Test Fixtures for Vibration Testing of Components

PER FALK
Abstract

The aim of this thesis has been to try to improve vibration testing of electrical and electromechanical components that are used within Scania’s segment of the vehicle industry. To do this, guidelines regarding the design of vibration test fixtures has been developed. Since every fixture is mounted on a bottom plate during vibration testing these bottom plates have also been under study and a proposal for a standardized bottom plate is presented. In order to be able to easily test components with basic designs a modular fixture has also been designed, manufactured and evaluated within the scope of this thesis.

Implementation of the proposed guidelines for fixture design is most importantly argued to enable power transfer between the component and fixture during testing that as far as possible resembles the power transfer that takes place when the component is mounted on/in an actual vehicle. Using the standardized bottom plate and producing new fixtures to fit the same hole pattern is argued to decrease the work load for each vibration test considerably. Finally good results are shown regarding the performance of the modular fixture.
Sammanfattning

Målet med detta examensarbete har varit att försöka förbättra och underlätta dagens vibrationstestning av elektriska och elektromekaniska komponenter som används inom Scanias segment av fordonsindustrin. Detta har gjorts bland annat genom att föra fram riktlinjer till ett mer standardiserat sätt att utforma vibrationsfixturen. Dessa fixturen fästs i sin tur alltid på en bottenplatta vid vibrationstestning och därför har även bottenplattor utvärderats och ett förslag till en standardiserad bottenplatta har tagits fram. För komponenter av enklare geometri har en modular fixture utformats, tillverkats och utvärderats.

Acknowledgements

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1. Introduction

Trucks and buses can be utilized in many different applications with a wide variety of working conditions. A competitive market in for example long haulage services, forces companies to steadily work on increasing the working capacity of their vehicle fleets. This leads to a lot of stress on the different components on the vehicle. As technology keep on progressing the amount of electrical and electromechanical components put into trucks and buses increase in number. They therefore play an increasingly larger role in assuring the proper functionality of a vehicle.

Evaluating whether a component will be viewed as able to withstand the environment on a certain vehicle or not is a cumbersome job. The basis of this dilemma lies in formulating suitable and definable requirements that corresponds well to the components eventual working conditions. As a natural progression it is necessary to find accelerated testing procedures that relate to these set criteria, as it is a practical impossibility to test all components during the whole lifecycle of a vehicle. A few of these accelerated tests concern the vibrations that will occur on all components built into a vehicle, and these are something that may act very detrimental on certain components. One of the challenges regarding this type of testing is to make sure that the device under test (DUT) is mounted onto the vibration equipment in a similar way as it would be mounted on a vehicle while in operation. This is done in the pursuit of having the forces from vibration transferred into the DUT in a way that is the most similar to the power transfer from its actual working conditions.

A vibration test fixture is the interface between the device under test and the vibration equipment. However, the test fixture needs to be more stiff and rigid than the corresponding part used for mounting on the vehicle, since vibrations during accelerated testing are much more severe than vibrations during true operation of the component. The way in which a DUT is mounted during a vibration test can make the difference between the component passing or failing a test. The way it is mounted should neither add nor subtract energy from the applied test.
2. Aim and scope of the thesis

Early formulations of the goals of this thesis concerned how to design and simulate vibration test fixtures using engineering design software like Catia. Would it perhaps be possible to use Catia in order to simulate the behavior of the complete system comprising of shaker, fixture and test object?

Discussions with Senior Engineer and supervisor Dan Magnusson however lead to the desire to develop the project in three different ways. From the need of performing a vibration test, this thesis thus aims at aiding that process by proposing solutions into these three areas of focus:

- Standardized bottom plate for vibration shaker.
- Guidelines for fixture design.
- Modular fixture.

Chapter 3 will give the reader a short background of vibration testing of components and present a few important criteria in regards to it. In chapter 4 vibration and shaker theory is presented. Chapter 5 provides a deeper look into the possibilities of using a standardized bottom plate and chapter 6 and 7 separately focus on guidelines for creating fixtures for arbitrary components, and the way to design a modular fixture. In chapter 8 sustainability is briefly discussed before the conclusions and recommendations for future work are brought up in chapter 9 and 10.
3. Background

A truck or a bus at Scania are designs where the environment and requirements for components mounted on it and in it varies greatly. It is therefore necessary to make the distinction between differently intended usages and mounting areas. The way in which an object will vibrate on a vehicle strongly depends on the place of mounting. Objects close to the engine or transmission generally tend to vibrate at a much higher frequency than objects mounted inside the cab for example. This is due to the fact that the engine and transmission themselves show high values of vibration frequency. The cab is suspended with an additional suspension between the cab itself and the chassis, thus lowering the frequencies that reach the cab.

There are different standards for what a device under test should be able to withstand, often depending on the intended use. The criteria regard electrical performance, the ability to withstand liquids, abrasive wear and also temperature cycling, to mention a few from a vast list of demands. Scania has developed their own set of criteria that electric and electronic components must meet, but these are not allowed to reach third party. They are though somewhat similar to ISO-standard 16750, which is a public document and hence will serve as an example of what these criteria look like [1]. Part three of this standard regulates testing of mechanical loads such as mechanical shock, free fall, gravel bombardment and so on.

One aspect of these criteria covers what type of vibrations a component under test should be able to withstand while still being able to perform according to specification. Some functionality should be maintained during the actual testing. A more comprehensive test verifying all functional requirements should be performed prior to, and after, testing. The test procedure is meant to reproduce the same amount of stress on a component which would or could be experienced during a lifetime when mounted on a vehicle. Trucks and buses usually run a lot further than cars and a general guideline is that a lifetime of a truck is either equivalent to 15 years or 1 000 000 km of service.
4. Theory

This section will provide some of the necessary understanding about vibration testing that the subsequent chapters will continue building on.

4.1 Vibration theory

All bodies with mass and elasticity are able to vibrate. A vibration in this sense is a periodic motion where a body or structure will move in alternately opposite directions around an equilibrium position. Vibrations can be random or sine, as well as single axis or multiple axis [2]. Electrical and electromechanical components in the vehicle industry sometimes vibrate due to motions inherent from the system itself. Dominating in these components however is usually the type of vibration known as forced vibration, where excitations from other parts or structures are transferred into the component under study. The supports of a component can be seen as having two different types of vibration input. One part is periodic, from the driveline or engine for example, and one part that is random as a result of stochastic inputs from, perhaps, the surface of the road. [3]

A small input in energy and amplitude can still have immense effects on a component if it is being excited at, or close to, one of its natural frequencies. At a certain frequency a system or a component will vibrate in a particular way. The relative amplitudes and phase at different parts of the structure will describe what is called the mode of vibration. If a structure is being excited near its natural frequency energy will build up leading to motions much greater than intended in the structure and it will move according to the corresponding eigenmode. If the excitation at the natural frequency are allowed to continue the structure might eventually break due to the fact that the energy being stored in the oscillations are greater than what the structure are mechanically able to withstand. This is why it is so often of crucial importance to try to avoid having the natural frequency of a component within the same frequency range as the components mounting place will vibrate. If this cannot be avoided it might be necessary to look further into the damping qualities associated with the object to see if the energy can be dissipated through damping in a high enough rate to avoid damaging the component. The effects of damping and natural frequency is schematically illustrated in Figure 1.
Due to the varying vibrational behavior of different parts of a vehicle, the demands on components are also different depending on the place where they will be mounted. Components that are meant to be fastened close to, or in connection with, the engine or driveline will usually experience vibrations at a substantially higher frequency than will components on the chassis for example that are isolated from the driveline and engine. A very important criteria when deciding how to test a component is evaluating what type of vibrations it will be subjected to. It is possible to read more about the vibration specifics at different parts of a truck in the ISO standard [1].
4.2 Shaker theory

The shakers (or vibrators) are large pieces of equipment that make it possible to subject devices to vibration. Seen in Figure 2 is the shaker that has been used for all testing in this thesis work. To the right is the actual shaker and to the left a slip table is seen. This is used when the shaker is oriented in a horizontal position. Then the armature of the shaker will be attached to the slip table as in Figure 3. To be able to run tests in this dimension the slip table is placed on an extremely smooth stone surface. A hydraulic pump will pump oil through holes in this stone reducing friction between the slip table and the surface on which it rests. It is of extreme importance that the shaker is aligned perfectly with the slip table when it is used horizontally. Severe damage to the equipment might otherwise arise. There are a few different practical tools one can use in the process of making sure that the shaker is placed in perfect position before operation, but these will not be explained here. Just note that it can be a very time consuming process to get it to work properly.
Vibrators used at Scania works in a similar manner as an ordinary loudspeaker. In a loudspeaker the voice coil will pull in and out causing sound waves as a result. In an electrodynamic shaker it is instead the armature coil that moves in and out causing the desired movement. The motion relies on the phenomena of electromagnetism, where, if an electric current is passed through a coil, a magnetic field will be produced around it. The body of the vibrator houses four field coils, as can be seen in Figure 4 (lower and upper field coils). When these are supplied with a direct current they produce a static magnetic field that is required for operation of the shaker. This field will allow the armature to move if the armature coil at the same time is being fed with an alternating current. A degauss coil is fitted at the edge of the armature table. This component is also seen in Figure 4. The current running in the degauss coil is controlled by the amplifier with the aim of reducing the stray field above the armature by counteracting the fields produced by the other coils in the shaker.
Figure 4. Vibrator body. Cut-away view [5]. Note that the bottom plate mentioned here is not the same one as the one described in following chapters.

The amplifier used with this type of vibration equipment (seen in Figure 5) also have, just like the vibrator and the loudspeaker, similarities with a corresponding component from a musical context. An amplifier works, as its name suggests, by amplifying its incoming signal. Controlling the shaker is done by means of operating a personal computer. Via the computer one has the possibility to decide

Figure 5. Amplifier used for amplifying control signals going into the vibrator.
exactly what the vibration profile should look like. Type of vibration, levels of acceleration, etc. is possible to tweak to one’s specific needs. The amplifier then makes sure that the signals that are being fed to the vibrator are large enough to accommodate the desired output.

Forces are produced in the vibrator as a result of the interaction between the magnetic fields from the field coils and the current passing through the armature coil follow Fleming’s left hand rule. A visual representation of this rule is seen in Figure 6.

Fleming’s left hand rule applies to motors where the thrust that is produced is an effect originating from the magnetic fields and currents, which hence are the causes of this effect. Following this rule while studying Figure 4 it becomes clear that depending on the direction of the current flowing in the armature coil (remember that it is being fed with an alternating current) the resulting thrust of the armature will either be pointing up or down in the vibrator, which is well in relation to what one could expect.

4.2.1 Performing vibration testing

Having found a suitable bottom plate and fixture for the component under evaluation (more on these topics later) and a way to connect them all, there are still a few different aspects in need of consideration before starting a vibration test. Depending on the bottom plate, fixture and DUT the weight mounted on the armature can vary greatly. When a load is mounted on the armature, the armature will be forced downwards from its mean position. How much depends on the mass of the load where a greater mass will compress the suspension more than a lighter one will. To avoid unnecessary harm of the armature suspension (see the suspension assemblies in Figure 7) this position can be controlled by the internal load support (ILS).
The internal load support works with an air plenum and a rolling diaphragm together with pressurized air which, depending on the air pressure used, will force the armature upwards or allow it to travel downwards by gravitational force. Controlling this air pressure is up to the system operator and is done by manual regulation of a certain control on the vibrators pneumatic control box. Feedback is provided from a gauge connected to the optical measuring unit. The ILS is utilized when the amplifier is turned off. When the amplifier is turned on however, it will utilize information from the center positioning unit (seen in Figure 8) comprising of sensor and target strip, and allows for automatic control of the vertical position of the armature table. This is done by outputting a signal controlling the fields and currents in the shaker, to have it stabilized at its mean working position. Having the vertical operating position controllable is crucial for
utilizing the full amount of displacement that can be produced by the vibrator. If a heavy body is mounted on the armature, leading to a lowering of the suspension, and if it was not possible to counteract this motion that would restrict maximum displacement and hence hinder certain measurements that make use of strokes with large amplitude.

Figure 8. Left: Target strip. The optical unit for measuring the vertical position is seen as the component fastened with four screws in the bottom of the picture. In this figure the armature is raised well above the mean working position so it is possible to see almost the entire target strip. Right: Infra-red fibre optics unit for sensing the armature position.

Figure 9 shows the shaker that has been used in the experimental part of this thesis and on the sides of the main part (the actual shaker, in white) it is possible to see black air bellows connecting the shaker to the structure keeping it in place (red).
Figure 9. Illustrating the position of the adjustable bellows.

Figure 10 displays a close-up of one of these air bellows that should be adjusted prior to testing. They are adjusted with air pressure to improve damping as it will lead to better performance if they are inflated just enough to keep the suspension in the middle of its operable range. Before testing commences it is therefore important to make sure that the piston (that can also be seen in Figure 10) protrudes equally much on the top side as it does on the bottom side of its mounting/guiding holes.
Ideally, when performing vibration testing, one would like to have the exact same phase and acceleration in the shaker table, the test fixture and the nearest and most far away parts of the DUT (seen from the armature). Having the vibration equipment affecting the DUT in the desired way would then be an easy task. That would however require that all these components would be perfectly rigid. This is in reality not possible.

Assessing the vibration performance of a specific component can easily be done by using an accelerometer. The most important feedback in the control loop of the shakers also come from accelerometers. Regarding the placement of these when assessing a structure or component, it is a good idea to try to place them on the most vibrating part of a structure. Thereby one minimizes the risk of having the accelerometer end up on a node where the movement might be very small or even non-existent. The end parts of a structure are usually a suitable spot for attaching an accelerometer. It is not always sufficient to use just one control accelerometer and these trends tend to be more obvious when dealing with larger structures. If the system comprising of the vibrator, bottom plate, fixture and DUT have a natural frequency in the test range then having the control accelerometer mounted near the DUT allow for having the desired amount of g-forces transmitted into the DUT over the entire frequency range. This is possible
because the feedback allows for lowering the vibrator output at the natural frequency, thus keeping the input into the DUT at the desired level [6].

It is a good idea to run tests at low g-levels before commencing full testing to allow for some insight regarding the dynamic characteristics of the system. By doing this it is possible to pay closer attention to the system behavior when it is being excited with frequencies for example near an eigenfrequency. These characteristics should preferably be found empirically and it is not suggested to assume certain behavior based on performances from separate parts as the complete system might, and most probably will, behave differently when assembled than the parts would do one by one. This is something that relates more to heavier objects since a small and lightweight component that is being tested will not significantly influence the bottom plate or vibrator.

### 4.2.2 Shaker limitations

When studying values of acceleration and force output of the vibrators, it is obvious that the weight of the DUT in most cases of testing the usually quite small electric and electronic components related to this thesis is not an issue [5]. The vibrators are able to handle substantial load and still output great force and thus also acceleration. A limiting factor however is how the payload (bottom plate, fixture and DUT) is distributed in relation to the armature center. Most electrodynamic vibrators are designed to produce motions with just one degree of freedom. Hence the moving part of the vibrator, the armature assembly, is forced to follow a single axis. The armature suspension is designed to have very low stiffness in the direction of motion and to have high stiffness in the cross-axial directions. This stiffness, however, is limited. Brüel & Kjær, the company behind the LDS line of shakers (the type of vibrators used in this thesis) recommend following certain guidelines to make sure that the limits of cross-axial forces are not exceeded. Every shaker type is associated with two constants A and B that together with the distance X will give the Offset Load Formula according to equation (1) when shakers are operated horizontally as in Figure 11 [5].

\[
F = \frac{A}{B + X}
\]  

(1)
This formula is valid when the payloads center of gravity is in line with the midline of the armature. A is the maximum allowable turning moment in Nm for a specific type of vibrator. For a V875 vibrator, with a 640 mm armature (the type used during all vibration testing related to this thesis), the value of A is 565 Nm. B is the horizontal distance for that same vibrator from the mounting face on the armature to a point around which the system can be considered to rotate. Its value is 0.263 m. X describes the horizontal distance from the mounting face to the payloads center of gravity and F thereby becomes the limit force in N as a function of A, B and X. The payload is made up of the bottom plate, fixture and DUT and it is their joint center of gravity position that is interesting.

Apart from this phenomenon it is, due to practical reasons, not always possible to mount payloads with their center of gravity symmetrical with respect to the vibrator axis. If this is the case, special consideration need to be made to make sure not to damage the shaker equipment. The detrimental aspects of having misaligned center of gravity during vibration testing (even when shakers are upright) are portrayed in Figure 12. Alternating moments might damage the suspension and bearings of the armature.
When using the shakers upright the value $A$ from equation (1) is instead used as a limit for the maximum moment that is allowed to be exerted on to the armature assembly as a result of the motion of the payload.

Having an offset center of gravity as well as a horizontally rotated shaker, as in Figure 13, will lead to a case when it is necessary to combine these two different contributions of moments of force to evaluate what the maximum level of cross-axial acceleration is. This level must not be exceeded due to the limits of the shaker. In Figure 13 an $M$ kg payload is to be tested at a horizontal acceleration level of $n \ g$. The following calculations will show what value to keep in mind as the maximum for cross-axial acceleration of the payload. It is then possible to monitor this with an accelerometer to make sure that it is not exceeded.
The moment of force stemming from the X m horizontal offset of an M kg payload in Figure 13 is for a V875 shaker with a 640 mm armature, with g being the standard gravity:

\[ M(B + X)g \text{ Nm} \] \hspace{1cm} (2)

and the moment induced by the Y m offset in the vertical direction when the payload is being shook at n g is:

\[ M \times n \times Y \times g \text{ Nm} \] \hspace{1cm} (3)

The total moment is thus

\[ M(B + X)g + M \times n \times Y \times g \text{ Nm} \] \hspace{1cm} (4)

and since the total allowable moment is A Nm the maximum allowable value of the moments induced by cross-axial motion is then

\[ A - [M(B + X)g + M \times n \times Y \times g] \text{ Nm} \] \hspace{1cm} (5)

and this is equivalent to a force of

\[ \frac{A - [M(B + X)g + M \times n \times Y \times g]}{X + B} \text{ N}. \] \hspace{1cm} (6)

Therefore the maximum allowable cross-axial acceleration of the payload is thus

\[ \frac{A - [M(B + X)g + M \times n \times Y \times g]}{n(X + B)g} \text{ m/s}^2. \] \hspace{1cm} (7)

Equation (7) however requires a perfectly rigid payload and fixture to be accurate.
Something that rarely is the actual case in real life testing. As a means of making the calculations more true to reality it is preferred to monitor the cross-axial accelerations at the face of the armature and hence mounting the control accelerometer there, instead of mounting it on the DUT. The value to use as a limit is then in need of recalculation according to the following equation:

\[
\frac{A - [M(B + X)g + M \times N \times Y \times g]}{n(X + B)g} \times \frac{B}{B + X} \text{ m/s}^2. \tag{8}
\]

Assuming knowledge of the parameters A and B regarding the vibrator and X, Y and M related to the mass and center of gravity of the payload together with the desired level of acceleration n the previous equations have been used to make sure that the vibrator has not been used in excess of its own limits in this thesis.

### 4.2.3 Fastening method and natural frequency

Characteristics of bolts used for fastening of bottom plates, fixtures or components need special attention. Ideally they should be as short as possible to prohibit de-coupling due to the stresses that they are put under while testing of components. A rule of thumb states that the distance between the tapped hole and the underside of the screw head should not be any greater than 2.5 times the diameter of the shank [5].

If close attention is not shown towards making sure to screw different parts together in an adequate way a major resonance will arise. There is an equation coupling the payload mass M with the stiffness of the system K to give an estimate of the mounted systems natural frequency.

\[
f = \frac{1}{2\pi} \sqrt{\frac{K}{M}} \tag{9}
\]

where

\[
K = \frac{EA}{L} \tag{10}
\]

utilizes the Young modulus for the material of the screws, E, the combined stress area of the screws, A, and the screw length L.
5. Standardized bottom plate

One part of this thesis work was to find an appropriate candidate for a standardized bottom plate to fit the fixtures on while components are being tested. The bottom plate is mounted on the armature face and then the fixtures are mounted on the bottom plate. In use right now are bottom plates with three different thicknesses, namely 20, 25 and 30 mm each. As there is no standard pattern for holes on these bottom plates new holes are usually drilled and threaded for each fixture in use. After a while it becomes harder and harder to find a suitable way of mounting the fixtures as there is no longer a way to find exact positions for new holes that does not clash with existing ones. See for example the bottom plate depicted in Figure 14.

![Figure 14. Typical bottom plate after usage with a large number of different fixtures.](image)

This leads to a two-part challenge. Firstly it’s of interest to choose a thickness for this standardized bottom plate and secondly a hole pattern that will make it less cumbersome when it comes to fitting the fixtures onto the bottom plate. This will not eliminate the need of sometimes having to drill new holes in a bottom plate. It will however make it easier in the future if new fixtures are designed according
to the chosen pattern. To aid in the process of choosing a suitable thickness for the standardized bottom plate, measurements have been carried out on the three existing thicknesses.

### 5.1 Experimental study

Each of the three different bottom plates was mounted on the vibration armature and all possible connection points to the armature were used. 45 M8 bolts used together with stiff washers were tightened with a torque wrench to 34 Nm as stated in the vibrator manual [5]. The bolt pattern is obvious in Figure 15 where the attachment points, the hexagonal raised parts, on the armature face are shown without the presence of any bottom plate.

![Figure 15. Pattern of attachment points on armature face.](image)

Once a bottom plate was mounted on the vibrator two accelerometers had to be put on it. One will control the drive of the vibrator, and one will be used together with a laser measurement system. The one controlling the drive is mounted with the intention of providing relevant feedback to the system controlling the vibration. If it turns out that, perhaps, a resonance is being excited and that it leads to motions with a too high amplitude or acceleration the program will end the test prematurely to avoid damaging the equipment. The laser measurement
system is provided by Polytec and consists of a scan unit and the Polytec Scanning Vibrometer software. The scan unit will (by means of laser) record the movement of the item under study and can be used with the software to display accelerations, displacement and other data.

The practical setup of the laser scanning head and the bottom plate can be seen in Figure 16. When using the laser measurement system the best results arise from having the component on which one is conducting measurements at a right angle to the incident laser beam. That is the reason for having the shaker tilted at an angle that is as close as possible to 90 degrees to the incoming laser beam. Attention was directed to making sure that this criteria was met by the use of an existing fixture which has the property that its sides are perpendicular to one another. By putting this fixture with one side flat to the center part of the fixture and then adjusting the tilting angle of the shaker on one hand, and the height and placement of the laser scanning head on the other hand it was possible to make sure that they were positioned in a way which later on would yield the best possible outcome from the measurements. The edge of the fixtures side that was perpendicular to the armature face then had to show half of the laser beam and the other half should be found at the spot where the two sides come together for this alignment to work.

![Figure 16. Test-setup when evaluating bottom plates using the laser measurement system.](image)
One necessary input into the controlling program is what measurement points you would like to use, since the program requires discrete points for measurement. These points can be chosen arbitrarily or can be made up from a certain amount of different predefined patterns, which by means of their input can be altered slightly. Choosing a predefined pattern for measurements makes it possible to later display results shown as a surface, were the computer software then will interpolate data to the areas that were not covered by one of the discrete measurement points. This is the reason for choosing a circular predefined pattern in this thesis. Due to the fact that the tested bottom plates all had holes in different positions it was not possible to use the same measurement pattern for any of them. It was necessary to find individual patterns for each plate. The choice of pattern when performing analysis on the 20 mm thick bottom plate is represented in Figure 17. Because of the lasers inability to measure correctly on highly reflective surfaces, all of the bottom plates’ aluminum surfaces were sprayed with a matte white paint during measurements.

As seen in Figure 17 the measurement points were not distributed in a completely symmetrical way in relation to the geometry of the bottom plate. It is hard to get
good readings from the tests if the points of measurement would coincide with one of the holes in the measurement plate. That’s the reason why thorough consideration was shown to make sure that the measurement points did not end up on any bolt or hole. Therefore the pattern is located slightly off center in relation to the bottom plate, since the measurement point in the center would otherwise end up on the same location as the center mounting hole. In Figure 17 the blue dots are not displayed with high enough resolution to depict the fact that the measurement points and holes/bolts never did coincide but attention was directed towards making sure that this was not the case before measurements began.

The first measurement conducted on each of the three bottom plates were white noise measurements. The eigenfrequency of the armature used for testing has a stated nominal value of 2200 Hz. Therefore the decision was made not to run the initial white noise test over 2200 Hz. It was ran from 20 to 2200 Hz as that well covers the practical testing range for all different vibration profiles a DUT can be subjected to at RECT. It was found that this spectrum was able to excite the first eigenmode when all the different bottom plates were used. Having found a peak in the frequency response spectrum for each of the bottom plates allowed to use that value as a midpoint for a future sine sweep. The sine sweep was performed with the intention of determining the exact value of the systems eigenvalue. This comparison was done for the two bottom plates with thicknesses 25 and 30 mm, respectively. It turned out that the eigenvalue found via the white noise spectrum was in close compliance to the value obtained from the sine sweeps in both cases. Therefore it was deemed unnecessary to do a similar comparison also for the 20 mm bottom plate.

A sine sweep however is not compatible with the laser measurement system. Since the laser system cannot measure simultaneously in all of the scan points, it is not possible to get valuable readings from sine sweeps with it. Seen in the animations from the initial white noise test was that the bottom plate had the greatest amount of movement in the middle when moving at the first eigenfrequency, see Figure 18. This was the reason for mounting an accelerometer as close to the middle of the plate as possible when performing the sine sweep.
5.2 Results

A summary of the results are shown in Table 1. It must be noted that these are not the true eigenvalues of each bottom plate, but the eigenvalues of the complete system that is made up of the armature having a bottom plate attached to it. The natural frequencies of each bottom plate when not fitted on the armature table can also be found experimentally but this has little interest since a bottom plate is never used without a shaker. As for the motion of the eigenmodes in Table 1, they all had a similar motion to what is seen in Figure 18.

<table>
<thead>
<tr>
<th>Thickness, mm</th>
<th>Resonance frequency, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>1895</td>
</tr>
<tr>
<td>25</td>
<td>1943</td>
</tr>
<tr>
<td>30</td>
<td>1924</td>
</tr>
</tbody>
</table>

*Table 1. Measured natural frequencies when different bottom plates were mounted on armature table.*
5.3 Bottom plate hole patterns

The bottom plates used at Scania today have to be adapted, as mentioned earlier, to facilitate the mounting of fixtures on the bottom plates' surfaces. Figure 19 shows a 30 mm thick bottom plate (left), which has not been used with many different fixtures. It has few holes in addition to the ones used for fixing the bottom plate onto the vibration armature. The 20 mm bottom plate to the right, on the other hand, has been used with considerably more fixtures. In an attempt to get away from having to drill new holes for newly produced fixtures so often, it has been under consideration to find a uniform way of spacing the mounting holes on the bottom plates, and of course then also the corresponding holes in the fixtures.

Of all the fixtures mounted on the different bottom plates, the absolute majority of them are fastened with M8 screws. Thus the chosen type of threads for the standardized bottom plate is M8. A suggestion regarding the pattern is that the holes should be spaced 50 mm apart from one another, in both x- and y-direction. That is when the surface parallel to the armature face is defined as the z-direction in a right handed coordinate system. This pattern allows for having a good resolution of holes but at the same time not making the number of holes extremely large. Having them spaced this way is also beneficial since the department at Scania that facilitates testing of mechanical components use the same pattern. The vibrators are despite of their robust look quite sensitive and it
is not uncommon that they break down occasionally. Having the same type of pattern for holes on the bottom plates between the departments makes it easier to maintain functionality within the company since departments then can utilize each other’s equipment if there is need and a possibility to do so. It will also decrease work load for the personnel performing the practical testing not just by removing most of the drilling and threading work but also by not having to change the bottom plates between different tests. The bottom plates are quite heavy and to torque the 45 M8 bolts requires both time and effort. Figure 20 shows the proposed pattern on a sketch of a bottom plate.

![Figure 20. Proposed bottom plate sketch with holes for fixture fastening 50 mm apart in both x- and y-direction.](image-url)
6. Guidelines for creating fixtures

In an attempt to shorten lead times there are many advantages to be found in being able to design and perhaps also validate fixtures before the actual component to be tested is produced. If possible, it would be beneficial to start designing the fixture as soon as the design of the component is done. By doing that one will make sure to minimize the time where one have the component at hand but not a suitable fixture. Electric and electronic components in the part of the vehicle industry where Scania is active have an enormous and increasingly growing set of different characteristics. It is because of this wide spread hard to find rules to ease the fixture making for specific components in an applied way, but it is believed that a few general measures can be taken to simplify and speed up the process of designing test fixtures.

As a starting point it can be argued that the working crew responsible for designing the interface for a certain product could be helpful when it comes to creating test fixtures. Ideally the test fixture and the mounting place on the vehicle have the same contact areas with the component. This is something to strive for since it makes sure that, for example, power transfer and spatial limitations as closely as possible mimics the components actual working conditions. This is also mentioned in SS-ISO 16750 [1]. A way to reach this goal is to utilize the surface on the vehicle that will be the interface towards the component. If the design engineer behind this surface is able to extract only the surfaces that relate directly to the component, this could be of great use for the fixture designer. There are commands in CAD programs like Catia that can easily cut away unnecessary parts of a construction to only leave the vital parts regarding the interface. Using this interface would lead to having the design, dimensions and tolerances related to it already set for the fixture to be. If the component have a complicated area of contact towards the rest of the vehicle it is not very efficient to have to start designing this interface again, from scratch. Having it exported to the fixture designer he or she would save both time and effort and make it possible to shorten lead times considerably for the component in question. Once the designer of the test fixture has access to the extraction of the contact surfaces that person can start using them in order to produce a sketch of the fixture. With experience from vibration testing and fixture design, the person making the fixture would use his or hers skill to modify this “extraction” into something usable in testing.

Since aluminum is usually the material of choice in fixture manufacturing at the department of Scania where this thesis has been performed, the guidelines must be able to meet the standards that applies when it comes to working with
aluminum. The aluminum in the fixtures is usually milled into the desired form, starting with a massive piece of the material. For large fixtures where it is necessary to attach different parts to each other, two or more pieces are welded together, or, guide pins are used in connection with screws. No matter which one of these procedures are used in manufacturing, there is a possible problem and that is when one of the areas of the interface is curved since it is very cumbersome to produce curved surfaces of aluminum. If a component is to be mounted to something made of plastic, surfaces are rarely absolutely flat, or even perpendicular to one another. The reason for not having the sides of a slot, for example, perpendicular to one another is related to the manufacturing. The component seen in Figure 21 for example does not have sides that are perpendicular to one another. When plastic pieces are produced via injection molding a draft angle is used to be able to separate the mold and the produced part once it has hardened. There are however, depending on the actual geometry, ways of getting around this problem.

![Figure 21. Bottom side of module with a few different controls on the other side.](image)

In Figure 22 for example, the component to be mounted in the depicted slots (the module seen in Figure 21) do not make contact with the entire walls of the slot. It will make contact in the grooves and the cutouts in the walls of the slot only. The narrowest part of the slot is the top, i.e. where the cutouts are made. When doing a virtual installation of the component into its slot in Catia it is clear that the component will not make contact further down the sides of the slot. Therefore there is no obvious reason why the fixture simulating the slot seen in Figure 22 can’t be produced with flat sides at right angles to one another. The contact surfaces between the fixture and the slot will
remain the same even though this simplification is utilized. Even if this would not be case, the method of having flat surfaces that are more easily produced by milling would most probably prove efficient and produce better results than the alternative. The alternative in this case is usually to attach the component to a fixture in a way that is nowhere near how it is done on the vehicle. As of today the mostly used fastening method for components that doesn’t have a particular fixture designed for them is a method where the component is fastened to a standard fixture by using cable ties, as in Figure 23.

![Figure 23. Components strapped to standard fixture using cable ties.](image)

This method of attaching components to a fixture might not only prohibit the natural motion of the DUT but will also have the forces transmitted into the component in a way that is widely different from what will be experienced in its eventual application. Even if a components electrical performance is unaltered after vibration testing, there might be damage caused to the housing. This unwanted behavior might never be found in testing if the component is mounted to a fixture with cable ties, and hence, if the component is prone to act faulty when correctly mounted, errors might not show until it is operational on a vehicle.

If the housing of a component for example is compromised it might facilitate the entry of water, which in later stages can lead to a faulty performance. If the component has a plastic housing meant to be fastened in the vehicle by the means of snap-in parts, these fasteners might be a critical section of the DUT. Testing a component without making use of these fasteners might not show if the fasteners
actually are not designed or produced well enough to withstand the strains they will be subjected to during operation. If these were to break, the whole part might come loose from its mounting position, which could lead to subsequent failures of many kinds. It might be left hanging in an electrical wire and/or end up somewhere where it will influence other components negatively, apart from losing its own functionality.

The shaking of individual components does not reveal the entire picture of performance testing when it comes to vibrations. Field tests are performed where trucks are fitted with components as they later on would be in true operation and used in the industry. This step might help show some of the problems related to vibrations for articles that have not been shaken while mounted like in real operation. Driving on a test track where different kinds of bumps and surfaces have been prepared is another way of testing. There are also rigs that will shake a big part of a vehicle simultaneously. The profile used for shaking is a recording of what happens at an actual test track. It is made to simulate driving around and around on the test track. If there are any latent faults related to the mounting of components, the stresses from this type of testing might also help to identify them.

One immense benefit of utilizing the virtual environment projected by the means of programs like Catia is that it is possible to use for making a virtual fitting. When evaluating the fit between a component and a fixture this tool will come in handy. It is a useful tool whether the surface of the components interface is used or if the fixture is designed from scratch. One of these virtual mountings/fittings is seen in Figure 24. The method does not work perfectly for all types of geometries and methods of fastening though. When a component is to be attached to the fixture with snap-in technique there are certain challenges due to that the snap-in parts in real life will deform and then relax at its set position. A virtual fitting can still be used however to give a good estimate of the degree to how well the pieces will fit together.
Figure 24. Virtual fit between a component and a fixture.

The fixture seen in Figure 24 (blue) is the three dimensional representation of a fixture that has been designed during this thesis in cooperation with the person who is in charge of the practical testing at RECT. It is to be used when testing the module that is seen in the same picture.
7. Modular fixture

Certain devices that are to be tested might have a very simple design. There might not exist a need for having to design a brand new fixture for these components but it might not either be a good idea to, as an example, just fasten them with cable ties to an existing fixture. As explained earlier this might inhibit the natural motion of the component which will decrease the similarities with the known applications and it will not allow forces to be transferred into the DUT in a way that mimics real working conditions as much as possible. In these cases a modular fixture could be of great use. When considering a modular fixture there have been a few properties that has been thought of as more important than others. Preferably the fixture should be:

- Versatile
- Reusable
- Durable.

Regarding material it has been mentioned that aluminum show superb qualities when vibration testing is being performed so therefore also this fixture is chosen to be made of aluminum. There are, apart from material considerations many other alternatives to think through when designing a modular fixture. For example, when deciding upon how components should be attached to the fixture there are many different techniques available. After discussions with the person who is in charge of the practical vibration testing at RECT it was deemed most appropriate to use threaded holes for attaching components to the modular fixture. Usually components that need screws for fastening utilize M6 or M8 screws.

To have the fixture able to also fit components with varying dimensions there’s a few different alternatives at hand. Possible for use are systems using holes spaced in various ways, different types of grooves and many more. Inspiration and overviews of different systems was found in [7]. The need of being able to alter the fastening dimensions when using different components and also the need of having to be able to use various thread sizes for attachment made the t-slot system an appropriate candidate for a modular fixture. To facilitate the use of different thread dimensions the choice was made to use interchangeable inserts with different diameter threads. A first version of how such a system can be designed is seen in Figure 25. In this picture two modular fixtures are present. Producing one fixture that attaches flat onto the bottom plate and one that erects from the surface is very time efficient when it comes to testing since it will allow
components to be tested in the three dimensions without the need of having to tilt the vibration equipment.

The “flat” fixture allow testing in z-direction (when the shaker is upright) and the upright standing fixture enables vibration of the components in x- and y-direction by simply turning the DUT 90 degrees in relation to the upright fixture between shaking events. With this setup the vibrator can hence be used in vertical operation throughout testing in all three dimensions which will save valuable time as there is no need to align it for use with the slip table. This process of tilting and aligning the shaker does not take very long if all go according to plan, but when there are problems in the process it can take an entire afternoon or possibly even longer. From an effectivity point of view and the need to minimize handling time between shaking events as an attempt to keep up the time used for shaking, there are major upsides in not having to tilt the shaker. Increased efficiency will shorten lead times and hence will lead to more tested products in an allotted time frame.

Figure 25. Proposed modular system for attaching components with simple geometries. Orange parts represent t-slot nuts with threaded holes.
When designing the system apparent in Figure 25 attention was deliberately put towards making it possible to mount the fixtures on the previously recommended standardized bottom plate. It was also viewed as a positive aspect to have the grooves in the fixtures align with one another. This concept enables components that have points of fastening on two sides (located in the same plane perpendicular to the armature surface) to be screwed onto both fixtures at the same time.

7.1 Designing prototype

Building on the idea presented in Figure 25 it was decided to design a prototype of the flat fixture and to then have that prototype produced. The fixtures seen in Figure 25 were deemed too large for initial production so from discussions with Dan Magnusson the decision to make a smaller version was made. Out of the three criteria presented earlier, the first two, namely that the fixture should be versatile and reusable, are both met by using the t-slot system. In the process of doing a more specific design then also the third part of these criteria, which stated that the fixture should be durable, must be evaluated and taken into consideration. Inspiration was drawn from [8] when evaluating what dimensions to use for the grooves in the fixture and hence also the t-slot nuts. The design must be robust enough to be able to withstand the stresses stemming from the vibrations and the loads attached to the fixture. The drawing of this component can be found in Appendix A and the three-dimensional sketch is found in Figure 26. When the components final design was decided upon there was no plan in place to order a bottom plate utilizing the suggested standardized pattern so this fixture does not match that hole pattern. The holes in the fixture are made to fit M8 screws and are counter bored with a dimension that will fit washers and make sure that the screw heads does not protrude over the plane on which the respective holes have been drilled. It is important to use washers in order to minimize the risk of having the bolt heads sink into the material thus lowering the clamping force.
7.2 Evaluating the prototype

After having the prototype and the t-slot nuts manufactured and delivered, see Figure 27 and Figure 28, testing was the natural step that followed.
Interesting to find was how the addition of this modular fixture to the shaker system would affect natural frequencies and what the motions in the structure would look like. Since this first prototype of a modular fixture was not so large a decision had to be made about where to mount it on the bottom plate.

In Chapter 5.2 it was described that the motion of the bottom plates (regardless of their thickness) had the largest amplitudes in the center part and in the outer parts. The least amount of movement was found on a firm radius roughly coincident with the second most outer ring of tightening bolts. It was decided to test the modular fixture for the worst case scenario where the movement was as large as it possibly could be so the fixture was mounted in the center of the bottom plate, as presented in Figure 29. To facilitate this attachment holes had to be drilled in the bottom plate and these holes were then threaded. This was done on the 25 mm bottom plate.
The setup of the test was very much like the one presented in the chapter regarding the bottom plates. With the fixture in place and the screws torqued according to standard the shaker was oriented as it was when it was used for testing when there was no fixture on the bottom plate. Regarding a measurement pattern the choice was made to use a rectangular one. The results showed that the motion that previously was seen in the bottom plate totally dominated the movement in the fixture as well. It was earlier found that 1943 Hz was the natural frequency of the system when the 25 mm bottom plate was the only thing mounted on the armature. Having the fixture added led to the lowering of the natural frequency of the system, which instead showed at 1716 Hz. This increase in weight will decrease the rate at which the system will resonate with, since it increases the systems inertia. Even though the fixture was designed to be very stiff it isn’t large enough to counteract the lowering of the frequency that follows from adding a weight of almost 2.7 kg to the center part of the system.

Of course the fixture will not be of any use if it does not perform well when a component is mounted to it. Therefore the fixture was also evaluated with a load attached to it. Serving as a load was an old existing fixture, weighting 1.7 kg. It had four holes in it already that were used when screwing it to the modular fixture using the four M8 t-slot nuts. Please refer to Figure 30. It would have been possible to use the two remaining holes that already existed on the load, but since the holes in it were made for M8 screws, and only four t-slot nuts utilizing a M8
thread was produced, the conclusion was that it wasn’t suitable to use any of the M6 t-slot nuts. It was hard to get a good visualization of the motion in the actual fixture when the load covered large parts of it. By assigning individual measurement points on the fixture on the surfaces that was not covered by the load, it was found however that the motion during the first natural frequency was coherent with the movements seen in earlier testing in the z-direction. Due to adding more weight, in addition to the fixture, the load made the natural frequency drop further, now reaching below 1700 Hz.

Since there has not been an upright modular fixture produced during the time of this thesis work, testing has also been performed with the fixture mounted on the slip table. This will allow for usage of the modular fixture in all three testing axes. If an upright fixture had been produced, and if testing proved it to meet the standards, then it wouldn’t be necessary to tilt the shaker like what is seen in Figure 30, but it is a valuable alternative when there is just one flat fixture produced.

During testing when the fixture was shook in horizontal direction the laser was not used. The area parallel to the direction of vibration was very small and hence it was deemed that the laser would not yield any further information than what would be found using an accelerometer. When at first testing the fixture without
any load the measurement accelerometer was located at the most distant edge away from the vibrator. The only natural frequency found in the test spectrum (20 to 2200 Hz) was 1743 Hz. Interesting to note is that when the load seen in Figure 30 was attached and the accelerometer as in the picture was mounted on the upper part of the load the natural frequency did not change. Even with the load present it was located at 1743 Hz.

Visual inspection after testing showed no signs of any kind of any bending, scuffmarks or other anomalies that may have arisen from the tests. Not on the fixture, load nor on the t-slot nuts.

8. Sustainability

Implementation of the suggestions made in previous chapters, mainly regarding the standardized bottom plate and the modular fixture, will enable testing of components in a more environmentally friendly way. Lesser need to produce new bottom plates and new fixtures will lead to smaller consumption and less waste. There is no natural shortage of aluminum but by not having to produce as many new parts as before the use of energy correlated to producing suitable bottom plates and fixtures will also decrease.

The overall increase in efficiency from also implementing the guidelines for fixture design will shorten lead times and by not having to change bottom plates as often as before this will lead to more tested products in an allotted time frame.
9. Conclusions

From the formulations in chapter 2 this thesis work has been performed with the goal of aiding the vibration testing process in three main areas. One area has been about the standardization of a bottom plate. Another focus was trying to establish general guidelines for creating fixtures and the final part has been about creating and testing a modular fixture.

Designing and producing a standardized bottom plate is a reasonable way of minimizing future alterations before commencing vibration testing. It will limit the need of adapting the bottom plates used for testing if subsequent fixtures make use of the same hole pattern as does the bottom plate. The standardized hole pattern will also make it possible for different testing departments to utilize each other’s equipment when and if there is a need to do so. Within each department that will use the standardized bottom plates the work load will also decrease regarding drilling and threading holes but also by removing most of the need to change bottom plates between measurements.

The guidelines proposed for fixture design would if implemented lead to a way of transmitting forces into the tested devices that better resembles their actual working conditions. By doing this the results from the vibration testing would better be able to portray the possible problems that might arise from real use within the vehicle industry. Utilizing the guidelines would also make it possible to shorten lead times between having finalized the design of a component and its subsequent testing since it by following the recommended workflow is possible to design the fixture as soon as the components design is decided upon.

Regarding the modular fixture it has been shown that the suggested design is sufficiently stiff and well designed not to induce any eigenmodes or natural frequencies within today’s frequency range of testing. This also holds true when it is used with a load. The system utilizing t-slot nuts for fastening components was shown to perform well, at least when subjected to short periods of testing.

The suggestions and results presented in this thesis form useful and relevant information which can help improve and simplify parts of the vibration testing process seen today. Therefore the goals seen in the statements of chapter 2 are viewed as being fulfilled.
10. **Recommendations to future work**

As a means of continuing the work that has already been performed it would yield interesting information to prolong the testing of the modular fixture to see if this might act detrimental on the fixture or its corresponding t-slot nuts. It would also be of interest to try to find the limit when the fastening system utilizing the t-slot nuts cease to be able to keep the DUT in place in a good enough way in cases where heavier devices is to be tested. Results stemming from this type of testing could then be of great use before finalizing the prototype design of the upright modular fixture. When working with the upright modular fixture and the possibility to combine the two modular fixtures a natural progression would be to look into different ways of increasing the span of geometries that the modular range of fixtures are able to test. There are plenty ideas to develop regarding how to design a greater range of modular fixtures and inserts to make these systems even more adaptable. For example a set of spacers could help make sure to fit geometries that does not match the distance between the grooves seen in Appendix A today. It is also a possibility to try to connect several modular fixtures on top of one another (somewhat like a layer cake). That would yield the benefit of being able to test several devices at the same time. Preferably each in a separate vibration axes to reduce effects of resonance. Being successful in this undertaking could heavily reduce test time.

Regarding the guidelines for fixture making the implementation of the suggested work flow would most probably lead to both new difficulties and new possibilities which would both be of interest to study.
Bibliography


Appendix A. Drawing of the modular fixture

All tolerances according to ISO 2768-M.

Section view B-B
Scale: 1:1.
Valid for all holes.

Detail A
Scale: 2:1
Valid for all six grooves.