Development of a Bending Fatigue Testing Method for Rings from a CVT-Pushbelt

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PREFACE

This thesis contains the research that is done for the development of a bending fatigue testing method for rings from a CVT-pushbelt. It is the conclusion of the master education Mechanical Engineering at Delft University of Technology.

The thesis starts with the paper that is written as a result of this research and is followed by the appendixes. The appendixes mainly contain smaller researches that are done to support and to elaborate on the global research discussed in the paper. At the end of this thesis a literature study can be found that was carried out earlier which forms a basis for this thesis.

The research is done in collaboration with Bosch Transmission Technology B.V. formally known as Van Doorne Transmissie in Tilburg, The Netherlands. This company is the inventor and manufacturer of the pushbelt that is used for continuously variable transmission in cars.

I would like to thank Duc Tran and Jan-Peter Elkhuizen of Bosch Transmission Technology for their supervision and for giving me the opportunity to do this research at the company. Also I would like to thank test operator Jean-Pierre van Eijndhoven for helping me with the practical side of this research. From Delft University of Technology I would like to thank Just Herder for the guidance and feedback I received during this graduation project.

L.J. van Leeuwen

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# Table of Contents

Paper: Development of a Bending Fatigue Testing Method with Application on Rings From a CVT-Pushbelt ................................................................. 1

Appendixes .......................................................................................................................... 9

- A. Proof of Concept ........................................................................................................ 11
- B. Proof of Concept in Simulink .................................................................................. 15
- C. Construction Options ................................................................................................. 19
- D. Testing of Concept Model ......................................................................................... 25
- E. Concept Model in Simulink ....................................................................................... 31
- F. Current Fatigue Test .................................................................................................. 37
- G. Concept Model Fatigue Test until Failure ................................................................. 39
- H. Test Variations .......................................................................................................... 43
- I. Bose ElectroForce Machine Specifications ............................................................... 47
- J. Design Proposal for Bosch Transmission Technology B.V. ..................................... 49

Literature study:

- A Comparative Study on Cycle Fatigue Testing Methods and Machines ..................... 57

Appendixes for Literature study ......................................................................................... 66
Development of a Bending Fatigue Testing Method with Application on Rings from a CVT-Pushbelt

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Abstract—Fatigue tests are a very time consuming process in engineering, in which also a lot of energy is required. Speeding up fatigue tests and making them less energy consuming, would lead to major profits. The aim of this study is to obtain these profits by the development of a new bending fatigue test. In addition, the objective is to apply this new method to the bending fatigue test of rings from a CVT-Pushbelt.

A new bending fatigue testing method is developed and is successfully applied to the bending fatigue test of the CVT-pushbelt rings. The new method makes use of spring capabilities of the ring for displacement amplification via resonance. A concept model is created with which tests are done to validate the new method. It is concluded that the speed is increased by almost 60% and the energy requirement is reduced by 95%. Tests until specimen failure show similar results as current tests, while the conditions of the fatigue test remain the same.

I. INTRODUCTION

Fatigue testing is a widely used test in engineering to find out if materials or constructions hold after a certain amount of cyclic loads. During a fatigue test, forces are cyclically applied to the material to introduce stresses in the material in order to find the number at which it fails or to study the crack growth. The constant need of making products smaller and lighter, while it still needs to withstand the same circumstances, makes that fatigue tests are more commonly executed nowadays.

The problem with fatigue testing is that these tests take often a very long time to finish. Especially when tests are executed that will load the specimen with a high number of load cycles to the Very High Cycle Fatigue regime (VHCF >10⁷ cycles). Since the existence of a never ending horizontal asymptote in a Stress vs. Number of cycles to failure Curve (S-N curve) is seriously doubted, more fatigue tests are done which reach the VHCF regime [1].

In addition, these tests most often require a large amount of energy to be executed. The machines used for the fatigue tests are generally high powered machines. Since the fatigue tests are wanted to be executed at a maximum rate, the machines are pushed to their limits to load the specimen at the maximal realizable frequency. And with these machines operating day in day out for months, at maximal power, it speaks for itself that fatigue tests are rather energy consuming.

An example of such a fatigue test is the bending fatigue test on rings from a pushbelt of a Continuously Variable Transmission (CVT) for a car. The CVT system with the pushbelt works with two pulleys over which a V-shaped belt runs, that allows changing ratio without steps. The CVT-pushbelt is the V-shaped belt consisting of a couple hundred metal elements and around twenty metal rings in two packs that hold the elements together. This belt transmits the torques over the pulleys by pushing forces via the elements rather than pulling. The configuration of the pushbelt in a CVT and its structure can be seen in Fig. 1 [2]

During the use of a CVT-pushbelt in a car, it is loaded with several forces. [3] One of the load cases that this belt has to withstand is bending of the rings when the belt runs over the pulleys. Bending fatigue tests are done to find the fatigue behaviour of the ring under these bending stresses. By help of
this test, adjustments to the ring can be proved on their lifetime under the cyclic bending stresses. This fatigue test is one of the earlier mentioned tests that take a long time to finish and require a large amount of energy. The high number of cyclic loads car components have to withstand [4], make the fatigue test a long lasting process. In order to reduce the testing time, the specimens of ring material are bent at the maximal achievable frequency. However, this is limited by the power of the machine. The fixed stroke it has to make for the required bending stresses together with the maximal power of the machine, result in a limited bending frequency, hence equation (1).

\[ \text{Power} = \text{mass} \cdot \text{acceleration} \cdot \text{velocity} \] (1)

This brings us to the objective of this study. The objective is twofold: first this research is aimed at developing a new method for bending fatigue testing for thin plates at low energy consumption and high frequency. In addition, the goal is to apply the new method to the bending fatigue test for rings from a CVT-pushbelt.

In the next section the requirements for the fatigue test are described. In the section about the conceptual design, the design options are discussed and decisions are made for a final concept. After that, it is described how a concept model is created and how test are done on this concept model. The results are discussed in section V after which the research is concluded.

II. DESIGN CRITERIA

Towards the design of a high frequency fatigue test for thin plates, some design criteria have been set up. These criteria can be considered as measures to determine if the new bending fatigue testing method is an improvement to current methods. To do so, a comparison will be made between the current bending fatigue test on the ring and the new bending fatigue test on the ring after application of the new method.

The frequency at which the test specimen is bent is an important criterion. It should be at least higher than 60 Hz, which is the current rate at which the specimen is bent. It is preferred to bring the bending frequency to a maximum, since testing times will then be reduced the most.

Another criteria on which the new fatigue testing methods will be judged, is the energy it requires to operate the test. At this moment, the testing machine requires 750W of electrical power to operate the bending fatigue test of the ring. The new fatigue testing method should reduce the energy consumption to a minimum.

The third criterion has to do with the stresses that are introduced in the material by the fatigue test. It is required that the stresses are concentrated at one point. If the same stresses are put over the whole length of a ring, one does not know where the ring will fail. When the stresses are concentrated at one point it is not only known where the ring will fail, but it also makes it possible to study the crack growth without searching for the crack.

Next to that, the stresses introduced by the fatigue test should be adjustable. In order to construct a Wöhler-curve (S-N curve), the variety of stresses is needed to determine the fatigue life at different effective stresses. The desired stresses range from 350 MPa to 600 MPa, which are typical stresses to reach the VHCF regime.

III. CONCEPTUAL DESIGN

The objective of developing a new improved fatigue testing methods by itself is a vague description. This leaves a lot of options open, which means that it is wise to get some insight in all possible options. Roughly a separation can be made between the steps that lead into the actual bending stresses in the test specimen. These steps function as a subdivision for possible options. The steps are:

- Actuation
- Movement Translation
- Bending Stress Introduction
In this section, the steps are discussed and the options are given. A decision is made between the options which are thought to lead in the most promising concept.

A. Actuation

There is a large number of options that can make repetitive motion for a fatigue test [5]. The options differ in frequency, stroke and force range of the movement that can be made. From this study it appears that actuating can be done by hydraulic, magnetic or piezo-electric actuation.

However, not all the options are suitable for the bending fatigue test which is discussed in this study. The fact that the test specimen is a flexible piece of steel which needs small forces and large displacements to be bent in the desired way, already limits the decision. For the design of the new bending fatigue for the CVT-pushbelt ring application, there is chosen to use the Bose ElectroForce 3330. During testing of new designs of the bending fatigue test, it comes handy that this machine is flexible in his operation characteristics. The frequency and stroke are freely adjustable within the limitations of the machine. This machine is also used for the current bending fatigue test of the ring.

B. Movement translation

Making the translation from the movement of the actuator to a displacement of the specimen in order to introduce the stresses in the material can be done in several ways. The most common technique for this is by a 1 to 1 application of the displacement of the actuator to the specimen, either directly or by help of some kind of grip to introduce a specific kind of stresses. [5]

Another way of introducing the stresses to the specimen can be done by help of resonance. With this kind of translation of the displacement the specimen can be seen as a spring. Together with a mass (i.e. additional mass or own mass), the system forms a mass spring systems which can resonate in its natural frequency. In this way the movement of the actuator can be amplified and larger stresses are introduced as there would have been by a 1 to 1 translation.

When using resonance for fatigue testing, one is dependent on the natural frequency of the system. The system will only be able to apply the desired loads in this frequency. This frequency can be adjusted by reducing the mass or increase the stiffness. It is obvious that this results in a maximum frequency in which the mass cannot be decreased anymore and the stiffness of the specimen cannot be increased anymore. However, if an additional spring is used to increase the stiffness, higher frequencies are still possible.

Another advantage of using an additional spring is that one is able to apply a pre-stress on the test specimen. The spring introduces a load on the specimen which results in stresses in the material, so a new initial state of the specimen is obtained. The resonant test will now stress the material around this new equilibrium.

This brings us to the new actuation concept that is used in this study to translate the motion of the actuator to motion in the specimen. The actuator makes a small displacement which is amplified through resonance to get a larger displacement of the specimen. A spring element is used to increase the stiffness of the systems and with it the resonance frequency. Next to that, the spring can be used for the actuation as well as for putting a pre-stress on the specimen. For the sake of simplicity, this concept is called Spring-Amplified-Resonance (SAR) in the remainder of the report. A schematic drawing of this principle can be found in Fig. 2.

However, this principle of SAR fatigue testing is not applicable for every fatigue test. In fact it is limited to fatigue tests that require a large displacement and do not require high forces. Tests that require small displacements and high forces can better be tested by a 1 to 1 application of the excitation movement on the specimen. Next to that, a required high force would damp a resonant system too much. For this Spring-Amplified-Resonance test, a low damping coefficient is crucial.

1) Proof of Concept

In order to test this concept on its feasibility, a model was created using a small piece of the ring, a small spring, and a loudspeaker as can be seen in Fig. 3. This model works as follows: the piece of the ring that is to be bent is fixed connected at one side and attached to the spring (with mass in the form of bolt and nut) at the other side. In turn, at the other side the spring is connected to the coil of the loudspeaker. The loudspeaker functions as the actuator of the system.

When a function generator is used to put a monotonic electric signal at the natural frequency of the system on the
loudspeaker, the system will resonate. The excitation of the voice coil is amplified by the spring and then applied to the ring piece. The ring is now bent at the natural frequency of the system with the amplified excitation of seven times the voice coil excitation. More about this proof of principle can be found in appendix A.

With this model it is proven that the SAR principle can be useful for fatigue testing. A larger displacement can be applied to the specimen at a higher frequency than there would have been when making use of 1 to 1 application of the movement.

C. Bending Stress Introduction

The bending principles are the ways in which a specimen can be bent. How the bending stresses are introduced into the material is different among the principles. In this section, an overview is given of the options and how they can be adjusted for a fatigue bending test.

1) Bending Principles

In general, six ways of introducing bending stresses into a specimen can be distinguished. These bending principles are shown in Fig. 5. Not all of them are relevant for the fatigue bending test this report is aiming at. The bending principles that are relevant for the bending fatigue test of a thin plate are the cantilever bending, two point bending, three point bending and by applying a moment. Four point bending works in this respect similar to three point bending. The principles of how the forces are executed to the specimen are the same, only slight changes have to be made to the fixations. The belt-rolling bending is not taken into consideration since this principle does not meet the requirement of introducing a concentrated stress into the ring.

2) Design Options

The four remaining bending principles can all be adjusted in such a way that they form a Spring-Amplified-Resonance system. In fact, they can be adjusted in several ways, while they still form a SAR system. For instance, compression springs can be used as well as tension springs, but also another specimen can function as the spring. Actuation can be done by applying the displacement to the spring as well as applying it to the specimen itself. The construction options combined with the bending principles give a wide variety of design possibilities for the SAR fatigue tests. An overview of the options is shown in appendix C.

3) Concept for Bending Fatigue Test

The concept that is chosen for further investigation is the symmetric two point bending concept combined with the specimen-specimen construction option. The schematic drawing of this concept can be seen on the left side of Fig. 4.

Although there is no spring anymore in this concept, it still makes use of the SAR oscillation. Only in this concept the spring is replaced by another specimen. This gives the advantage of testing two specimens at the same time. Together with the symmetry that is used in this bending principle this means that bending stresses are introduced at four places at the same time during a test.

Another advantage is that this concept is relatively simple. Due to the stability that is obtained by the symmetry and by the use of another specimen instead of a spring it is not needed to get rid of unwanted degrees of freedom. The only degree of freedom of the mass between the specimen that is left is the up and down motion. This gives the wanted motion in order to bend the specimen in the right way.

The biggest advantage in fact is that the highest stresses in the material can be found in a place that is free of clamps and constraints. This is a result of using the two-point bending as the bending principle. The highest bending stresses can be found on the places where the bending radius is the smallest. And as can be seen in Fig. 7 this is right between the mass and punch. This results in clean stresses without local peak stresses at the contact points. These peak stresses will only lead into deviating fatigue behaviour, which is not relevant for the fatigue behaviour of the ring in a CVT. In order to introduce the stresses which are more comparable to the bending stresses in a CVT-Pushbelt ring, this clean bending is preferable.
IV. CONCEPT MODEL

A model of the chosen concept for the fatigue test has been created. This model can be seen at the right side of Fig. 4. The model is constructed out of two pieces of ring with the same dimensions (130x12.5x0.2 mm). The pieces are bent in to two circles to apply the pretension and are fixed in the middle with a connection point that also functions as the mass. It is this mass that makes the oscillating movement that is needed to introduce the stresses for the fatigue test.

A. Testing the Concept

The concept model is tested in order to find the frequencies it will resonate at and to find out if the amplified displacement is sufficient for the fatigue test. For this test the mass of the connection point is varied, which should lead into two different resonance frequencies. For both weights, a variety of tests are executed in which the input frequency as well as the excitation displacement is varied. In this way it was possible to find the resonance frequencies of the system.

The test rig for this test can be seen in Fig. 6. It consists of the Bose ElectroForce machine for the actuation and a high-speed camera for tracking the behaviour of the system. The high-speed camera makes it possible to have a closer look afterwards on the forming of the specimen and it allows determining the displacement of the mass. This displacement and the deformation of the specimen during a test can be seen in Fig. 7. This displacement determines the stresses in the material, which can be determined by FEM calculations if the form is consistent. The tests are described in more detail in appendix D.

1) Test Results

The results of the tests with the heavier mass can be found in table 1 and the tests with the lighter mass in table 2. Due to resolution limitations of the high speed camera, the resulting mass displacement could only be determined at around 0.3 mm accuracy.

<table>
<thead>
<tr>
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<tbody>
<tr>
<td>67</td>
<td>2</td>
<td>5.3</td>
<td>2.6</td>
</tr>
<tr>
<td>68</td>
<td>2</td>
<td>5.3</td>
<td>2.6</td>
</tr>
<tr>
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<td>3.6</td>
<td>3.6</td>
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<tr>
<td>74</td>
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<td>3.9</td>
<td>65.8</td>
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<tr>
<td>74</td>
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<td>4.9</td>
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<tr>
<td>74</td>
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<td>9.5</td>
<td>9.5</td>
</tr>
<tr>
<td>75</td>
<td>1</td>
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</tr>
<tr>
<td>79*</td>
<td>0.2</td>
<td>16.8</td>
<td>83.8</td>
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* Test with non-uniform mass displacement

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<tbody>
<tr>
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<td>0.06</td>
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<td>104.2</td>
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<td>0.1</td>
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<td>1</td>
<td>14.1</td>
<td>14.1</td>
</tr>
<tr>
<td>95.2</td>
<td>0.1</td>
<td>11.2</td>
<td>111.8</td>
</tr>
</tbody>
</table>

B. Concept Model in Simulink

In order to get a better understanding of the behaviour of the concept model a Matlab Simulink model has been made. This model is shown in Fig. 8 where the different components of the concept model can be distinguished. On the outer left side, the excitation displacement is applied. The forces that are executed on the mass by the specimen are calculated separately in the function blocks and divided by the effective
mass to get the acceleration of the mass. By integrating the acceleration, the position of the mass is calculated.

The forces executed by the specimen are deduced from earlier studies to two-point bending [6][7]. The resulting function is given in equation (2). In this function, $E$ is the Young’s Modulus of the material and $I$ is the second moment of area. $D$ stands for the distance between the specimen ends.

$$F = 0.847 \cdot E1 \cdot \left(\frac{2}{D}\right)^2$$

From this function also the stiffness can be deduced. The results are shown in Fig. 9. Equation (2) gives the forces executed by only one side of the circle of the concept model like it is shown in Fig. 2b. That is why it has to be multiplied by two for implementation in the function blocks of the Simulink model.

The input for this Simulink model is the excitation displacement, which can be varied in amplitude and frequency. In order to investigate the behaviour of the system the input amplitude is varied from 0.1 mm to 0.2 mm and the frequency is varied from 73 Hz to 84 Hz. Simulations are done with increasing frequency as well as constant frequency. The results are shown in Fig. 10.

**C. Test until Specimen Failure**

In order to find out if this concept model is suitable for the actual fatigue test, tests are done until failure of the specimen. For these tests, five test pieces where manufactured which were loaded via this resonance amplification until the first crack.

For setting a reference also tests are done with the current bending fatigue test. This way of fatigue testing also uses two-point bending to introduce the stresses at a concentrated point in the material, only the movement of the specimen is generated by 1 to 1 application of the movement of the actuator. To make the test comparable, the same displacements are applied to the material; both the distance for pretensioning the specimen as well as the distance travelled by the mass for cyclic stress introduction.

During both test methods the electrical current through the actuation machine is measured. Multiplication of this value by the voltage over the machine (230 V) allowed determining the electrical power requirement of the machine for operation of the fatigue test.

---

**Table 3**

<table>
<thead>
<tr>
<th>New Fatigue Testing Method</th>
<th>Current Fatigue Test</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>95 Hz</strong></td>
<td><strong>60 Hz</strong></td>
</tr>
<tr>
<td>14,531,146</td>
<td>31,771,409</td>
</tr>
<tr>
<td>59,373,463</td>
<td>1,548,892</td>
</tr>
<tr>
<td>18,873,903*</td>
<td>95,458</td>
</tr>
<tr>
<td>70,145</td>
<td>1,172,608</td>
</tr>
<tr>
<td>5,143,879*</td>
<td>41,500,000*</td>
</tr>
<tr>
<td>8,912,593*</td>
<td>85,755,021</td>
</tr>
<tr>
<td>9,358,772*</td>
<td>93,158,772*</td>
</tr>
</tbody>
</table>

* Test stopped manually, no failure yet.

**1) Test Results**

In table 3, the results are shown of these tests. The table gives the number of cycles at which the specimen failed during the current and new test method. In order to save time, the lighter weight is used, so the specimens are bent at a frequency of approximately 95 Hz.

The power the machine required during the current fatigue test was measured to be 750 W. The new test method reduces the power to 60 W.

**V. DISCUSSION**

**A. SAR-Principle**

From the results in table 1 and 2 it can be said that the Spring-Amplified-Resonance principle works for this fatigue test. This concept of movement translation generates a mass displacement that is at maximum 122 times the movement of the actuator. With this mass displacement, the stresses are introduced into the material.

From the results it also appears that there is no spring needed for the SAR-Principle to work. With the concept model it is shown that if the spring is replaced by another specimen, the excitation displacement will also be amplified into a larger mass displacement.

**B. Frequency**

From the results in table 1 and 2, it appears that there are indeed two resonance frequencies found. The system with the heavier mass has its optimal resonating point around 77 Hz,
the system with the lighter mass around 95 Hz. Both of these frequencies are higher than the current 60 Hz.

1) Variable Resonance Frequency

It also appears that the resonance comes to a higher frequency when the excitation displacement is larger. An example can be found in the gray rows of Table 2. The system stopped resonating when exceeding 92.6 Hz at an excitation displacement of 0.06 mm, while at an excitation displacement of 0.1 mm or 1 mm it still resonates at 95 Hz.

This behaviour is the result of the rings that act like non-linear progressive springs under two-point bending. This behaviour can be seen in Fig. 9, where the force-distance diagram is shown for the ring under two-point bending. From earlier studies, it appeared that the force that is needed to bend the ring under two-point bending is quadratic to the distance travelled. [6][7]

This progressive spring behaviour is the reason why the stiffness of the rings is constantly changing when the rings are bent. So when the mass makes a larger displacement, the rings are bent further and thus the forces the ring acts on the mass are relatively larger. This means a larger effective stiffness of the ring as well. And from equation (3) it can be obtained that this results in a higher resonance frequency on its turn. On the other hand, a smaller mass displacement as a result of a smaller excitation displacement will lead to a lower resonance frequency.

\[
f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \tag{3}\]

It is also for this reason that the excitation displacement should be applied to the model at an increasing frequency. If for example the excitation displacement is immediately put on the concept model at the resonance frequency of 77 or 95 Hz, the system will not move. The natural frequency around the stall position is much lower, so the frequency should be build up from the ‘stall natural frequency’ to the resonance frequencies of 77 or 95 Hz.

2) Simulink validation

The behaviour that is described and the theory that is used to explain the behaviour are validated by use of a Simulink model. This model is shown in Fig. 8.

Like in the real model, it appears from this Simulink model that the resonance frequency is indeed dependent on the excitation displacement that is applied to the model. At an excitation displacement of 0.2 mm, it is possible for the model to reach 84 Hz with an amplification of 133. However, at an excitation displacement of 0.1 mm it only reaches 75.6 Hz and an amplification of 119. See Fig. 10 a, b and c.

Also the start-up behaviour of the concept model is validated using the Simulink model. The excitation displacement should be applied to the concept model at an increasing frequency from the stall natural frequency to the frequency that gives the highest mass displacement. If the starting frequency is too high, the system will not resonate, so the mass displacement stays small, which can be seen in Fig. 10 d. The Simulink simulations are described in more detail in Appendix D.

With this model, it is also possible to find the theoretical frequency of the system of infinite small motion around the equilibrium point, the ‘stall natural frequency’ as it is called earlier. The frequency found is 58 Hz for the heavier weight and 74 Hz for the lighter weight. This value is a lot smaller than the frequency found at the larger mass displacements. This shows that the resonance frequency is highly dependent on the displacement the mass makes, and therefore also on the excitation displacement.

C. Stresses

Now the displacements of the mass are known the stresses in the material can be calculated. From the results it appears that the displacement can vary from 0 mm to 14.7 mm at tests that give stable displacements. This means that the effective stress introduced in the material can vary from approximately 290 MPa to 700 MPa. The desired 350 to 600 MPa from the design requirements falls within the reach of this fatigue test.
The two-point bending principle that is used for this concept also gives a concentrated maximum stress. The highest bending stresses are introduced in the material at the places where the ring makes the smallest radius. As a result of two-point bending the places in this concept model are right in middle between the mass and the connection points with the actuation machine. This gives the clean concentrated stresses in the material that are wanted for the fatigue test.

D. Test to failure

The results in Table 3 show that the number of cycles to failure is comparable to the current test. Until now, only five tests using the new method are done, which is a small number to draw a conclusion from. However, these five tests already show similar results as the current fatigue test.

Table 3 shows relatively more lower number of cycles to failure results for the new fatigue testing method than for the current fatigue test. This is due to the fact that the new method stresses the material at four points, where the current fatigue test has only one stress concentration. The test is stopped when the first crack in the surface is noticed. These cracks mostly appear at weak spots at the surface. The chance that a test at the new method stresses the material at a weak spot, is four times higher than when it is tested at the current fatigue test. The results from the current fatigue test show that a low number of cycles to failure occurs every once in a while.

Crack initiation after fatiguing the material with the new testing method is located in the middle between clamps and mass. This means that the stresses at the connection points with mass and clamps do not give peak stresses that will cause the material to fail. It can be said that the rings were purely failing due to the bending stresses introduced by this test.

E. Comparison with current fatigue test

As it is already mentioned, the frequency is raised around 35 Hz with respect to the 60 Hz of the current fatigue test. For this study this is the maximum value that was reached since introduction of SAR for the bending fatigue test. However, it is believed that the frequency can be increased even more if another bending principle is used. See appendix C.

The power reduction from 750 W to 60 W is already a large step forward in terms of efficiency. Taking the reduction of operating time also into account, the electrical energy used for one fatigue test is reduced by 95%.

The stresses that can be introduced into the material are the same. Effective stresses of 290 to 700 MPa are possible using the new fatigue testing method. If one wants to construct a Wöhler-curve (S-N curve), this variety is needed in order to determine the fatigue life at different effective stresses.

VI. CONCLUSION

A new method is developed for bending fatigue. This new method uses resonance in order to amplify the movement of the actuator. It is the first fatigue test which makes use of the Spring-Amplified-Resonance principle to increase the motion of a specimen for stress introduction.

A concept model of this new method has been constructed for tests on rings from a CVT-pushbelt. Tests on this model showed that the resonance amplification principle works for this bending fatigue test. It was found that the specimen in the concept model act as non-linear progressive springs under two-point bending. This behaviour causes the dependency of resonance frequency on the excitation displacement, which is validated using a Matlab Simulink model.

Overall it can be concluded that the speed of the fatigue test is increased by 50% with respect to the current bending fatigue test on rings from the CVT-pushbelt, while the energy requirement of a test is reduced by 95%. Actual fatigue tests until failure of the specimen showed similar results, when testing conditions were kept the same.

ACKNOWLEDGMENT

The author wishes to acknowledge Bosch Transmission Technology B.V. for supporting this research and providing the testing facilities.

REFERENCES

APPENDICES
APPENDIX A

PROOF OF CONCEPT

1) Introduction

In an early stage of this research the principle of Spring-Amplified-Resonance (SAR) fatigue testing had to be proven. For this reason, a small model was created in order to find out if a small displacement could be amplified by a spring through resonance. The amplified displacement could then be used for fatiguing a specimen. This appendix describes this proof of concept and how it is used for further development of this concept.

2) Method

A. Design Choices

For the sake of simplicity the bending principle that is used for this proof of concept is cantilever bending of which a schematic drawing can be found in Fig A.1. This is the simplest way a specimen can be bent and therefore allowed that the model would be easy to build and easy to actuate. The actuation of the system is done by a loudspeaker that was removed from a radio/CD-player. A spring is at one end connected to the voice coil of the loudspeaker and the other end is connected to the specimen via a bolt. This is done with a small pre-tension in order to prove that the system works with pretension. The bolt was not only used for the connection, but it also functions as a mass which improves the resonating capabilities of the system. The other end of the specimen is clamped to the stationary part of the loudspeaker, in order to obtain the fixed connection needed for cantilever bending. The actual model can be seen in Fig A.2
B. Controlling

The system control is done by Matlab. The loudspeaker is connected to the line-out of a computer via a mini-jack connection. A Matlab script is written to generate a sinusoidal signal with a constant frequency, which is sent to the line-out of the computer. If the frequency of the signal is the natural frequency of the system, the mass will then resonate. At other frequencies there will be just the monotonic sound of the loudspeaker, the mass will not move.

Since the system is very dependent on the actuation frequency, it is important that the right frequency is found. Determination of this frequency is done in the opposite direction as the actuation of the system. The loudspeaker mini-jack is connected to the microphone input of the computer and the system is given a push so it vibrates in its natural frequency. Via a Matlab script the signal can be read and the frequency can be determined using the Fourier transformation function in Matlab. The frequency that is found for the proof of concept is 94.5 Hz, which will then be the frequency of the sinusoidal signal that will be generated by the earlier mentioned Matlab script.

C. Measurements

In order to determine if the motion of the mass is really amplified with respect to the motion of the voice coil, measurements are done to the system. These measurements should verify quantitatively what already can be seen.

The setup that is used for these measurements can be seen in Fig A.4. The sensor that is used for measuring the displacement of the mass as well as the displacement of voice coil is *ILD 1401-20 Laser displacement-sensor 30mm to 50mm*. The data is logged at 1000 Hz with Labview software via a USB-data acquisition station of National Instruments.

Next to the displacement measurements, also the power of the signal that is used for actuating the system is measured. Two multimeters are used to measure the voltage over and current through the system, so the power can be calculated.
3) Results

A. Observations

When the system is actuated it can be seen that the mass makes a peak to peak displacement of around 1 mm. The displacement of the voice coil is too small to be perceived by the human eye. Along the spring it becomes clear that the amplification becomes larger for every point closer to the mass.

When the input signal frequency is set 1 Hz lower or higher the system still resonates, however, clearly with a smaller amplification. If the input signal frequency differs more than 4 Hz from the natural frequency the system does not resonate anymore.

B. Measurements

The measured displacements can be seen in Fig A.3. It was measured that the peak to peak displacement of the mass is 1.01 mm and the p-p displacement of the voice coil is 0.15 mm. This means that the system reaches an amplification of 6.95.

The measured voltage over the system was 0.149 V and the current through the system 33.4 mA. This means that the power that the system requires to operate reaches a value no higher than 5 mW.

4) Discussion

It can be obtained from these results that the concept of SAR fatigue testing is proven to work. The results show that the displacement from the input of the voice coil is amplified by this spring-mass-system so a larger displacement at the end of the specimen is obtained. This is the phenomenon that was wanted to be proven by this model.

However, when looking at the results it can be seen in Fig A.3 that the peak displacements of the mass appear to make a waving motion according to the measurements. This waving effect is due to the sampling rate of 1000 Hz. The sensor
cannot track every peak the mass makes, so that is why the waving effect occurs in
the results.

It can also be seen that the system needs some time to start-up. About half a
second after the input signal is started the mass reaches its maximum peak-to-peak
displacement. During this time, with every cycle the voice coil makes, it amplifies the
displacement a bit more. In the same way the mass movement will not be stopped
immediately after the input signal is stopped. The movement eventually damps out in
time. This is all because the energy in the system (potential and kinetic energy) has to
be built up at the start and damped out at the end of the voice coil excitation.

5) Conclusion

With the model that is created as a proof of concept it is proven that the concept of
SAR fatigue testing works. This proof of concept shows that a small displacement can
be amplified via a resonating spring, in order to get a larger displacement that can be
used for fatiguing a specimen. The amplified motion obtained by the model is almost
7 times as large as the input motion. This is achieved very efficiently since the power
requirement of this model is only 5 mW.
APPENDIX B

PROOF OF CONCEPT IN SIMULINK

1) Introduction
In order to get a better understanding of how the system works a Simulink model is created. Together with the parameters of the concept model and the measurements that are done, it is possible to make an accurate simulation of the system behaviour.

2) Method
There is started with a simplified schematic drawing of the system. This can be seen in Fig. B.2. The input of the system is the displacement the voice coil applies on the spring, the output the displacement of the mass. Both the spring and the specimen are modelled as a spring, only the specimen has a damping effect as well. It is assumed that the damping of the spring is negligible compared to the damping of the specimen. The other end of the specimen is connected to the fixed world.

Putting this into a Simulink model it would look like the model of Fig. B.1 At the summation block the forces on the mass come together. The resulting force divided by the mass gives the acceleration of the mass, with which the speed and the displacement can be calculated by integrating. With the displacement on its turn, the forces of the springs on the mass can be calculated. The damping of the system depends on the speed of the mass.

A. Parameter Determination
The values for the parameters in this model are: already known, can be calculated or can be estimated. The first parameter to be determined is the stiffness of the specimen. This value can be obtained by the use of Classical beam theory. The dimensions of the specimen are known and so are the material properties. By help of equation B.1 a value for the stiffness of 0.36 N/mm is found.

\[
k = \frac{3EI}{L^3}
\]

(B.1)
Now this value is known, also the stiffness of the spring can be determined. Since the value was not known beforehand, graphical analysis from Fig. B.3 allowed determining an approximate value for the stiffness of the spring. In these pictures the relation between the stiffness of the specimen and the spring can be found if the deformation of both is measured at different pre-loads. From this analysis it appears that the spring stiffness has a value of 0.48 N/m. The mass of the bolt and nut of the connection point was found to be 1.9 gr.

These parameters together also allow finding the frequency analytically. By use of equation B.2 it was found that the frequency for the Simulink Model would be 94.2 Hz.

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad \text{(B.2)}$$

In this equation the stiffness of the spring and the stiffness of the specimen can be added up to get the stiffness of the system. The reason for this, is that the spring elements can be considered to work parallel instead of in series. It is the displacement of the mass that matters for this stiffness and not the input displacement [J.C. Cool; Werktuigkundige Systemen]

The only parameter that is still unknown is the damping of the specimen. To determine this value the measurements on the real system will be used. It is measured that the input displacement is 0.15 mm and the mass displacement is 1.01 mm, so the
damping constant can be adjusted, in such a way that the resulting amplification is the same. The value for the damping is found to be 0.04, which is a plausible value for the damping.

3) Results
With the parameters for this Simulink model estimated, the model can be run. The results are shown in Fig. B.4 The maximum amplification of the system during the steady operation is 7.

4) Discussion
The resulting figure from the simulation is similar to the figure that is constructed from the measurements. The amplification the system reaches is the same and almost the same frequency is found.

However, there is a small difference. At the results from the simulation it appears that the start-up and damping behaviour slightly differ. In the Simulink simulation these processes go faster. The faster response of the mass displacement is probably because the damping is not accurate enough. The damping of the real system is most likely not linear (i.e. the damping is more than proportional higher at larger displacements), where in the Simulink model it is modelled as a linear damping.

Another reason for this difference can be found in actuation of the system. Where the voice coil displacement in the Simulink model is assumed to have no time needed to start up, there can be seen in the measurements on the real system that it does need a certain time before it reaches the maximal input displacement. The larger displacement of the mass also influences the displacement of the voice coil. This means that at the beginning the forces that the spring puts on the mass are in the real system not as high as they are in the simulation model. This causes a slower start and damping of the real system.

The frequency that is found by the analytical way in this simulation is almost the same as the frequency measured on the real system. The minor difference (94.2 vs. 94.5) can be considered negligible.
5) Conclusion

The behaviour of the Simulink Model can be considered to be comparable with the behaviour of the proof of concept. Despite of the differences in start and damping behaviour, the steady-state behaviour is the same. And since during a fatigue test the system would only operate at this steady-state, it makes that this model is valuable for further evaluation of the system and the Spring-Amplified-Resonance principle.
1) Introduction

For the actual introduction of bending stresses into the ring material there are a wide variety of options. First there are the options of the bending principles and second the options of how to combine the bending principle with the Spring-Amplified-Resonance fatigue testing method. In this appendix there is looked at the options that exist and there is made a comparison between the design options.

2) Method

A. Bending Principle Options

Introducing bending stresses into a material can be done in six ways. These bending principles are shown in Fig. C.1

Fig. C.1: Six bending principles with (a) cantilever bending, (b) two-point bending, (c) three-point bending, (d) four-point bending, (e) applied moment and (f) rolling bending.
Not all of them can be used for the fatigue bending test this report is aiming at. The bending principles that can be used for the bending fatigue test of a thin plate are the cantilever bending, two point bending, three point bending and by applying a moment. Four-point bending works in this respect similar to three-point bending. The principles of how the forces are executed to the specimen are the same, only slight changes have to be made to the fixations. The belt-rolling bending is not taken into consideration since this principle does not meet the requirement of introducing a concentrated stress into the ring.

B. Construction Options

The four remaining bending principles can all be adjusted in such a way that they form a spring-amplified-resonance system. In fact they can be adjusted in several ways, while they still form a spring-amplified-resonance system. For instance compression springs can be used as well as tension springs, but also another specimen can function as the spring. Actuation can be done by applying the displacement to the spring as well as applying it to the specimen itself. All these options give a wide variety of construction possibilities for the spring-amplified-resonance fatigue tests of which a schematic hierarchy tree structure can be found in Fig. C.2. The six open tree ends in the gray blocks are suitable options for the SAR fatigue testing design.

C. Comparison

The bending principles and construction options together give quite some design options. To get a better understanding of how the options hold against each other, a comparison is made. For each of the relevant bending principles an indication of the maximum achievable frequency will be determined.

3) Results

Now there are found several bending principles and several actuation principles these can be combined in a table to give an overview of the possible design options. This table is shown on the next page. Please note that for the two-point bending principle there are two options mentioned in this table. For this principle there is made a variation which makes the construction symmetric and therewith also more stable.
A table with all construction possibilities for a spring-amplified-resonance system can be found on the next page. The options that can be found in this table all have their own characteristics. The bending principle gives a certain stiffness of the specimen, the connection between the spring and specimen gives a certain mass, which together form a basis for the frequency the construction option can operate at. This frequency is considered to be one of the most important characteristics, so a value is estimated by help of equation (C.1) in order to compare the systems with each other.

\[
f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad \text{(C.1)}
\]

A. Frequency Comparison

To make the estimation some assumptions have to be made. The first estimation is that the maximum frequency is equal for all systems with the same bending principle. For all bending principles it holds that the stiffness is maximal at a minimum length of the specimen. However, due to construction restrictions there is a minimum length, which is estimated. With classical beam theory it is possible to determine an estimated stiffness for the specimen. The next assumption is that the stiffness of the spring is equal to the stiffness of the specimen. For similar values of the stiffness it is proven that the SAR principle works, for distinctive values this is not certain yet. The last estimation that is needed for finding the resonance frequency is the mass of the connection between specimen and spring. Since the frequency will be maximal at a minimal mass, it is estimated what the minimal mass would be for the corresponding bending principle.

With this information and by help of equation (C.1) it is possible to find an estimated value for the maximum frequency at which the systems can operate at. The results are shown in Table C.1.

Table C.1: Estimated frequencies for the bending principles.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Cantilever Bending</td>
<td>15</td>
<td>1</td>
<td>244</td>
</tr>
<tr>
<td>2. Two-Point Bending</td>
<td>30</td>
<td>2</td>
<td>138</td>
</tr>
<tr>
<td>2a. Two-Point Bending Ring</td>
<td>100</td>
<td>1</td>
<td>144</td>
</tr>
<tr>
<td>3. Three-Point Bending</td>
<td>25</td>
<td>4</td>
<td>245</td>
</tr>
<tr>
<td>4. Applied Moment</td>
<td>15</td>
<td>3</td>
<td>380</td>
</tr>
<tr>
<td></td>
<td>Compression Spring Spring actuation</td>
<td>Compression Spring Specimen actuation</td>
<td>Tension Spring Spring actuation</td>
</tr>
<tr>
<td>---</td>
<td>-------------------------------------</td>
<td>---------------------------------------</td>
<td>---------------------------------</td>
</tr>
<tr>
<td>1</td>
<td><img src="image1.png" alt="Image" /> Max. 244 Hz</td>
<td><img src="image2.png" alt="Image" /></td>
<td><img src="image3.png" alt="Image" /> Max.</td>
</tr>
<tr>
<td>2</td>
<td><img src="image6.png" alt="Image" /> Max. 138 Hz</td>
<td><img src="image7.png" alt="Image" /></td>
<td><img src="image8.png" alt="Image" /> Specimen actuation</td>
</tr>
<tr>
<td>2a</td>
<td><img src="image10.png" alt="Image" /> Max. 144 Hz</td>
<td><img src="image11.png" alt="Image" /> Specimen actuation</td>
<td><img src="image12.png" alt="Image" /> Specimen actuation</td>
</tr>
<tr>
<td>3</td>
<td><img src="image14.png" alt="Image" /> Max. 245 Hz</td>
<td><img src="image15.png" alt="Image" /> Specimen actuation</td>
<td><img src="image16.png" alt="Image" /> Specimen actuation</td>
</tr>
<tr>
<td>4</td>
<td><img src="image19.png" alt="Image" /> Max. 380 Hz</td>
<td><img src="image20.png" alt="Image" /> Specimen actuation</td>
<td><img src="image21.png" alt="Image" /> Specimen actuation</td>
</tr>
</tbody>
</table>
4) Discussion

As can be seen in the table on the previous page, there are many possible design options when using the SAR-principle for a bending fatigue test. However, it must be said that not all of the options are feasible. Some options do not leave enough space for actuation and others have degrees of freedom that have to be fixated. This makes some of the options less suitable for application in an actual bending fatigue test.

It should be taken into account that the estimated maximum frequencies for the bending principles as they are calculated in table C.1 are not accurate. The assumptions made for the comparison are too rough and highly negotiable. This means that the value for the maximum achievable frequency can only be considered as an indication. Actual maximum frequency for the bending principle can differ.

These two factors make it hard to draw a conclusion of which system best to use. Experiments should be carried out on real models to validate if the options will really work with the SAR-principle and what the maximum frequency would be.

5) Conclusion

An overview is presented of possible combinations of construction options and bending principles. For each bending principle and indication is given of the maximum frequency. Since the feasibility of the options differs a lot and the maximum frequency is only an indication, it is at this moment not possible to conclude which system is best to use for the bending fatigue test.
APPENDIX D

TESTING ON CONCEPT MODEL

1) Introduction

Of one of the design options from the overview table in appendix C, there is made a prototype for further investigation. The two-point bending principle combined with the specimen-specimen construction option is a promising concept for the bending fatigue test. It is easy to construct and it introduces the bending stresses by two-point bending, which means that there are concentrated bending stresses at clean spots with no peak stresses due to fixations. Next to that the stresses are introduced at four places in the test specimen, which can be interesting for crack growth studies. The constructed model can be seen in Fig. D. 1.

2) Method

This model is constructed out of two pieces of the ring of about 130 mm. The rings are 12.5 mm wide and 0.2 mm thick. They are bent in such a way that they form an ‘8’ together. The connection or the rings in the middle of the ‘8’ is done by two 10x10x1 mm plates which clamp the rings by help of an M4x5 hex socket head bolt. This can be seen in Fig. D. 1

The connection point also forms the mass that is used for resonating. In this model the two pieces of rings form the springs and the connection point forms the mass. For the test two different masses are used. The lighter mass construction of the two consists only of the above mentioned rings, plates and bolt. The heavier of the two also consist of a 30 mm long plate in between the rings and a heavier bolt. (M4x10 hex socket head). The corresponding masses are 3.54 g and 6.76 g. The variation in masses should lead to a difference in resonance frequencies of the models.

In order to test the models on their resonance behaviour, a small displacement is applied to one of the two rings. This displacement should be amplified, so the mass makes a much larger displacement.

The actuation is done by the Bose ElectroForce machine. This machine is flexible in its use and can generate a wide range of displacements at a wide range of
frequencies as long as the combination stays within the power limits of the machine. This flexibility and its ease to use make it a suitable machine for the test.

In order to follow the oscillating motion and to make sure that the model will stay on its place, it is fixed to the machine. This is done with two super magnets which stick the model at both connection points to the machine. Finally a high-speed camera is used to find out what happens to the rings during the tests. By help of the frames taken by the high-speed camera it is possible to track the displacement the mass makes. Next to that it also allows to check whether the rings make the right motion and if the mass is just making an up and down motion. The test rig can be seen in Fig. D. 2.

The tests are started by applying a constant amplitude at an increasing frequency. In this way the resonance frequency can easily be found by watching the movements of the mass. When the resonance frequency was found several movies with the high-speed camera were shot of the concept model reacting to frequencies around the natural frequency. This is done for multiple excitation displacements, so the differences in the behaviour at different input amplitudes could be studied as well. When all this was done for the first mass, the concept model was adjusted after which the same procedure is followed for the other mass.

A. Obtaining Results

The results that are wanted to be obtained from these tests are the displacement the mass makes and the amplification. The movement of the mass is determined by the frames of the videos that are shot by the high-speed camera. For each frame the height of the mass is determined by finding at which pixel of the frame the most top part of the bolt is. This gives the height of the bolt and by doing this for all the frames in the movie, the peak to peak displacement of the mass can be determined. Dividing this displacement by the excitation displacement gives the amplification of the system. Two frames of a video captured by the high-speed camera to determine the mass displacement can be seen in Fig. D. 3.
3) Results

All tests that are done with the concept model are listed in tables D.1 and D.2 together with the resulting values for the mass displacement and the amplification.

TABLE D.1
Results of the tests with the 6.76 g mass

<table>
<thead>
<tr>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td>67</td>
<td>2</td>
<td>5.3</td>
<td>2.6</td>
</tr>
<tr>
<td>68</td>
<td>2</td>
<td>5.3</td>
<td>2.6</td>
</tr>
<tr>
<td>69</td>
<td>1</td>
<td>3.6</td>
<td>3.6</td>
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<tr>
<td>74</td>
<td>0.06</td>
<td>3.9</td>
<td>65.8</td>
</tr>
<tr>
<td>74</td>
<td>0.1</td>
<td>4.9</td>
<td>49.3</td>
</tr>
<tr>
<td>74</td>
<td>1</td>
<td>9.5</td>
<td>9.5</td>
</tr>
<tr>
<td>75</td>
<td>1</td>
<td>11.5</td>
<td>11.5</td>
</tr>
<tr>
<td>77</td>
<td>0.1</td>
<td>12.2</td>
<td>121.7</td>
</tr>
<tr>
<td>77*</td>
<td>1</td>
<td>16.1</td>
<td>16.1</td>
</tr>
<tr>
<td>78</td>
<td>0.2</td>
<td>14.1</td>
<td>70.7</td>
</tr>
<tr>
<td>78*</td>
<td>1</td>
<td>17.4</td>
<td>17.4</td>
</tr>
<tr>
<td>79*</td>
<td>0.2</td>
<td>16.8</td>
<td>83.8</td>
</tr>
</tbody>
</table>

* indicates non-uniform mass displacement

Fig. D. 3: Two frames as captured by the high speed camera for determining the mass displacement.
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>92</td>
<td>0.02</td>
<td>1.6</td>
<td>82.2</td>
</tr>
<tr>
<td>92.6</td>
<td>0.06</td>
<td>6.3</td>
<td>104.2</td>
</tr>
<tr>
<td>94</td>
<td>0.1</td>
<td>9.2</td>
<td>92.1</td>
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<tr>
<td>95</td>
<td>1</td>
<td>14.1</td>
<td>14.1</td>
</tr>
<tr>
<td>95.2</td>
<td>0.1</td>
<td>11.2</td>
<td>111.8</td>
</tr>
</tbody>
</table>

4) **Discussion**

From these results it can be obtained that the natural frequency of the lighter mass can be found around 77 Hz, and for the heavier mass around 95 Hz. At these frequencies the specimen exhibits the largest amplification with respect to the excitation displacement.

During the test it was seen that the displacement of the mass damped out if the excitation frequency exceeds the natural frequency. However, if the excitation displacement was large, the frequency at which the mass displacement damped out was higher. For example at the heavier mass the systems damped out at 78 Hz if the excitation displacement was 0.1 mm, but for larger excitations it did not yet damp out. This means that the natural frequency increased if the displacements enlarge.

A reason for this behaviour can be found in the non-linear stiffness (progressive) the specimen exhibits when it is bent with two-point bending. Earlier research by Bosch Transmission Technology as well as others ([6], [7] from paper) shows that the stiffness of a thin plate under two-point bending is non-linear. Results from the research done by Bosch are shown in Fig. D.4. In this figure it can be seen that if the specimen is bent further, the force as well as the stiffness increase non-linear progressively.

Knowing the displacements of the specimen during the resonant fatigue tests, together with the results from the research to the stiffness under two-point bending, the stiffness of the specimen during these tests can be found. It appears that at a test with a large displacement of the mass the stiffness is fluctuating between

![Graph showing force and stiffness vs distance between specimen ends](image-url)
approximately 0.2 N/m and 0.7 N/m. This means that the higher the displacement of the mass, the higher the stiffness will be, which results in a higher natural frequency as well. Validation of the theory that the progressive spring behaviour is the cause of the frequency dependency on the excitation displacement is done by a Simulink model and described in Appendix E.

5) Conclusion

A concept model has been build which combines the two-point bending principle with the SAR principle. The amplified mass displacements that are found at different excitation displacements and frequencies show that the SAR-principle works. The excitation displacement was at maximum 121 times amplified into the mass displacement. With these findings it can be concluded that the combination of this bending principle together with the SAR-principle forms a very promising fatigue test for the bending of the ring from a CVT-pushbelt.
APPENDIX E

CONCEPT MODEL IN SIMULINK

1) Introduction
From the experiments on the concept model (Appendix D) it appeared that the resonating frequency is dependent on the excitation displacement. A theory is that the progressive force-displacement characteristic of the ring under two point bending makes that higher frequencies are in reach if the specimens are bent further. In order to validate this, a Simulink model has been created on which simulations will be done.

2) Method
A. The Simulink Model
The basics of the simulink model for the Concept Model are the same as for the simulink model of the proof of concept (Appendix B). However, determining the forces the specimens act on the mass is more complex now. Form earlier studies it appeared that the specimen act like non-linear progressive springs under two point bending ([6], [7] form paper). This non-linearity means that the forces the specimens act on the mass should be calculated separately. This can be seen in fig E.1, where the two function blocks each calculate the forces of the corresponding specimen on the mass. For the lower specimen that is fixed to the ground at one side, the only input is the displacement of the mass. On the other specimen also the excitation displacement is of influence and therefore this value is subtracted from the mass displacement in order to get the force the upper specimen is acting on the mass. The formula behind the function blocks of the simulink model can be found in equation (E.1). This is the result of the earlier research, of which a graph can be found in Fig. D.4.

\[ F = 2 \cdot 0.847 \cdot EI \cdot \left( \frac{2}{D} \right)^2 \]  
(E.1)
Now the forces on the mass are known, the acceleration of the mass can be calculated by dividing the sum of forces by the effective mass (of the specimen and lighter mass together). By integrating also the velocity and displacement for the mass are determined.

B. Simulations for Validation

From the results of the experiments described in Appendix D it can be gathered that with a higher excitation displacement, also higher resonance frequencies are obtained. The same experiment is also going to be executed by a Simulink simulation on the model that has been created. Simulations will be done with different excitation displacements in order to validate the theory that the progressive spring behaviour is the cause of this phenomenon. These experiments will be executed with input parameters that are the same as for the tests on the Concept Model.

First a simulation will be run to set a reference in which a standard test is simulated. The excitation displacement during the simulation will be applied at increasing frequency for the first ten seconds. During the last ten seconds the frequency be constant at the value it reached during the first period. This final frequency is the maximum frequency that can be reached at this excitation displacement. The second test will be the same as the first, only the excitation displacement will be decreased. During the third test the maximal frequency at this decreased excitation displacement will be found. Finally the start-up behaviour is investigated by a simulation where directly the excitation displacement is applied to the model at the final frequency of the first simulation.

3) Results

On the next pages the graphs that result from the simulations are shown. The two displacements look like two surfaces, but in fact the oscillations of the signal are so dense in time, that the separate oscillations cannot be distinguished anymore. The information that can be obtained from these figures is the peak to peak displacement of the mass and punch.
Fig. E.2: Graph of the displacements where the excitation displacement of 0.2 mm is put on the model at an increasing frequency form 73 to 84 Hz in the first 10 seconds, after which the reached conditions remain the same for the last 10 seconds. 84 Hz is the maximal frequency for this excitation displacement, any higher frequencies lead to damping out of the mass displacement. The reached amplification is 133 times the excitation displacement.

Fig. E.3: Graph of the displacements where the excitation displacement of 0.1 mm is put on the model at an increasing frequency form 73 to 84 Hz. It can be seen that the systems cannot follow the input frequency, so it damps out after only a few seconds.
Fig. E.4: Graph of the displacements where the excitation displacement of 0.1 mm is put on the model at an increasing frequency form 73 to 75.6 Hz. This is the highest frequency that can be reached by the simulation for this excitation displacement. The reached amplification is 119 times the excitation displacement.

Fig. E.5: Graph of the displacements where the excitation displacement of 0.2 mm is directly put on the model at 84 Hz. It can be seen that the system does not resonate at this frequency if the frequency is not build up.
4) Discussion

With these results it is found that the dependency of the resonance frequency with respect to the excitation displacement is indeed due to the nonlinear spring behaviour of the specimen. A larger excitation displacement will lead into a larger mass displacement, which results in a higher stiffness and therefore a higher possible resonance frequency.

In the results section this becomes apparent when the first three graphs are considered. If the excitation displacement is 0.2 mm a resonance frequency can be reached of 84 Hz, while at an excitation displacement of 0.1 mm, a resonance frequency of only 75.6 Hz can be reached. Any higher frequencies led to damping out of the system, as can be seen in Fig. E.3

To validate the start-up behaviour, Fig. E.5 becomes of interest. When the excitation displacement of 0.2 mm is directly applied to the model at 84 Hz, the system does not get the change to increase the mass displacement, and with that neither the stiffness nor the resonance frequency. The amplification will therefore be smaller than when the excitation displacement is applied to the model at an increased frequency as in Fig. E.2

The reason why there is chosen to start at 73 Hz is because of the natural frequency of the concept model around its equilibrium point. This ‘stall resonance frequency’ is found to be 73.8 Hz, and to be on the good side there is chosen to start at 73 Hz.

The effective mass that is used for these simulations is the mass of the lighter connection point. However, for the real concept model the system reached a resonance frequency of 95 Hz instead of the simulated 84 Hz. In addition, the amplification is larger for the simulation (133 times the excitation displacement) than for the real concept model (112 times). These differences are because of simplification of the concept model in the Simulink model. For example at the fixation points of the specimens it is assumed that the specimens can move more freely in a hinged way with respect to the mass and the connection with the punch. However, the specimens are clamped at the connection point in the middle and hold at two sided by the magnets. Both these impositions do not allow the specimen to move as freely as it could have moved when they were hinged. The effect is that the pulling forces of the specimen on the mass are not taken into account in the Simulink model.

For validation of the theory that the progressive spring behaviour is the cause of the excitation displacement dependent resonance frequency these simplifications do not make a difference. First, the pulling forces on the mass the specimen act on the mass are much smaller than the pushing forces, so their effect on this dependency is minimal. Next to that, the specimens still act the forces on the mass according to the progressive behaviour that was found earlier, which is the main reason of the dependency.

5) Conclusion

Form the simulations done with a simulink model of the concept model it can be concluded that the dependency of the resonance frequency on the excitation displacement is due to the progressive spring behaviour of the specimen under two-point bending. It is also shown that the excitation displacement should be applied to the model at an increased frequency in order to obtain the displacements that are wanted for the fatigue test.
APPENDIX F

CURRENT FATIGUE TEST

1) Introduction
The newly introduced fatigue test discussed in this report is an improvement on the current bending fatigue test. This appendix describes the current fatigue test in order to give a better understanding in how the new fatigue test is an improvement to the current test.

2) Test specifications
The current fatigue test is shown in operation in Fig. F.1. The basic principles of a fatigue test are used to describe the test setup.

A. Actuation
The actuation of the current bending fatigue test is done with the Bose ElectroForce 3330 machine. Specifications of this machine can be found in Appendix I.

B. Movement Translation
Movement translation is done by one to one application of the actuator. This means that the stresses are introduced into the material by a displacement of the specimen that is equal to the displacement of the actuator.

C. Bending Stress Introduction
The bending stresses are introduced in the material by two point bending. There is chosen for two point bending for the same reason why there is chosen for this bending principle for the new bending fatigue test. This bending principle give a clean spot of concentrated stresses free of contact points that can influence the bending stresses.
3) Operational Properties

The current fatigue test is able to bend the specimen at a rate of 60 to 65 Hz. At this frequency the punch makes a displacement of 10 mm that directly bends the specimen. The initial distance between punch and basis is 25 mm. The punch makes an oscillating movement of +5 mm and -5 mm around the initial distance, for the fatiguing stresses in the specimen. This specimen is held by two magnets on each side, that prevent the specimen from sliding off the test rig. Measurement show that the required electrical power during this test is 750 W.

4) Discussion

The reason why the this test cannot bend the specimen at a higher rate is due to power limitations of the actuation machine. According to equation (F.1), moving a mass requires power.

\[
Power = Force \cdot Velocity = Mass \cdot Acceleration \cdot Velocity \quad (F.1)
\]

According to this equation the frequency and the excitation displacement have a huge effect on the power requirement of the machine. The power scales quadratically to the displacement and even cubically to the frequency. Since a higher frequency is wanted, improvements can only be achieved by reducing the displacement.
APPENDIX G

CONCEPT MODEL FATIGUE TEST UNTIL FAILURE

1) Introduction
The development of the new fatigue test can only be considered successful if actual fatigue tests can be done with the new method. That is why some tests are done in which the specimen is bent with the new fatigue testing method until failure to see if the obtained results are expected and conform the current fatigue test.

2) Method
To make the new fatigue test comparable with the current fatigue test, the same stresses should be introduced into the material. The stresses introduced by current fatigue test as it is described in Appendix F are different from the stresses introduced in the tests done on the concept model in Appendix D To make the test comparable the displacements the specimen makes will be kept the same. This means for the new fatigue testing method an initial distance between the clamps of 50 mm and a mass displacement during fatiguing of 10 mm.

Due to this smaller initial distance between the clamps with respect to the test from Appendix D the mass is now also able to move horizontally as well. The specimens move horizontally by rolling over the magnets. To limit this degree of freedom magnets at the clamps are now placed at both sides of the specimen see. Fig. G.1.

To start the test the excitation displacement (+0.1 and -0.1 mm) is applied to the specimen at 85 Hz. From this point the frequency is increased until the mass movement is visually perceptible. Now the mass displacement is measured, and the frequency is increased until the mass displacement reaches 10 mm. The mass is now making the desired movement, so the specimen is left for fatiguing in this way. Finally stopping criteria for the measured force and displacement are set, so the machine will stop when the material fails.

In order to judge if the results from this test are relevant, the same rings are also tested by the current fatigue testing method, as the test described in Appendix F. During this test also the current through the system is measured. Multiplication of this
value by the voltage over the systems (230 V) gives the electrical power that is required for this test.

3) Results

In total five specimens were tested until failure with the new testing method of which the results can be found at the left side of table G.1. The reference values of the rings that were tested by the current fatigue test can be found in the right side of the table.

Measurements showed that the electrical power requirement during the new fatigue testing method is 60 W. The current fatigue test reaches 750 W.

<table>
<thead>
<tr>
<th>New Fatigue Testing Method 95 Hz</th>
<th>Current Fatigue Test 60 Hz</th>
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<tbody>
<tr>
<td>306,683</td>
<td>31,771,409</td>
</tr>
<tr>
<td>159,848</td>
<td>1,548,892</td>
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<tr>
<td>18,873,903*</td>
<td>95,458</td>
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<td>70,145</td>
<td>1,172,608</td>
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<tr>
<td>5,143,879*</td>
<td>41,500,000*</td>
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<td></td>
<td>39,154,429*</td>
</tr>
<tr>
<td></td>
<td>85,755,021</td>
</tr>
<tr>
<td></td>
<td>93,158,772*</td>
</tr>
</tbody>
</table>

*Test stopped manually, no failure yet.
4) Discussion

The results in Table G.1 show that the number of cycles to failure is comparable to the current test. Until now only five tests using the new method are done, which is little for drawing a conclusion, but these five tests already show similar results as the current fatigue test.

The crack initiation point is located in the middle between clamp and mass at the spot where it is expected to be after the bending fatigue test. This can be seen in Fig. G.2, where the crack can be found at the edge of the ring. This is the place where the highest stresses are introduced into the material during bending. This is verified by models of Bosch Transmission Technology of which the result can be seen in Fig. G.3. The blue line, that gives the stresses in the most outer fibre of the ring, reaches the highest value at the edge of the ring.

This also means that the stresses at the connection points with mass and clamps do not give peak stresses that will cause the material to fail. It can be said that the rings were thus purely failing due to the bending stresses introduced by the test.

Table G.1 shows relatively more low number of cycles to failure results for the new bending fatigue test than for the current test. This is due to the fact that the new method stresses the material at four points, where the current fatigue test has only one stress concentration. The test is stopped when the first crack in the surface is noticed. These cracks mostly appear at weak spots at the surface. The chance that a test at the new method stresses the material at a weak spot is four times higher than when it is tested at the current fatigue test. The results from the current fatigue test show that a low number of cycles to failure occurs every once in a while.

One might wonder why the number of cycles to failure deviates highly for the new fatigue test as well as the current fatigue test. The number of cycles to failure reach form 70,000 to over 93,000,000 cycles. This is because the tests stress the material at a stress level equal to the stress at the horizontal asymptote of the S-N curve. A sketch of the S-N curve of the ring as it is obtained from by the bending fatigue test can be seen in Fig. G.4. The horizontal asymptote can be found at an effective stress in the outer fibre of around 625 MPa. And as it can be seen in Fig. G.3, this is the stress...
level at which the tests to failure are done at. This means that the number of cycles to failure is likely to deviate highly for this stress level.

The electrical power requirement of the bending fatigue testing method has been reduced massively. A reduction of 92% is measured with respect to the current fatigue test. If also the reduction of testing time is taken into account it can be calculated that the energy requirement for one fatigue test is reduced by 95%.

As it was mentioned in Appendix F, the power is dependent on the moving mass, the frequency and the displacement. The frequency is for the new test higher than for the current test and so is the moved mass (two magnets instead of one used for fixation). This means that the power reduction is purely because of the excitation displacement that is much smaller for the new testing method. In fact, the excitation displacement for the current fatigue test is 50 times larger than for the new testing method. Since reduction of excitation displacement is because of the Spring-Amplified-Resonance principle that is introduced for fatigue testing in this research, it can be said that the efficiency that is gained for the fatigue test can be attributed to the SAR-Principle.

The reason why the number of cycles to failure is low compared to the number of load cycles a ring has to withstand in a CVT in a car, is that stresses during the fatigue test are larger. The fatigue test is intended to prove materials under bending stresses and for construction of the S-N curve for bending stresses; not to simulate the actual stresses in a ring during use in a CVT.

5) Conclusion

Until now only five tests have been done using the new fatigue testing method, which is too little for drawing conclusions. However, similar results are obtained if the results are compared with the result of the current fatigue test. Next to that, a large benefit is obtained in terms of power requirement. The reduction of 95% that is gained by the new bending fatigue testing method can be attributed to the Spring-Amplified-Resonance principle.
APPENDIX H

TEST VARIATIONS

1) Introduction

In order to increase the frequency or to test thicker material some variations to the new fatigue test have been made. These variations are tested, which will be described in this appendix.

2) Method

A. Increasing Bending Frequency

Increasing the stiffness is one way in which the resonance frequency can be increased. In case of this 8-shaped-model it is the fastest and easiest way to makes large steps forward. That is why a test is done in which additional springs are used to increase the stiffness.

The additional springs are put right in the middle of the model as can be seen in Fig. H.1. In this way the stiffness in the vertical direction is increased and the spots of concentrated bending stresses are still as ‘clean’ as before. The springs are at one side connected to the mass. To prevent sliding of the spring over the ring at the other side, double-sided tape is used to increase friction. Since the whole system is put under pretension before testing, the springs will not get loose of the rings. At both sides the model is placed on two super-magnets in order to get a better fixation.

B. Testing Thicker Material

At this moment also bending tests are done on thicker material. In this way a smaller displacement can be applied to the specimen, while still the same stresses are introduced. This thicker material is the same material as of the rings only in an earlier stage of the production process of the ring. The thicker ring gets the same surface treatment, so the surface is the same as the surface of the ring which is used for the pushbelt. Since the highest bending stresses in the material are at the surface of the ring, comparable results are obtained by testing the thicker material.
For this test one ring of the original concept model is replaced by a thicker ring, see Fig. H.2. It can be seen that the ring of the thicker material (0.4 mm thick) is larger than the ring of the thinner material (0.2 mm thick). This is because the thicker material should be less bent for the same stresses. This test is mainly executed to find the influence on the resonance frequency. On beforehand it is not known if the stiffer ring will increase the frequency with respect to the concept model, or that the extra mass of the ring will decrease the resonance frequency.

3) Results

Both test setup variations were able to resonate and resulted in an amplified oscillating motion of the mass. The found maximum resonating frequency for the additional spring setup was 140 Hz. However, the mass displacement did not reach a larger displacement than approximately 5 mm, before it was damped out again. The test setup with one thicker ring reached a natural frequency of 68 Hz; the mass displacement reached over 10 mm.

4) Discussion

The setup with additional springs to increase the frequency does not reach the mass displacement that is sufficient for a fatigue test. The extra springs and probably the double sided tape as well, introduce more damping in the system that prevent is from a larger mass displacement. However, it is demonstrated that the testing frequency can be increased by increasing the stiffness with additional springs, so with a proper design and the right springs it is possible to reach a sufficient mass displacement at a higher frequency.

The setup with the thicker material did reach over 10 mm mass displacement, which is at least large enough for a fatigue test on the thin ring. It is not known if this displacement is also sufficient for a fatigue test on the thicker ring.
However, the higher stiffness of the thicker material could not prevent that the extra mass of the ring decreases the resonance frequency. Where the concept model reached to 95 Hz with this mass, the setup with one thicker ring only reaches 68 Hz. This is only 8 Hz faster than the current fatigue test and will be reduced even more when two thicker rings will be used. For this reason the new fatigue testing method cannot be called an improvement for the test of thicker material.

5) Conclusion
Tests are done with variations to the new fatigue testing method. It was found that additional springs will increase the frequency, so this concept can be valuable for future improvements to the new bending fatigue testing methods. Tests on thicker material do not show direct improvements, so this testing concept can be considered less valuable.
APPENDIX I

BOSE ELECTROFORCE MACHINE SPECS

1) Introduction
Here the specifications of the Bose ElectroForce machine can be found. This machine is used for the current and new bending fatigue test of the CVT-pushbelt ring.

2) Specifications

<table>
<thead>
<tr>
<th>Specifications</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Developer/type</td>
<td>Bose ElectroForce 3330</td>
</tr>
<tr>
<td>Working Principle</td>
<td>Electromagnetic</td>
</tr>
<tr>
<td>Frequency Range</td>
<td>1-100 Hz</td>
</tr>
<tr>
<td>Max. Displacement</td>
<td>Peak to Peak</td>
</tr>
<tr>
<td>Max frequency</td>
<td>1 mm</td>
</tr>
<tr>
<td>Min Frequency</td>
<td>25 mm</td>
</tr>
<tr>
<td>Maximum Force</td>
<td>3 kN</td>
</tr>
<tr>
<td>Power</td>
<td>750 W</td>
</tr>
</tbody>
</table>
Fig. I.1: Displacement-Frequency characteristic of the Bose ElectroForce 3330, at different moved masses.

Fig. I.2: Picture of the Bose ElectroForce 3330 machine.
APPENDIX J

DESIGN PROPOSAL FOR BOSCH TRANSMISSION TECHNOLOGY

1) Introduction
Although the new bending fatigue test already increased the frequency with more than 50%, there are possibilities to increase it even more. As it was found in Appendix C, the frequency can be gained most by using another bending principle. Next to the differences in maximum frequency, there is a large difference between the options in terms of feasibility. Not all of the design options are equally suitable for a high frequent fatigue test. The fixation of degrees of freedom and the resulting friction, together with damping that might be high for some options, make that the options differ largely in feasibility for the bending fatigue test.

2) Proposed design
The option that is selected for further investigation is the cantilever bending of which a schematic drawing can be found in figure below. This is one of the simples design options and is already proven by the proof on concept in Appendix A to work very easy.

Fig. J.1: Schematic drawing of the proposed design.
Different characteristics of the system influence the resonance frequency of the design. Therefore a study is done to find the optimal values for the characteristics in order to obtain a maximum resonance frequency, while still the requirements are met.

A. Frequency Dependency Analysis

The main goal of this design is to reach an as high as possible bending frequency. This is however dependent on many properties of the systems. The properties that have the largest influence on the resonance frequency, which can be varied for this study, are:

- Specimens free length
- Point mass
- Spring stiffness

In order to describe the influence of the different system properties for every property an frequency dependency analysis has been done. In this analysis the specific property is varied, while the others remain the same. By varying one property at a time it can be determined what the isolated influence is of this property on the system frequency. The standard length is 25 mm, the standard mass is 2 g and the standard spring stiffness is 500 N/m. These are the approximate values as they hold for the proof of concept, which result in the approximate resonance frequency of 95 Hz. In this way, the proof of concept is a reference for the variations of the properties.

There are other properties that can influence the resonance frequency as well. For example the specimen thickness and the Young’s Modulus of the material. These properties are considered constant, since they are constrained by the specimen that is to be tested.

Specimen Free Length

The free specimen length is the first property to be varied. As can be seen in Fig. J.2, the natural frequency decreases rapidly as the length increases. If the system is simplified by the formula \( \omega_n = \sqrt{k/m} \) where the stiffness of the specimen is determined by classical beam theory, \( k = \frac{3EI}{l^3} \), there can be found that the resonance frequency is scaling to \( l^{-2} \). In this scaling factor the influences of mass variations of the system are taken into account as well.
It could be said that to obtain the highest frequency, the specimen free length should be as small as possible. However, the length cannot be chosen to small. There should be at least enough space to create the stresses in the material that are desired for the test. Next to that the specimen should be long enough for the connection point with the spring.

**Point Mass**

The mass of the point mass that functions as the connection point is very determining for the resonance frequency of the system. As it appears from Fig. J.3, the maximum resonance frequency is 360 Hz, at the point where there is no point mass and the system only resonates by the own mass of the spring and specimen. If a point mass is added, the frequency decreases rapidly by each heavier gram and then slowly converges to 0 Hz, where an infinite mass gives an infinitely small resonance frequency. Using the earlier mentioned formulas is can be determined that the resonance frequency scales to $m^{-1/2}$.

Adding no point mass to the system is physically not possible. At one way or another, the spring has to be connected to the specimen, which always adds some weight. From these results it can be said that connection point should be chosen to be minimal in order to get a maximal resonance frequency.

**Spring Stiffness**

The spring that actuates the system is also of influence for the resonance frequency. Since the stiffness of the specimen and the spring stiffness can be considered to work parallel, the values can be added up in order to get the system stiffness. This means that the resonance frequency will increase as the spring stiffness increases, scaling by $k^{1/2}$. This is shown in Fig. J.4. Here the minimal frequency is obtained if no spring is used, so the specimen only resonates with the mass. There is no theoretical maximum since and infinite stiffness will lead to an infinite resonance frequency.

However, there are reasons to believe that there are some restrictions to the spring stiffness. These restrictions are not clear at this moment, but damping and size of the system are likely to influence the maximum spring stiffness. For this research it is chosen to take a spring with a stiffness that is in the same order at the stiffness of the

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Fig. J.3: Frequency dependency on the mass.
specimen. In this way there is still a significant increment of the resonance frequency, and it is sure that the Spring-Amplified-Resonance principle will still work.

B. Stress Concentration

Next to the criteria of obtaining a frequency as high as possible, there is also the criterion of the concentrated stresses. The stresses as they are introduced in the proof of concept are concentrated at one point, at the fixation point of the specimen. This is unwanted since the stresses are then influenced by the fixation point instead of being purely bending stresses. This is why the model of the proof of concept is not perfect jet for the bending test this study is aiming at.

To solve the problem of maximal bending stresses at the fixation point there can be made use of contact aided bending. This principle uses a surface over which the specimen can be bent in order to introduce the maximum stresses at another point than the fixation point. The shape of the surface defines the stresses that are introduced in the material. This principle is schematically drawn in Fig J.5.

As a consequence of using this principle, is that you need more space on the specimen between the fixation point and the connection point with the spring. After all, the shape has to fit in between. However, this does not necessarily mean that the resonance frequency is lower. The free specimen length can now be considered to start at the contact point with the shape instead of at the fixation point. But then there should always be contact between the specimen and the shape, otherwise the frequency will be reduced. This can be achieved by adding enough pre-tension to the specimen.

Fig. J.5: Schematic drawing of contact aided bending
C. Rough Model

In SolidWorks and ANSYS a rough model is made of this principle. The values that are taken for the properties discussed in the section ‘Frequency Dependency Analysis’ are not optimal yet. This model is more of a sketch that gives an indication of how a new optimized design could look like.

In this rough model the fixation of the specimen is left out, so only the specimen, the mass and the shape over which the specimen is bent are visible. From this model it appears that it is indeed possible to introduce a concentrated bending stress in the material if contact aided bending is used. Also the contact point with the mass results in stresses in the material, however these are lower than the concentrated stresses due to the bending and therefore acceptable. During optimization of the values it should be kept in mind that the mass should not be too close to the bending stress concentration, otherwise these stresses will be influenced.

3) Discussion

Finding the optimal values for the variables that are discussed in the ‘Frequency Dependency Analysis’-section is not straightforward. There are a lot of factors that will affect the final results. The variables have their restrictions and they will influence each other. A larger shape will lead into a longer specimen, etc.

A rather determining factor of the system is the damping factor of the system. The damping determines how fast the system will damp out after the system is stopped, and it determines the maximum amplification of the system. This factor is a result of the three variables and is difficult to predict on beforehand.

These unclarities make that tests need to be done in order to find the optimal values to obtain a maximum frequency for the fatigue test with this principle. To do so a test rig should be built in which the specimen length, mass and spring stiffness can be varied to find out how they affect the frequency and damping in real life. This will clarify the affects and give a better understanding on the mutual influences of the properties.
4) Conclusion

This Appendix is intended to indicate some improvements that still can be obtained. A design is proposed which potentially reaches frequencies significantly higher than the Concept that is studied in this report. A frequency dependency analysis is done for different properties of this design, to find their influence. Further testing should be done to find out what the limitations for the properties are that determine the resonance frequency. In this way it can be found for a real model how these properties influence and limit each other. With the maximal frequency obtained by the tests it can be decided whether or not to develop a fatigue test according to the proposed design.
LITERATURE STUDY
A Comparative Study on Cycle Fatigue Testing Methods and Machines

Literature survey

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Abstract — Since fatigue tests are more commonly executed nowadays, the variety of machines available for these tests increases. For all sorts of fatigue tests, a variety of systems is available of which there can be chosen from for a desired fatigue test.

The objective of this study is to guide in the decision in this respect. This study is aimed to be an easy accessible comparison that informs about the options there are with respect to fatigue testing equipment.

In this study an overview is presented of available fatigue testing equipment. The working principles of the different systems are explained and a representative selection of systems is judged on their characteristics. There is focused on equipment that is at least capable of reaching the high cycle regime and that makes an oscillating motion to introduce the stresses in the test piece. Two new comparative metrics are introduced to give more insight in how the systems differ from each other on the field of working range and fatigue test efficiency.

The wide variety in which fatigue testing equipment is available, make that the decision for a system a delicate procedure. Two comparative metrics are introduced in this research, which are found to be helpful in this respect.

Keywords: High cycle fatigue testing; fatigue testing equipment

I. INTRODUCTION

Fatigue testing is a widely used test in engineering to find out if materials or constructions hold after a certain amount of cyclic loads. During a fatigue test forces are repeatedly applied to the material to introduce stresses in the material in order to find the number at which it fails or to study the crack grows. The constant need of making products smaller and lighter, while it still needs to withstand the same circumstances makes that fatigue tests are more commonly executed nowadays.

Performing a fatigue test is mostly a very time consuming undertaking. Reaching a high number of load cycles at a limited rate can make this a long lasting process. Since the existence of a never ending horizontal asymptote in a Stress vs. Number of cycles to failure Curve (S-N curve) is seriously doubted, more fatigue tests are done which reach to more than $10^7$ cycles [1]. In order to reduce testing time to a minimum, the need for machines that are able to apply loads at a very high rate is higher than ever.

With the years more fatigue testing systems came available of different types and sizes, with the result that there is a wide variety of machines these days. For all sorts of fatigue tests, systems are available which are suited for the tests. This makes that there can be chosen from a wide range of machinery which can be used for a particular fatigue test.

Studies have been done to make an inventory of available fatigue testing systems. However, these studies only focus on one type of driving mechanism [18], are outdated due to the fast development of fatigue testing systems [2] or are too concise to describe the world of fatigue testing [3]. Until now there is no modern and comprehensive overview of available cycle fatigue testing machines and methods.

The purpose of this study is to help in the decision for a fatigue testing machine or method. It should give an overview of different types of fatigue testing machines, which enables one to make a better comparison between the systems. From this study the reader should be informed about the options there are for fatigue testing and should be guided in the decision for a system.

In the next section it is described what is done to achieve this goal. The third section gives the results of this study, containing the basic working principles of fatigue testing.
systems and their characteristics. The results are discussed in the fourth section and concluded in section five.

II. METHOD

A. Search Method

To obtain the information that is needed for this literature survey there is looked at different databases for articles and patents. Finding useful articles and patents requires the right places to search together with the right search criteria. Searching for relevant information is done in online resources as well as offline resources. The used sources can be found in the references.

1) Patents

The most commonly used online database to find patents for this paper is Espacenet [4]. This is a free online service developed by the European Patent Office (EPO) in order to offer an easier and better way to search through patents. One way in which they improved the search result is to categorize the patents and add a feature on the website to show all patents within a selected classification.

The classification for fatigue testing machines can be found in G01N3/32. This category is found by choosing out of the categories the category that is most relevant to fatigue testing machines. The most abstract category in which fatigue testing machines are categorized is G; Physics. Basically the machines are used to test the physics properties of the specimen. The next specification is G01; Measuring (Testing). The other choices within the Physics category are not relevant for such machines or methods. Within G01 only G01N; is relevant. This category is described as Investigating or analysing materials by determining their chemical or physical properties. Since the machines of interest are only applying stress to the specimen to test the properties, the next specification of interest is G01N3; Investigating strength properties of solid materials by application of mechanical stress. The last category which fully describes all fatigue test machines is G01N3/32; By applying repeated or pulsating forces.

Within G01N3/32 there can be made several more choices in categories, but these will only specify the type of fatigue testing. The following categories are there to specify in the kind of stresses that are applied and by what means the stresses are applied. All of this falls within the scope of this literature survey.

2) Articles

Finding relevant articles regarding fatigue testing is mostly done with Google scholar [5] and Scopus [6]. With this kind of websites it is important to use the right terms in order to find good information. Various combinations of the following words have been used as search terms in the mentioned search engines.


These search terms result in a lot of articles of which not all of them are relevant. Among these results there is made a selection of the most useful articles for this literature research.

3) Offline resources

For offline resources such as books, the same search criteria are used as for the online resources. Next to that also the also some books were found that were used as references for articles that were found earlier. This gives the chance to find more detailed information on the relevant part of the articles.

B. Selecting the systems

If the described search procedure is followed, it becomes apparent that there are too many fatigue testing systems to be discussed and compared in one paper. For this reason the field in which the machines operate is restricted. Next to that, there is made a selection of available systems that describes the world of fatigue testing equipment.

1) Restrictions

The first restriction that holds for this paper is that only fatigue tests to reach at least the high cycle regime are taken into account. This means that the systems within the scope are fatiguing the test material on such high rates that the high cycle regime (> 10^6 cycles) can be reached in an acceptable amount of time. Low cycle fatigue tests are not taken into account, since the tests differ too much from test systems that are used for high cycle fatigue tests.

Next to that only equipment that makes an oscillating motion in order to introduce the stresses in the test pieces are taken into account. This restriction allows making a judgement on the basics principles of the machine. It can be said that only systems capable of testing specimens on tension/compression and bending are within the scope of this study then. Machines only capable of testing the torsion fatigue of the specimen in general do not make the oscillating motion and are therefore not comparable enough to be included in this research.

2) Selection

Some of the systems are selected because of their unique way of testing. The principles they use to introduce the stresses in the material are different from the usual methods. These systems are mostly the experimental setups that will be discussed in this paper.

Next to these unique fatigue testing methods, there are also more commonly used industrial machines for fatigue testing. These machines are suited for a larger variety of fatigue tests than the experimental setups. The number of industrial machines that are made these days is quite high. It speaks for itself that not every system can be included in this study. That is why a further selection of representative machines is made. Each of the selected machines stands for a range of machines that generates the oscillating motion in the same way.

C. Criteria

To make a comparison between the different systems, several criteria will be evaluated. These criteria are mostly characteristics of the systems operating behaviour. Next to these general specifications also some more advanced criteria are presented in order to make a better comparison between the fatigue testing machines and methods.
1) Frequency range

The operating frequency is one of the most important criteria considered in this study. The operating frequency gives the rate at which the system is capable of applying a load on the specimen. All systems are bounded by a maximum frequency and some systems are also bounded by a minimum frequency at which they can operate.

2) Displacement

The displacement (also stroke) indicates the size of the oscillating movement the system is capable of making in order to introduce the stresses in the specimen. For some cases the movement is dependent on the frequency the system is operating at that moment. At higher frequencies these systems do not have the power to still use the full reach of which it is capable at low frequencies.

3) Load cases

The specimens can be tested in various ways in which they are loaded differently. The load cases of the machine indicate how the machine is able to load the specimen. The possible load cases that can be applied to the specimen are tension/compression, bending, and torsion.

The tension and compression load case is schematically shown on the left side of Fig. L.11. This is the most common fatigue test to be executed on test specimens. This load case includes tension-tension tests and compression-compression tests and intermediate test such as tension-compression tests. The bending and torsion load cases (right side of Fig. L.11) are less common for fatigue testing.

4) Maximum force

Also a rather characterizing specification is the maximum force. It gives a measure of the forces the systems are able to apply dynamically to the specimen and gives therefore also an indication of the size of the specimen that can be used with it and the stresses it can apply to the specimen. The maximum force mentioned for each system in this paper is the dynamic force and the systems do not necessarily be capable of applying the maximum force at their top frequency as well. For some fatigue testing setups there can be found a relation between the frequency, displacement and the maximum force. The forces investigated for this study are the highest forces that can be found in the dynamic loading reach of the systems, whatever the corresponding frequency may be.

5) Power

The power of the system gives a measure for the energy the system requires to operate at one of its limitations (maximum frequency, displacement etc.). Not all tests require the machines to operate at this maximum power. The value given for the systems is the power that is specified by the manufacture, or that can be calculated from the component specifications.

6) Working Range Size (WRS)

The frequency and the displacement together give a good impression of the working range of the system. If these criteria are set off against each other in a graph, some specific capabilities of the systems can be made visual. For instance the dependency of the displacement with respect to the frequency will become apparent.

While some systems can apply a large range of frequencies, others can generate a large stroke within a smaller range of frequencies. It is interesting to see how these criteria hold against each other for each system and therefore a new metric is introduced. This metric is equal to the integral of the maximum displacements at the corresponding frequencies, according to equation (1). In this equation $f$ is the frequency and $s$ is the maximum displacement possible at the corresponding frequency. Basically this gives a measure for the amount of tests that can be done with the machine. Low WRS values indicate a limited machine; high WRS values indicate that the machine can be used for a wider variety of tests.

$$WRS = \int_{\min(f)}^{\max(f)} s \, df \quad [\text{mm} \cdot \text{Hz}] \quad (1)$$

However, the working range size does not fully describe the behaviour of the systems. A high working range size may be at the cost of the forces a system can apply on the specimen. Next to that, systems having a large WRS, might have a high power consumption to achieve that. Therefore yet another metric is introduced.

7) Fatigue Test Efficiency (FTE)

The second metric to be introduced in this paper combines multiple physic specifications of the systems. The criteria that are desirable to be of a large value are the frequency, displacement (WRS) and force. The specification that is desired to have a low value is the power consumption. If the desired values of the systems are multiplied with each other and divided by the power, a dimensionless quantity is obtained, see equation (2).

$$FTE = \frac{WRS \cdot Force}{Power} \quad [-] \quad (2)$$

The Fatigue Test Efficiency can act as a measure to compare the systems with each other in terms of efficiency. A high FTE indicates that the system is able to do fatigue tests in its working range in an efficient way. This means that the FTE makes it possible to judge the systems on how they hold against each other, in spite of the fatigue tests the system is originally designed for.
III. RESULTS

The methods and machines that are within the scope of this study are described here. First the working principles are described, after which an overview is presented of the selected fatigue testing methods and machines and their characteristics.

A. Driving systems

Fatigue testing systems do not generate the forces and displacement in the same way. How these systems are driven is actually quite different. The most common driving systems to be used for high cycle fatigue testing are discussed shortly in the next paragraphs to give a better understanding about the systems and their differences.

1) Electromagnetic

Magnetic driven fatigue tests are commonly used in the high cycle fatigue industry. The working principle of the electrodynamic magnetic driven test machines is derived from the loudspeaker. The core is basically the same, only some adjustments have been made to make it suitable for fatigue testing (e.g. the parts in the test machine are more robust). It all starts with a signal generator which produces an electric signal that is desired for the test. The voice coil is the part that generates the motion and force that are to be applied to the specimen. The motion of the voice coil is caused by a magnetic field that arises from the signal. A schematic drawing of a voice coil is shown in Fig. L.12.

The capabilities of the electrodynamic magnetic driven test machines make them suitable for a large range of test setups. There can be found machines which are capable of applying high forces, large displacement or apply them very high frequent. [23]

Next to this regular voice coil principle there is also the linear moving magnet motor. The basic principles are the same for this type of actuation, only here the magnet makes the translational movement, and voice coil holds still and only generate the magnetic forces needed for the movement. [25]

2) Servo-hydraulic

Also a very commonly used drive for fatigue testing machines is servo-hydraulic. Basically, these systems contain three parts. An actuator, a servo valve and a high pressure oil supply. The actuator applies the actual forces and displacements to the test specimen by converting oil pressure into movement or force. The actuator is a hydraulic cylinder with an oil inlet at both ends. The oil pressure is caused by the servo valve which is schematically drawn in Fig. L.13. What this servo valve does is letting the pressurized oil pass through to one of the two ends of the actuator. By means of a magnetic coil, a flapper is pushed against either one of the two nozzles. This causes a pressure drop at one end of the servo and a pressure rise at the other end. The servo will move and so the oil is free to pass the servo at one side to the corresponding end of the actuator.

Servo-hydraulic fatigue test systems can apply the forces repeatedly to around 100 Hz. The displacement and force are limiting the maximum frequency of the system. However, there are special cases with a slightly different servo valve which can load specimen up to 1000 Hz. In the mentioned high frequency hydraulic fatigue test machine the magnetic coil drives the servo directly and not via a nozzle system [22].

3) Piezoelectric

Piezoelectricity is generated by certain solid materials (e.g. crystals, ceramics) if a mechanical stress is applied to the material. It also works the other way around. If an electric voltage is applied to the material, it will change its static dimensions. This phenomenon is relatively new for fatigue testing setups.
However, the mechanical deformation of the material is limited. For that reason the movement of the piezo converter has to be amplified to obtain the desired displacement. Basically all piezo driven fatigue tests have an amplification mechanism to increase the displacement. Only then the specimen will be loaded with the stresses that are desired for fatigue testing. If the amplification mechanism is not used, the specimen has to be small to obtain the desired stresses in the material. Scaling factors can be of influence on the fatigue tests then.

4) Other working principles

For the sake of completeness two other working principles should be discussed as well. These working principles are too different form the already mentioned working principles to be included in the comparison. However, these machines are still frequently used these days and can therefore not be excluded from this research.

The oldest and simplest way of driving mechanism the electromotor-rod combination in which the rotational movement of the motor is translated to a reciprocating movement of a rod which is connected with test specimen. A schematic representation of a bending test is given in Fig. L.6. This example is to illustrate how fatigue tests used to be done. Nowadays fatigue testing systems are more advanced and are therefore not comparable with systems like this. The speed and cycle regime are not sufficient for such systems to be further considered in this research.

Another type of fatigue testing is the rotating bending test. It was one of the first to be used to reach the high cycle regime in fatigue testing. The simplicity combined with the feasible speeds is what makes it still popular to be used these days. In this test a test specimen is rotating, while it is clamped at both ends. On both clamps a force is applied, which makes the specimen bend. Since the specimen is rotating, the test can be considered as a bending fatigue test. This is schematically shown in Fig. L.5. Since stress introduction is done without an oscillating movement this type of test is left out of this comparison.

B. Fatigue testing machines and methods

As it was mentioned before, there is made a selection of fatigue testing equipment, which describes the world of fatigue testing. The selection consists of experimental fatigue testing setups as well as industrial machines. The experimental setups are unique in their way of testing and they give insight on the diversity there is among fatigue testing machines. These test are specifically designed to do only one type of test and are therefore limited applicable for other tests.

The industrial machines that are selected can be used for a wider variety of tests. Since there is a large amount of machines available, there is made a division in this study to get a manageable and clear overview. The division is based on the four basic principles of actuation of the machines:

- Electro-Magnetic Resonator: Resonance amplification. (8. Rumul 20 kN Resonator)
- Servo-Hydraulic Oscillator: Hydraulic Actuation. (Shimadzu Hydraulic Servopulser)

From each of these categories there is chosen one machine that serves as a model for this category. The selected machine is named in parenthesis behind the category. The selected machine is thought to be average for this kind. For the selection a critical look is given at the characteristics of different machines and the one that is selected represents the machinery of that kind. In this way the amount of machines to be judged is decreased while still a comparison can be made between the systems.

In the appendixes at the end of this paper, all the discussed fatigue testing machines and methods, with their specifications, a graph and a picture are presented as a quick reference to the tests. A summary of the system characteristics can be found in Table 1. This table gives in short the capabilities of the machines. The values in this table are discussed in the discussion part of this paper.
Table 1  Overview of the fatigue testing systems and their characteristics. Systems 1-4 are the experimental setups, systems 5-9 the industrial test machines. The * indicates the fatigue testers that make use of resonance.

<table>
<thead>
<tr>
<th>System data list</th>
<th>System specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Bathias*</td>
<td>100-20000</td>
</tr>
<tr>
<td>2. Straub*</td>
<td>100-8500</td>
</tr>
<tr>
<td>3. Kenner-knecht*</td>
<td>100-8500</td>
</tr>
<tr>
<td>4. Kim</td>
<td>1-250</td>
</tr>
<tr>
<td>5. MTS</td>
<td>1-1000</td>
</tr>
<tr>
<td>6. DPC</td>
<td>1-7000</td>
</tr>
<tr>
<td>7. Bose ELF</td>
<td>1-200</td>
</tr>
<tr>
<td>8. Rumul*</td>
<td>60-250</td>
</tr>
<tr>
<td>9. Shimadzu</td>
<td>1-200</td>
</tr>
</tbody>
</table>

IV. DISCUSSION

A. Criteria Evaluation

Table 1 shows the system specifications of the selected systems. This table can help in the decision for a fatigue testing system in multiple ways. If one is interested in performing only one type of fatigue test, the second to the fifth columns are of interest. If one wants to purchase a system that is more generally useful, the last two columns will also become of interest.

For most criteria the differences between the systems are fairly big, as can been seen in Table 1. In this table a distinction can be seen if the experimental setups (systems 1-4) are compared with the industrial test machines (systems 5-9). The values for displacement, WRS and FTE of the experimental setup are in general lower than the values of the industrial machines. The most relevant distinctions are discussed below.

1) Frequency

The frequency range is an evaluation criterion that varies a lot among the systems. Where a system like the MTS can cover a range of 1000 Hz, the Rumul machine can only cover a range of 190 Hz. The Ultrasonic test of Bathias is even capable of reaching 20 kHz. However, it has to be rebuilt to reach other frequencies. The system is designed for a specific frequency and is therefore only able to deal with a limited range of frequencies. If other frequencies are desired, some dimensions of the setup have to be adjusted. This makes it a typical example of an experimental test setup.

Typical for the resonant cycle fatigue test is that they have a minimum frequency. This is because these systems are only able to operate in the natural frequency of the specimen and the test setup together. Since normally the natural frequency of the specimen cannot be found in the low regions (1-100 Hz), the machines are not able to operate at these frequencies. However, it is all very dependent on the dimensions and material of the specimen.

The highest frequencies can be reached by systems that use the natural frequency of the specimen to resonate. Fatigue tests are done at 40 kHz [8]. Non-resonating systems can manage to reach up to ± 7000 Hz for electromagnetic machines and ± 1000 Hz for hydraulic systems. These values can be considered as upper limits. More common maximum frequencies for this kind of machinery can be found in the region of 200 Hz.

2) Displacement

From the data of all the non-resonant fatigue testing system it can be seen that the higher the testing frequency, the smaller the displacement of the testing grip. This is because it requires a lot of power to move mass (of the grip, specimen) at a high frequency. Hence:

\[ \text{Power} = \text{mass} \cdot \text{acceleration} \cdot \text{velocity} \]  

The power of the systems is limited, and therefore the displacement at a certain frequency as well. With the resonant systems this is different. The resonant systems have a very efficient way of applying stresses to the specimen. In fact they use the resilient capabilities of the material to apply the desired stresses. By the continuous transformation from kinetic energy to elastic energy of the specimen and vice versa, the energy is kept in the system. This means that only a small amount of energy has to be added to keep the system working. This makes the resonant systems very efficient.

So where the displacement of the non-resonant systems is limited by the power of the machine, is the displacement of the resonant systems limited due to other reasons. These limitations could be due to the specimen specifications (material, dimensions) or system specifications (resonance limits, control limits, dimensions).

3) Load cases

The fatigue testing systems are not all capable of applying every kind of desired load. Actually the systems are mostly designed for only one type of fatigue testing and can only apply other load cases if adjustments are made to the system. For instance special grips can be placed, which make it possible to apply three or four-point bending stresses to a specimen or torsion loads. Since this will add mass to the moving parts, it will cause the maximum displacement and/or frequency to be reduced.

A special case is the ultrasonic fatigue test by Bathias et al. The principle used in test setup can be used for other load cases as well, although large adjustments to the system have to be made then. The small scale multi-axial fatigue test by Straub et al. is the only setup that does not need physical adjustments to switch from bending loads to torsional loads. The only thing to be changed is the input signal for the piezo stacks.

4) Maximum force

As it appears from the data, there is a wide variety in maximum force of the systems as well. The MTS and the Rumul machines have a peak force of 20 kN, while the thin film fatigue test by Kim has only a maximum force of 5.8 N.
This enormous difference is mainly because of the machine purposes. The thin film fatigue test is specially designed for testing fragile films, so there is no need for high forces. The two high force systems are specially designed to apply such high forces. Specimens tested with these systems are bigger and more robust than the thin films tested with the low force machine of Kim. This gives the essence of the maximum force; the higher it is, the more applications the systems can be used for.

5) Power

As it appears from Table 1, the extreme performances of the MTS system require a high power machine. From this table it can also be deduced that servo-hydraulic systems in general cannot be considered as low-powered machines. They have the highest power consumption of all.

The systems that should do best with this respect are the resonant systems. Since energy in the system is transformed instead of getting lost, only low powered machines are needed to apply high stresses in the specimen. However, this is hard to verify since the machines are not designed to perform the same tests. In order to do further investigation to this new metrics are introduced, which are discussed in the next paragraphs.

6) Working range

In order to give a visual representation of the working range there is made a graph for each system which shows the relation between the frequency and the displacement. The graphs can be found in the appendixes. In every graph an area is colored to visualize the working range of the machine.

The surface of the area in these graphs can be calculated according to equation (1). This working range size gives an indication of the usability of the machine for tests at a different frequency or displacement. The exact numbers can be found in the sixth column of Table 1 and the result visually shown in Fig. L.7.

From these result it can be found that the working range of the MTS system is by far the largest. This system is designed to achieve high displacements at high frequencies. From the data it becomes also clear that there is an obvious difference between industrial an experimental fatigue test setups. Where the industrial machines are made to deal with a wide variety of tests, the experimental setups developed for a small variety of tests. This results in a lower working range for the experimental setups compared to the industrial ones.

Next to that it can also be gathered that the WRS is a useful metric for the comparison of fatigue testing systems. As described before, this metric can make the distinction between industrial machines and experimental setup. Since the industrial machines are designed for a wide variety of tests, it is self-evident that the working range among these machines is larger.

7) Fatigue Test Efficiency

The values of the FTE metric are calculated according to equation (1) and presented in Table 1 and Fig. L.7. Basically this metric combines the relevant characteristics of the oscillating motion generated by the systems, which then gives a measure of how efficient this is done.

From the resulting data it can be derived that the machine that does best with respect to this comparative value is the Rumul machine. Although the working range is not large, it is able to compensate this with a high force and a large displacement at low power consumption. These performances are achieved by making use of the elastic capabilities of the specimen through resonance.

Second best value with respect to the FTE metric is the 1000 Hz system of MTS. Due to a newly designed servo valve, this machine is able to operate efficient in comparison with other servo-hydraulic machines with an regular valve like the machine of Shimadzu.

From these results it cannot be concluded that resonating systems in general operate most efficient. Where the Rumul machine performs exceptional in this respect, the experimental resonating systems stay behind. However, the Bathias setup has a similar value as the lowest values for the industrial machines, but the differences are too small for drawing conclusions.

Something else that can be derived from the results of this Fatigue Test Efficiency metric is that it gives a useful number to compare the systems by. It combines the usability of the machines and methods with their efficiency and is therefore a valuable criterion in the comparison of the systems. The values can be considered to be significantly different and this metric recognizes the systems that perform above average in terms of working range or efficiency. This makes that the FTE can be considered as a useful tool in the decision process for fatigue testing equipment.

B. Test Allocation

There is a large variety of fatigue tests that can be performed by the test systems. The tests can differ in load case, number of cycles and specimen specifications. The characteristics of a system determine which tests can be done.
with this specific system. A table is created to summarize this in a clarifying overview. See Table 2.

In this table the boundary between big and small specimens is set to be 10 mm² for the cross-sectional area of the specimen. Specimens are called flexible if during the test the specimens are deformed more than 1% of their length. High cycle fatigue tests (HCF) apply under 10^7 load cycles, while very high cycle fatigue tests (VHCF) apply more than 10^7 load cycles.

In order to classify the different systems, they are put into the table according to the specimens they can test and the tests they can perform. The numbers in the table represent the test setups corresponding to the numbers in Fig. L.7. If the number is bold, it means that the setup is designed for this test, or operates best while performing the test. The non-bold numbers indicate which (other) machines can be used for the test represented by the box.

C. User Guidelines

When using this study in the decision process for fatigue testing equipment, it should be kept in mind that the systems evaluated in this report is only a small selection. This study is meant to be a guideline in selecting the kind of system and a corresponding working principle. The system that can be selected from this study stands for all other machines with the same basic principles of actuation. These other systems can be considered for the test as well. A similar procedure as in this study can be followed for further selection.

V. CONCLUSION

In this study it was found that there is a large variety of fatigue testing systems for a large variety of fatigue tests. To guide in the decision for a fatigue testing system an overview is presented, which consist of experimental setups as well as industrial machines. Their performances are examined and compared to each other and it can be concluded that there are large differences among fatigue testing equipment.

In this research various differences between the systems are discussed. The most relevant differences are found when the systems are grouped in industrial and experimental or resonant and forced vibration systems. The industrial machines are characterized by a large working range, the experimental setups by their unique or innovative way of testing. The resonant systems can apply large displacements at high frequencies and the forced vibration setups can operate at a wider range of frequencies. These fundamental differences should be kept in mind while considering the machines for a test. For all fatigue tests that are within the scope of this study, machines are selected that can execute the test according to their characteristics.

The two new metrics that are introduced in this study can be considered valuable in the comparison between fatigue testing systems. The metrics are able to recognize exceptional performances of the systems.

REFERENCES


Table 2 A fatigue test overview with an indication of possible test systems that are able to perform the test. Bold numbers indicate that the system is designed for the test represented by the box. The numbers correspond to Fig. L.7.

<table>
<thead>
<tr>
<th>Specimen against test type</th>
<th>Big</th>
<th>Small</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>Stiff</td>
<td>Flexible</td>
</tr>
<tr>
<td>Tension/Compression</td>
<td>HCF</td>
<td>9.5,6.8</td>
</tr>
<tr>
<td></td>
<td>VHCF</td>
<td>5.6,1</td>
</tr>
<tr>
<td>Bending</td>
<td>HCF</td>
<td>8.5,9</td>
</tr>
<tr>
<td></td>
<td>VHCF</td>
<td>1.5</td>
</tr>
</tbody>
</table>

[26] I. Klopfer, “Resonant testing machines, a very fast and economic way to perform high cycle fatigue (HCF) and fracture mechanics testing,” Atti del Congresso, IGF 19, pp 301-315, July 2007
[31] Shimadzu: http://www.shimadzu.de/, last access 12 June 2012
I. ULTRASONIC TEST METHOD FOR FATIGUE TESTING IN THE VERY HIGH CYCLE REGIME

This testing method is developed in order to test at a high frequency so testing time will be reduced to a minimum. These high frequencies are obtained by designing the whole system so that it resonates in one desired frequency. The result of this is that it is not possible to execute other test with this test setup. A new system has to be developed if another frequency is wanted that is not one of the natural frequencies of the system. Comparable systems have been developed [13]-[18]

II. SMALL SCALE MULTIAXIAL FATIGUE TEST METHOD

This method is developed in order to test the bending and torsion capabilities of a test sample. It uses the natural frequency of the sample to increase the movement so that the wanted stresses in the material can be obtained. This means that this method as it is shown here is only able to test at one frequency. However, by adjusting the sample dimensions other frequencies are possible to obtain. [19]
III. MICRO-SCALE TENSION FATIGUE TESTING METHOD

This fatigue testing setup has the same basis as the previous test setup, which is also developed at Karlsruhe Institute of Technology. Where the previous setup is used for torsion and bending, this setup is used for tension fatigue testing. By adjusting the mass there can be made a variation in the testing frequencies. In order to obtain high frequencies, the specimen is small. The cross section is only 0.03 mm$^2$. [20]

<table>
<thead>
<tr>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Developer</strong></td>
</tr>
<tr>
<td><strong>Working principle</strong></td>
</tr>
<tr>
<td><strong>Load cases</strong></td>
</tr>
<tr>
<td><strong>Frequency range</strong></td>
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<tr>
<td><strong>Max. displacement</strong></td>
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<tr>
<td><strong>Max. frequency</strong></td>
</tr>
<tr>
<td><strong>Min. frequency</strong></td>
</tr>
<tr>
<td><strong>Maximum Force</strong></td>
</tr>
<tr>
<td><strong>Power</strong></td>
</tr>
</tbody>
</table>

IV. VOICE COIL SETUP FOR THIN FILM FATIGUE TESTING

This fatigue test setup is developed in order to test thin films under tension-tension loading. This means that the specimens are small and the system does not need to be powerful to apply the desired forces and stresses. The basis of this fatigue testing setup is a low powered electrodynamic shaker. [21]

<table>
<thead>
<tr>
<th>Specifications</th>
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<tbody>
<tr>
<td><strong>Developer</strong></td>
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<td><strong>Max. frequency</strong></td>
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<td><strong>Min. frequency</strong></td>
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<tr>
<td><strong>Maximum Force</strong></td>
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<tr>
<td><strong>Power</strong></td>
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</table>
V. MTS SERVO-HYDRAULIC FATIGUE TESTING SYSTEM

This servo-hydraulic driven system is designed by MTS Systems Corporation. This high power machine is able to produce a fairly large stroke over a large range of frequencies. The system is adjusted in cooperation with Michigan Technological University in order to increase the frequency range to 1000 Hz while the forces that can be applied are still high. This required modifications to the frame as well as a new servo valve (schematically shown). [22]

![Servo-Hydraulic System Diagram]

VI. FATIGUE TESTING ON AN ELECTRODYNAMIC SHAKER

For some fatigue tests an industrial shaker is used as a basis. On top of the shaker the actual test is build. An example of a test rig on top of a shaker is shown in the picture, where a specimen is put between two masses. The masses together with the vibration introduce the stresses in the material. In fatigue testing shakers are more often used as a basis. Although specifications can vary among shakers, this shaker is set as a model for other shakers. [23]-[24]

![Electrodynamic Shaker Diagram]

**Specifications**

<table>
<thead>
<tr>
<th></th>
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<tbody>
<tr>
<td>Working principle</td>
<td>Servo-hydraulic</td>
</tr>
<tr>
<td>Load cases</td>
<td>Tension and Compression, Bending,</td>
</tr>
<tr>
<td>Frequency range</td>
<td>1 - 1000 Hz</td>
</tr>
<tr>
<td>Max. displacement</td>
<td>Peak to Peak</td>
</tr>
<tr>
<td>Max. frequency</td>
<td>0.1 mm</td>
</tr>
<tr>
<td>Min. frequency</td>
<td>50 mm</td>
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<tr>
<td>Maximum Force</td>
<td>20 kN</td>
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<tr>
<td>Power</td>
<td>18.5 kW</td>
</tr>
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</table>

**Specifications**

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<tr>
<th>Developer</th>
<th>Data Physics corporation – BAM: Federal institute for material research and testing</th>
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<tbody>
<tr>
<td>Working principle</td>
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<td>Load cases</td>
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<tr>
<td>Min. frequency</td>
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<tr>
<td>Maximum Force</td>
<td>650 N</td>
</tr>
<tr>
<td>Power</td>
<td>500 W</td>
</tr>
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</table>
VII. Bose Electromagnetic Testing Machine

The Bose ElectroForce is a widely used machine. The ease to use, the flexibility and its small size and weight make this a popular machine. The maximum performances on the other hand lag behind the other machines. The Machine works on the same principle as a loudspeaker. It can be used for a wide range of tests for bending as well as tension and compression material tests. [25]

<table>
<thead>
<tr>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Developer</strong></td>
</tr>
<tr>
<td><strong>Working principle</strong></td>
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<tr>
<td><strong>Load cases</strong></td>
</tr>
<tr>
<td><strong>Frequency range</strong></td>
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<tr>
<td><strong>Max. displacement</strong></td>
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<tr>
<td><strong>Max. frequency</strong></td>
</tr>
<tr>
<td><strong>Min. frequency</strong></td>
</tr>
<tr>
<td><strong>Maximum Force</strong></td>
</tr>
<tr>
<td><strong>Power</strong></td>
</tr>
</tbody>
</table>

![Bose ElectroForce](image1)

VIII. Rumul Magnetic Resonance Fatigue Test

The Rumul Mikrotron test machine works on a magnetic basis supported by a resonance system. The magnet (1) in the system gives a pulse to the system so that the natural frequency can be determined. After that the magnet pulses with this frequency so the whole system is vibrating in its natural frequency by the mass (4) and springs (3). This means that the machine applies high forces, with low energy consumption. The range of frequencies is fairly limited. [26]

<table>
<thead>
<tr>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Developer</strong></td>
</tr>
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<td><strong>Working principle</strong></td>
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<td><strong>Load cases</strong></td>
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<tr>
<td><strong>Frequency range</strong></td>
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<td><strong>Max. displacement</strong></td>
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<tr>
<td><strong>Max. frequency</strong></td>
</tr>
<tr>
<td><strong>Min. frequency</strong></td>
</tr>
<tr>
<td><strong>Maximum Force</strong></td>
</tr>
<tr>
<td><strong>Power</strong></td>
</tr>
</tbody>
</table>

![Rumul Mikrotron](image2)
IX. SHIMADZU SERVO-HYDRAULIC PULSER

This is a servo-hydraulic driven system designed by Shimadzu Scientific Instruments. This high power testing system can be used for multiple test setups. It is capable of applying loads to the specimen dynamically as well as statically. The testing setup is a regular servo-hydraulic fatigue test machine. In this report it is set as a model for other servo-hydraulic systems. Specifications can vary among these systems. [29]-[31]

<table>
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<td>Working principle</td>
<td>Servo-hydraulic</td>
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<tr>
<td>Load cases</td>
<td>Tension and Compression, Bending,</td>
</tr>
<tr>
<td>Frequency range</td>
<td>1 - 200 Hz</td>
</tr>
<tr>
<td>Max. displacement</td>
<td>Peak to Peak</td>
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<tr>
<td>Max. frequency</td>
<td>0.02 mm</td>
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<tr>
<td>Min. frequency</td>
<td>50 mm</td>
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<tr>
<td>Maximum Force</td>
<td>1 kN</td>
</tr>
<tr>
<td>Power</td>
<td>5 kW</td>
</tr>
</tbody>
</table>

![Diagram of the testing machine](image)