp. 30–36.
[16] Mr. Neill Doertenbach, Senior Applications Engineer, Qualmark Corporation “The Function & Purpose of Repetitive Shock vs. ElectroDynamic Vibration” originally prepared for the 2014 RAMS conference
[17] “Repetitive Shock Stimulation for ESS, HALT & HASS.” [Online]. Available: http://www.qualitymag.com/articles/91406-repetitive-shock-stimulation-for-ess-halt-hass. [Accessed: 27-Mar-2017].
[18] J. Pei and H. Jianjun, “Characteristics and principle of the new cyclostationary and non-Gaussian omni-axis vibration environment,” in *2010 2nd International Asia Conference on Informatics in Control, Automation and Robotics (CAR 2010)*, 2010, vol. 3, pp. 117–121.
[19] *Applied Reliability Engineering*. RIAC, 2006.
[20] Thermotron, Thermotron Webinar: Innovative Advancements in Product Reliability with HALT/HASS. [Accessed: 2017-03-03]
[21] Thermotron, Thermotron Webinar: Unintended Consequences - The Importance of Table Uniformity with HALT/HASS. [Accessed: 2017-03-03]
[22] “Thermotron issued patent pending approval: Extreme Multi-Zone Vibration Control,” *News*, 20-Jun-2012. [Online]. Available: http://thermotron.com/news/thermotron-issued-patent-pending-approval-extreme-multi-zone-vibration-control/. [Accessed: 10-Mar-2017].
[23] “Multi-Zone Control | Thermotron Features.” [Online]. Available: http://thermotron.com/features/multi-zone-control. [Accessed: 10-Mar-2017].
[24] D. Kern and T. Scharton, “NASA handbook for spacecraft structural dynamics testing,” in *Environmental testing for space programmes*, 2004, vol. 558, pp. 375–384.
[25] National Aeronautics and Space Administration (NASA) “NASA Technical handbook, NASA-HDBK-7010” 2016-02-01
[26] J. C. Forgrave, K. F. Man, and J. M. Newell, “Acoustic and Random Vibration Test Tailoring for Low-Cost Missions,” in *Annual Technical Meeting-Institute of Environmental Sciences and Technology*, 1998, vol. 44, pp. 165–171.
[27]ECSS-E-HB-32-26A, “Space engineering, Spacecraft mechanical loads analysis handbook”, European Corporation for Space Standardisation, 2013.

[28] S. Goyal and E. K. Buratynski, “Methods for realistic drop-testing,” *Int. J. Microcircuits Electron. Packag.*, vol. 23, no. 1, pp. 45–52, 2000.
[29] “Drop and Shock Acquisition - Shock and Drop Testing - Drop Shock Tests, Applications - Data Physics Corporation.” .
[30] H. Berger, “Accelerated reliability testing for telecommunication systems using the highly accelerated life test (HALT) and highly accelerated stress screening (HASS),” in *TELESCON 2000. Third International Telecommunications Energy Special Conference (IEEE Cat. No.00EX424)*, 2000, pp. 49–53.
[31] A. Barnard, “Ten things you should know about HALT amp; HASS,” in *2012 Proceedings Annual Reliability and Maintainability Symposium*, 2012, pp. 1–6.
[32]Keithley Instruments, Inc.“Fundamentals of HALT/HASS Testing”. 2000
[33] “Piezo vibrations and piezo shakers for material characterization, modal analysis.” [Online]. Available: https://www.piezosystem.com/products/piezocomposite/applications/vibration_excitation/. [Accessed: 07-Mar-2017].
[34] B. Payne, K. K. Harper, G. W. Vogl, and E. P. Tomasini, “Piezoelectric shaker development for high frequency calibration of accelerometers,” in *AIP Conference Proceedings*, 2010, vol. 1253, pp. 373–382.
[35] “Accelerated Reliability Test Results.” [Online]. Available: http://www.udco.com/resources/applications/accelerated-reliability-test-results/. [Accessed: 17-Mar-2017].

8 Appendix

A. RS system PSD graph in x and y direction

Here are complementary PSD results from the RS system in the x and y direction respectively.

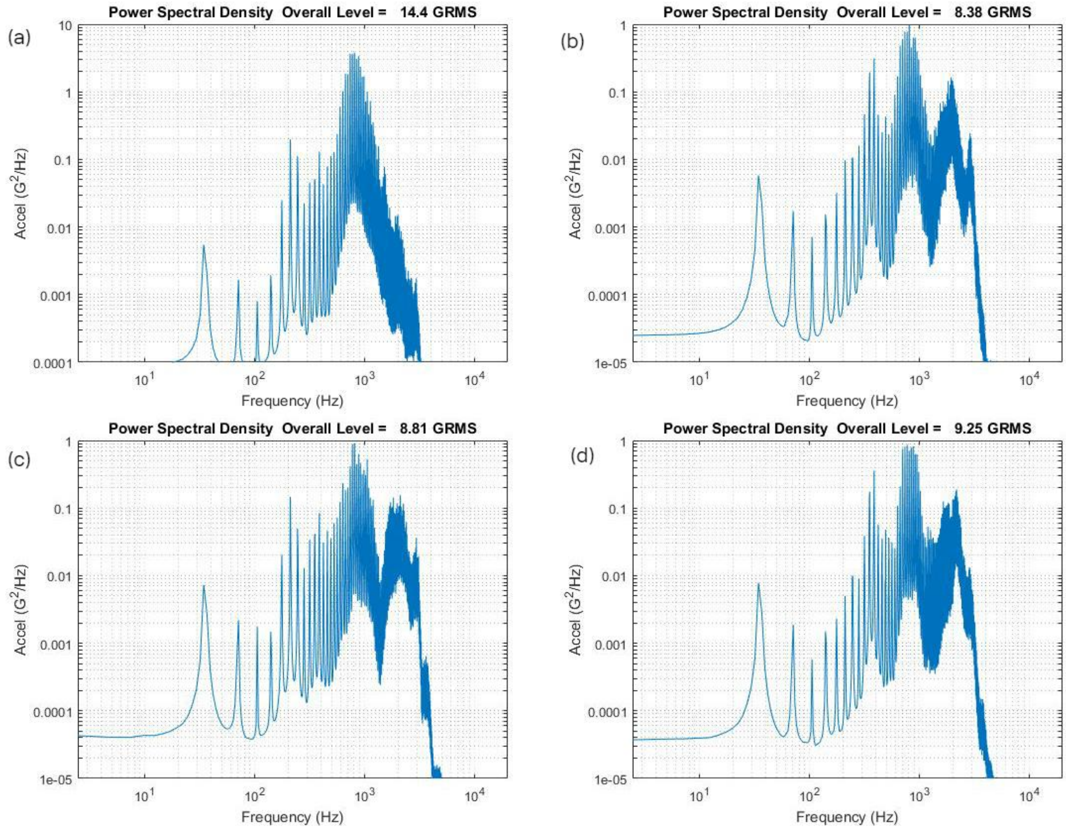

Figure 8-1. RS system empty table response X-direction

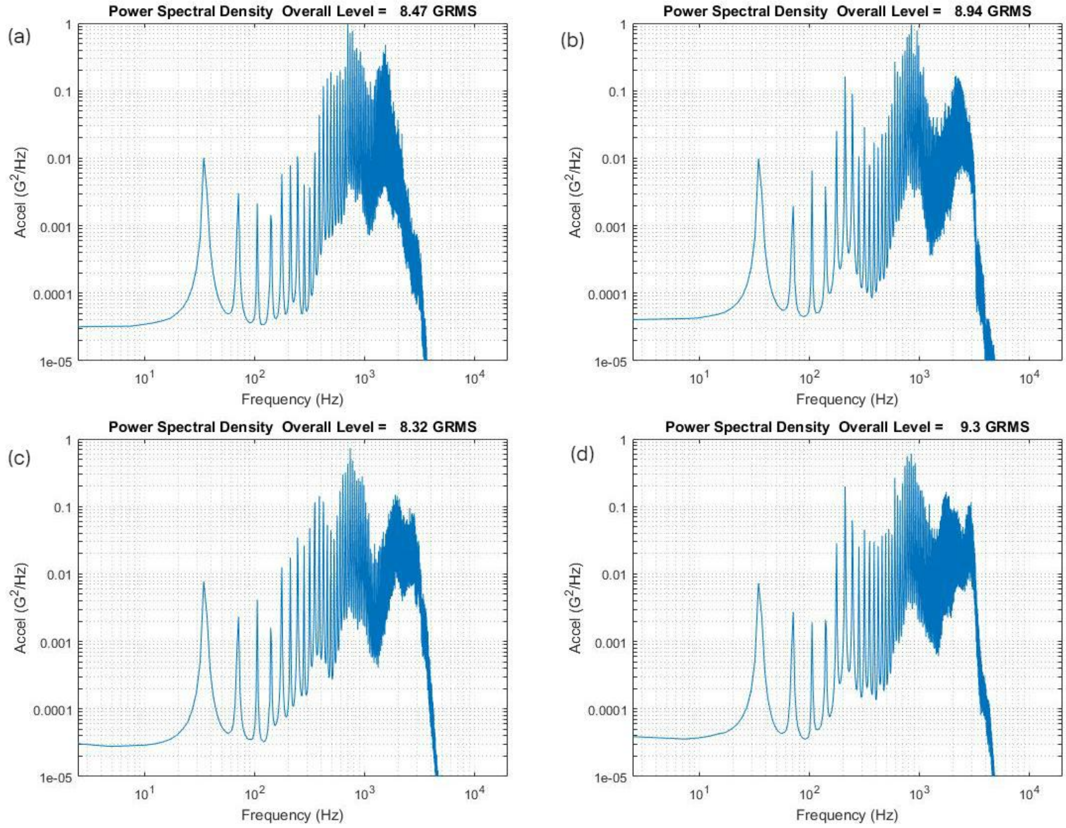

Figure 8-2. RS system empty table response Y-direction

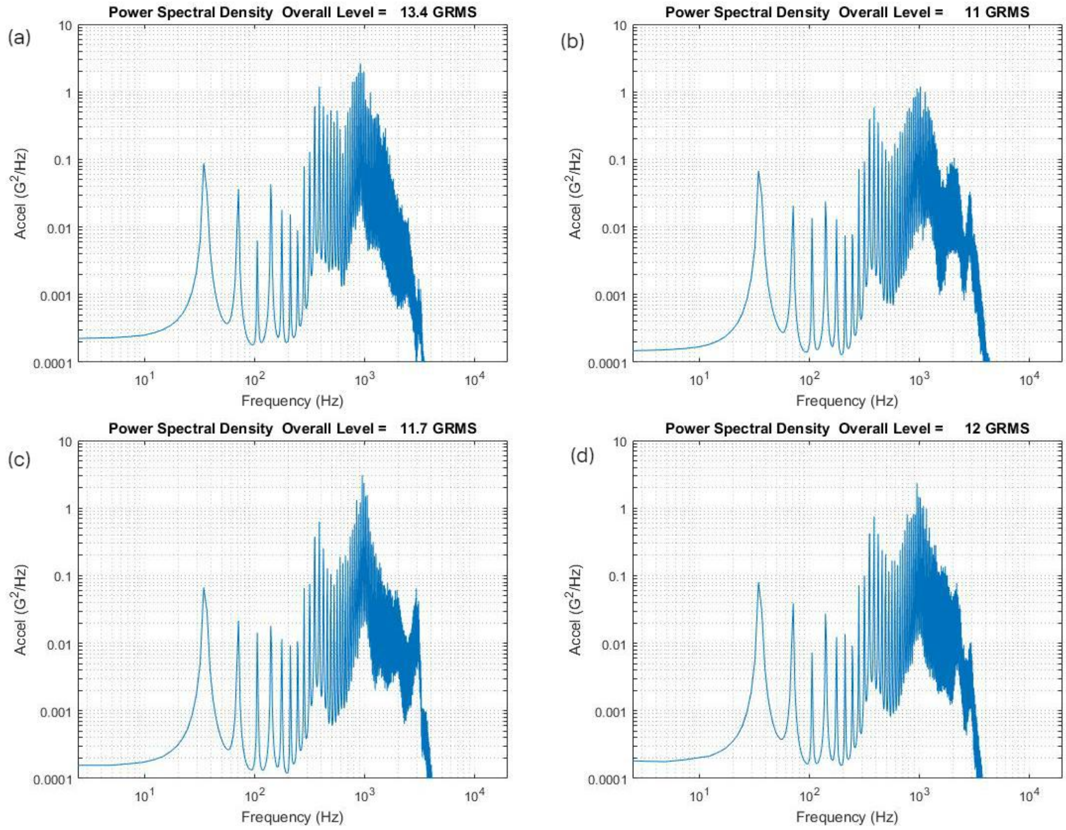

Figure 8-3. RS system empty table response Z-direction

B. Specifications on different Electrodynamic shaker systems

http://www.globalspec.com/specsearch/partspecs?partId={C6690380-1314-47AA-BA4D-47D2610D7AA0}&comp=2991&vid=93111&srchwebhitid=607479188&sqid=20192230

Specifications		Features	
Peak Sine force	12000 lbs (5442 kg)	Armature / Table Sizes	16"
Random Force - RMS	12000 lbs (5442 kg)	Payload Capacity	1000 lbs (454 kg)
Peak Acceleration	120 g	Stray Magnetic Field	Less than 8 Gauss
Displacement	2 inch (50.8 mm)	Technology	Electrodynamic
Peak Velocity	90 in/sec (137 m/min)		
Frequency Range	5 to 2750 Hz		

http://www.globalspec.com/specsearch/partspecs?partId={BA18B03E-EAE8-45B2-A60A-CA1DCBDAEC61}&comp=2991&vid=93046&srchwebhitid=607567944&srchitem=5&sqid=20192230

Specifications		Features	
Peak Sine force	1521 lbs (690 kg)	Moving Element / Armature Weight	Armature Effective Nominal Weight - 13.2 lbs
Random Force - RMS	1323 lbs (600 kg)	Armature / Table Sizes	Table diameter - 7 in
Peak Shock Force	2977 lbs (1350 kg)	Payload Capacity	662 lbs (300 kg)
Peak Acceleration	115 g	Technology	Electrodynamic
Displacement	2.01 inch (51 mm)		
Peak Velocity	78.74 in/sec (120 m/min)		
Frequency Range	4500 Hz		
Resonance	3500 Hz		

http://www.globalspec.com/specsearch/partspecs?partId={93468A23-0E95-4F2A-ACF2-4B2C628F46AC}&comp=2991&vid=93046&srchwebhitid=607567944&srchitem=12&sqid=20192230

Specifications		Features	
Peak Sine force	17640 lbs (8000 kg)	Moving Element / Armature Weight	Armature Effective Nominal Weight - 176 lbs
Random Force - RMS	17640 lbs (8000 kg)	Armature / Table Sizes	Table Diameter - 18.9 in
Peak Shock Force	35280 lbs (16000 kg)	Payload Capacity	2646 lbs (1200 kg)
Peak Acceleration	100 g	Stray Magnetic Field	< 10 gauss (1 mT)
Displacement	2.01 inch (51 mm)		
Peak Velocity	78.74 in/sec (120 m/min)		
Frequency Range	5 to 2500 Hz		
Resonance	2100 Hz		

http://www.globalspec.com/specsearch/partspecs?partId={5C5366EA-235A-4E51-B6B3-6FD3A3EE38C5}&comp=2991&vid=93111&srchwebhitid=607479188&srchitem=11&sqid=20192230

RS-16 Vibration Table

Specifications	
Peak Acceleration	50 grms
Sampling rate	100 kHz/channel
Frequency Range	2.0 to 20 000 Hz
Number of impactors	4
Features	
Armature / Table Sizes	16 x 16 in.
Payload Capacity	300 lbs (136 kg)
Technology	Pneumatic
Axes Excited	3 linear, 3 rotational

http://www.globalspec.com/specsearch/partspecs?partId={C61456CE-EBFB-46B4-B5B4-7F3C349E29B5}&comp=2991&vid=93111&srchwebhitid=619534461&srchitem=25&sqid=20267783

Specifications	
Peak Acceleration	50 grms
Sampling rate	100 kHz/channel
Frequency Range	2.0 to 20 000 Hz
Number of impactors	16
Features	
Armature / Table Sizes	48 x 48 in.
Payload Capacity	1,110 lbs (499 kg)
Technology	Pneumatic
Axes Excited	3 linear, 3 rotational