Realization of Ultrasonic Transducer Test Set-Up

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Report no: ME 10.027
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Specialisation: Mechatronical system design
Type of report: Master Thesis
Date: 23 June 2010
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Acknowledgements

Soon after I returned safely home after being abroad, the opportunity was given to me to start an internship at ITEC. I did not plan to move at all, but now, a year later, I am still here in Nijmegen and will even stay here to continue my research.

ITEC, and especially Thomas Porck, showed me all aspects of being a true mechatronical engineer. From top down I could make my first mechatronical design, which probably will have the highest bandwidth I will ever design. Realizing this system could not have been done without the support of Thomas, which I’m grateful for. I also want to thank Jasper Wesselingh for his critical view, especially in reviewing my report.

Finally I should be thankful towards my friends, family and in particular my parents. Even though I was hopping from place to place, I was always welcome in Nijmegen, Delft and Ambacht. It is great to feel home in three different places.
Summary

One of the processes in the semiconductor back-end industry is ultrasonic (US) wire bonding. This process comprises the formation of a wire connection from the integrated circuit to the leads of the package. A producer of these wire bonders is Itec, a subdivision of NXP. Itec is specialized in making high speed wire bonders for bulk products. It has developed the Phicom, which can produce 36k standard two wire products per hour.

However to stay competitive the Phicom should increase its throughput to 48k products per hour. With the current architecture of this machine, the prospect to reach this goal is minimal. The large moving mass of the ultrasonic (US) transducer, which is the bonding instrument, is one of the limiting factors. For bonding a two wire product it takes approximately 80% of the total cycle of 100 ms to move this instrument in X, Y and Z direction, where the actual bond time is the remaining 20 ms. A second problem of the transducer is the lack of monitoring options, which is a result of the resonance principle. The only knowledge available is based on quality measurements of the bonds produced by the current 65 kHz bond process.

New research is showing the benefits of using higher ultrasonic frequencies. This would not only result in a speed increase, but a quality improvement as well. This information together with the increased throughput requirement of the machine leads to a demand for a new transducer design. Preliminary to this new transducer design, an experimental set-up is to be designed with a variable frequency of 50-200 kHz to investigate the bond process on the fly in order to find bond optima.

Currently a motional amplitude of ±300 nm is used in bonding. At higher frequencies the amplitude can be scaled down linearly with the frequency to obtain the same bond energy. However to investigate higher energy levels as well in order to speed up the bond process, the requirement was set to have an amplitude of ±1 µm on the full frequency range.

This report deals with the realization of this 50-200 kHz experimental transducer test set-up (TTS). Since it is desired to have various bonding frequencies an off-resonant principle was selected. This gives the opportunity to ‘scan’ the frequency range and have good monitoring options with respect to the bond process. The reason for this is a constant amplitude independent of the bond load, due to high actuation force of the TTS compared to a resonator.

For the TTS it was chosen to use piezo actuators for their high dynamic bandwidth. The selected actuators are used in a symmetric push pull configuration, to create a standalone vibrating system and thereby avoiding influences from the environment. The
actuator moves the weld tool, i.e. capillary, with its holder by pushing against a weakly supported defined mass.

The design of the TTS is verified by means of a finite element model (FEM). This model is showing a first resonance of the capillary at 47 kHz, where the push pull mode of the TTS is located at 168 kHz. In the desired frequency range the response of the capillary tip showed an amplitude above the specified 1 µm. The presence of the resonances in this domain did not prevent this.

Once the feasibility of the TTS was proven by the FEM model, the design was realized. By means of a laser vibrometer the frequency response of the TTS was measured. The measurement showed similar modes as modeled. However the modes of the vibrating system were found at a factor 0.7 of the calculated frequencies, which corresponds to a stiffness decrease of a factor two. By a more detailed analysis of the holder 10% of the loss can be explained. Other causes have not yet been found, although a presumption is made towards the contact stiffness of the holder.

The off-resonant amplitude of the capillary tip in the frequency response showed a level of 5nm/V, which is only 25% of the modeled value. This could be for the most part attributed to non-linear behavior of the actuators. Time domain measurements at higher voltage show already an increased stroke. Still half of the stroke was lost by degradation of the actuator and asynchronous operation of the actuators due to hysteresis. Nevertheless the remaining stroke is compared to the current bond settings enough to be used for wire bonding.

Since the response shows clear resonance peaks in the range of interest, an inverse of system response is to be used as a filter. Thereby the chance to damage the sensitive actuators is minimized. Then the TTS is ready to be used in the near future to find out the optima in the bond process.
1. Problem description

1.1 Background

NXP is a semiconductor company founded by Philips more than 50 years ago. The company operates in the back-end side of the semiconductor industry, meaning their core business is packaging and testing of integrated circuits (ICs). For the production of standard electronic products like diodes and transistors a single assembly line is used. In this line the incoming wafers are processed to become fully tested and ready to be used single chips.

This thesis deals with one of the machines of this assembly line, which is a wire bonding machine. This machine shown in Figure 1 is called Phicom and is developed by ITEC in Nijmegen. It is used for creating gold or copper wire connections between the electrodes on the die and the connection pins i.e. leads. In Figure 2 an example of a bonded transistor is shown. The two wires on top of the die are the connections to the base (right pad) and the emitter (left pad). The bottom side of the die makes the third contact to the collector.

The present Phicom is able to make 36k products per hour in case of two wire products, like the transistor of Figure 2. The formation of these wire connections is achieved by ultrasonic welding under elevated temperatures, also called thermosonic bonding. To generate this welding vibration an ultrasonic (US) transducer is used as an instrument shown in Figure 3. This device can be seen as the heart of the machine.

Figure 1: PhiCom wirebonder

Figure 2: bonded die (leads are not shown)

Figure 3: ultrasonic transducer
The US transducer consists of a piezo stack (1) that converts electrical energy in a mechanical vibration in the form of a compression wave. By actuating this instrument in its eigenfrequency of typically 65 kHz, the amplitude of the vibration builds up in the system, due to the interaction of electrical and mechanical energy in the piezo stack. The produced vibration is then amplified by a mechanical horn (2) and transferred to the capillary (3). This capillary is the weld tool by which the gold wire, which runs through this tool, is bonded to a surface i.e. bond pad. The last important component is the clamping bush (4), which connects the resonator to the machine.

Although it looks as if this clamping bush is a solid cylinder, in fact this cylinder is only attached to the horn with a thin diaphragm, to avoid energy loss. It is held exactly at a node of the waveform as can be seen in Figure 4 by the vertical line. The compressive waveform is indicated by the red dots, where the blue line is an orthogonal projection of the mode. This mode, belonging to this typical frequency of around 65 kHz, is the so-called $3/4 \lambda$ mode, referring to the $3/4$ sine wave from the fixation to the tip. This tip thus corresponds to an anti-node, where the amplitude is maximal. By the tapering of the horn this amplitude is increased by a factor 5. By this amplification and the use of resonance the initial displacement of the piezo stack of typically 1-10 nm is increased to typically 300-600 nm.

Figure 4: $3/4\lambda$ waveform of US transducer. The compressive waveform is indicated with the red dots, where the blue line is an orthogonal projection of the waveform.

Besides the ultrasonic process other handling steps are performed in order to make a wire bond. The complete bond cycle, the variance in the process and actual bonding measurements can be found in Appendix A.
1.2 Problem statement

Until today ITEC never examined the fundamental physics of wire bonding. First of all the US transducer is purchased from an external supplier and secondary enough improvement in speed could be made by redesigning other machine elements. Recently this approach is changed due to three developments:

- Benchmarking and literature surveys show that higher ultrasonic frequencies have a positive effect on quality, speed and yield of the wire bonder. A drawback is however the decreased process window. In Appendix B.1 the main observations concerning this development can be found.

- The mass of the current transducer and its supporting geometry cannot be reduced much further, implicating the maximum possible accelerations are already achieved for the current design and thereby the speed as well. In Appendix B.2 the limitations towards the mass of the transducer assembly on the current Phicom can be found.

- There is a demand to monitor the ultrasonic movement to address quality issues. When using higher frequencies this monitoring becomes even more important due to the limited process window

At the moment the bond time is typically 5 ms and the complete bond cycle including 4 bonds takes 100 ms as shown in Appendix A.1. By developing a lighter transducer which operates at higher ultrasonic frequencies, both bond time and handling time can be reduced. This new design should make it possible to make the next step by increasing the speed from 36k to 48k, which corresponds to a cycle time of 75 ms. However at the moment the level of expertise in the wire bonding process is not yet sufficient to develop an own instrument. Therefore more knowledge is required, especially concerning the way the mechanical energy should be brought in the bond.

1.3 Project description

To gain more knowledge on the subject of the wire bond process, it is useful to do a study in the physical phenomena taking place during bonding. Only then explanations can be given regarding the required energy, time and process conditions to make a solid wire connection. This analysis can be qualified as a thorough process study, which requires expertise in for instance micro welding and material behavior.

A first step in this investigation is to design an experimental setup that can determine the parametric relations and possible optima in the bonding process. This project will be about the realization of this experimental setup. This includes the process of designing, manufacturing and testing this device. The setup should be able to make actual bonds in the frequency range of 50 to 200 kHz with an amplitude of 1 µm. Furthermore it should
have the ability to monitor and control most important bond parameters. With the results of this variable frequency transducer test setup (TTS), a recommendation can be done relating to future transducer designs. The project description can thus be formulated as:

**Design, build and test an US transducer experimental set-up, which can achieve motions from 50-200 kHz at 1μm amplitude**

### 1.4 Outline of thesis

In chapter 2 the requirements regarding the working range of the bonding process are described. Based on these requirements architectural choices for the TTS are made in chapter 3. Then in chapter 4 the first calculations are done to investigate the feasibility of the conceptual design. Chapter 5 and 6 will describe the final design of the TTS. Where chapter 5 discusses the different mechanical parts, chapter 6 comprises the finite element modeling of the TTS. In chapter 7 the FEM model is compared with actual measurements performed on the TTS. Finally in chapter 8 the results are evaluated to make a conclusion and give recommendations for a new transducer design and further research.
2. System requirements
In order to have the ability to bond, the TTS is subjected to a number of requirements. These requirements are partially derivatives from the current bond process and supplemented by a number of additional specifications.

2.1 Important parameters
In the present thermosonic bond process as described in Appendix A, 5 parameters are found as most important regarding the quality of the bond. With the exception of the frequency all parameters can be varied in a broad range. In Table 1 this range is specified followed by typical process settings.

Table 1: Current process settings in Phicom

<table>
<thead>
<tr>
<th>Variable</th>
<th>Range</th>
<th>Typical process setting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>[kHz]</td>
<td>65</td>
</tr>
<tr>
<td>Temperature</td>
<td>['°C]</td>
<td>20-400</td>
</tr>
<tr>
<td>Bond force</td>
<td>[N]</td>
<td>0-2</td>
</tr>
<tr>
<td>Amplitude</td>
<td>[µm]</td>
<td>0-±1</td>
</tr>
<tr>
<td>Bond time</td>
<td>[ms]</td>
<td>0-20</td>
</tr>
</tbody>
</table>

2.2 Requirements

- Adjustable frequency of 50-200kHz

To find an optimum in frequency this variable this parameter should be variable. In the problem description the requested frequency range is set to 50-200kHz. This range follows from the current 65 kHz operating frequency, supplemented with higher frequencies based on the research on high frequency bonding, explained in Appendix B.1. Although there are already ultrasonic micro welding systems using frequencies of e.g. 330, 600 and 780 kHz[1], these do not have the possibility to vary the frequency. With that aspect in mind the 200 kHz is a first challenge that can be adjusted up or downwards in a later stage.

- Mechanical amplitude of 1µm

A typical amplitude for bonding at 65 kHz is 300 nm. Higher US frequencies require a decreased amplitude to obtain the same energy level. From the transferred mechanical energy approximation following in §3.1.6 it turns out that the amplitude is inversely proportional to the frequency in order to have the same level of energy. However to investigate higher energy levels as well in order to speed up the bond process, the requirement was set to have an amplitude of 1 µm on the full frequency range.
• Designed for Phicom

In order to make use of the supporting bonding facilities of the Phicom during testing it is required that the TTS fits in and can be mounted to the Phicom. Any mechanical changes to realize this may not disturb the bonding process and should not be time consuming. In Figure 5 the Phicom is shown without the transducer. The yellow interface of the transducer yoke can be used for the TTS. If necessary the camera in front can be disassembled. The spark electrode at the right cannot be removed. Thereby there is only free space available at the left side of the view of Figure 5.

![Figure 5: Phicom front view, showing the interface of the transducer yoke on which the TTS can be mounted.](image)

• Use of replaceable capillary

In the complete bonding industry one makes use of capillaries as for guiding and welding the wire. Although it is possible to redesign this tool as well, a requirement for the TTS is to stick to this proven technology. Thus the constraint that follows is that the system can hold and vibrate a common capillary. Furthermore it should be possible to replace this tool, since it wears out during operation.

• Withstand thermal load

The bonding process is qualified as a thermosonic process. The elevated temperatures may not cause dysfunction of the ultrasonic process. This requires an actuator and sensor that can withstand the environmental heat.
- **Requirement duty cycle**

The present Phicom is able to make 36k two-wire-products an hour. This means a cycle time of 100 ms. In this time 4 bonds are made to make the two connections. Based on the current bond times of around 4 ms the current duty cycle of the transducer is:

\[ \eta = \frac{4 \cdot 4e^{-3}}{100e^{-3}} = 16\% \quad (2.1) \]

To obtain 48k operation the cycle time is reduced to 75ms resulting in a duty cycle of 21\%. For the commercialization of a future transducer this duty cycle is an absolute requirement. For an experimental setup this requirement can be dropped. Then it is sufficient if 1 bond per second can be made.

- **Requirement lifetime**

For lifetime the same consideration holds as for duty cycle. In case of commercialization it will be important. Now the TTS should be able to survive half a year of testing. Therefore it does not require to be fully damage proof, still it is required that the TTS is evaluated on its mechanical stability.

- **Monitoring of bond parameters**

An important aspect of the TTS will be that all parameters mentioned in §2.1 can be monitored. Bond time, bond force and temperature are already monitored in the Phicom. The missing parameters relating to the ultrasonic process should be monitored with new equipment and software. For the amplitude of motion it is required that it can be monitored on the capillary as close as possible to the bond, without obstructing the vibration. Only this will give valuable information relating to the bond process.

- **Easy to produce and assemble**

Since the time and money for this project is limited, the TTS should be easy to produce and assemble. Production techniques like drilling, milling etc. are preferred and complex geometries requiring time consuming techniques as e.g. 3d prototyping should be avoided. This way the delivery time can be kept short.

Regarding assembly it is important that the TTS can be assembled and disassembled easily, without being damaged or changing a setting. This holds for the assembly to the Phicom, but as well for the assembly of the different components in the TTS, such as actuators, capillary etc. It is likely that these assembly steps need to be done more frequent for an experimental set-up than for a commercial device. To avoid spending days on assembly a simple assembly is required.
3. Architecture

In this chapter a system design is introduced based on the requirements of Chapter 2. The most challenging claim is to get an amplitude of 1 µm at 50-200 kHz. Hence, the way of actuation plays a crucial part in the design. Therefore the first paragraph is dedicated to the selection of a suitable actuator. This comprises the choices whether or not to use resonance, which kind of actuation principle to be used and how to drive the actuator. Then paragraph §3.2 and §3.3 deal with respectively the sensor and control design. Finally on the basis of the selected actuators and sensors the mechanical architecture is worked out in §3.4. In Figure 6 it can be seen how the architecture comprises the different elements of a mechatronical system.

![Figure 6: Schematic system architecture](image)

3.1 Actuator Design

3.1.1 Resonant versus off-resonant actuation

The current design is based on a resonant drive system. This resonating system has always been a logical approach to create a vibrating system. However there are some good reasons as well to work off-resonant, which will be explained in this paragraph. Then a complete different architecture is necessary. To make a good choice for the new setup a comparison of both is done based on Figure 7.
The resonant principle is based on adapting the excitation frequency to the mechanical resonance frequency of the system. Then the impedance of the electromechanical system is minimal and purely resistive. Tuning the system to this frequency is usually done using a phase lock loop (PLL). The phase shift occurring at the eigenfrequency of the system results in the frequency lock. At this frequency the system starts to resonate. Every cycle energy is put in the system, which is stored in the relative large vibrating mass. This way the energy is build up exponentially which makes it possible to get a high energy level with only a low voltage level. The drawback is that is takes time before the set point is reached. The time constant \( \tau \) of this system can be defined as a function of the resonance frequency \( \omega_0 \) and the damping ratio \( \zeta \) according to equation (3.1)

\[
\tau = \frac{1}{\omega_0 \cdot \zeta}
\]  

(3.1)

When the set point is reached there is equilibrium between the input energy and the dissipation. This dissipation is mainly coming from mechanical losses.

The resonant system is thus very energy efficient. However the controllability is poor, because of the slow response and the sensitivity of the resonator.

This sensitivity expresses itself in case of an external load. Contact of the resonator to the fixed world via the bond will cause the resonance frequency of the system to shift a bit, due to the introduced mass, damping and stiffness of the contact. By using the PLL the excitation frequency is shifting with the resonance frequency, to prevent a large drop in amplitude. However the shifted resonance peak is not of the same amplitude as before. The cause of this change is first of all the dissipation at the load. Secondary there is also dissipation at the fixation of the resonator to the outer world. To avoid this loss this fixation is designed to be on a node of the waveform. However the changed
frequency shifts the node of the waveform causing energy dissipation after all. Monitoring of the load during contact is therefore difficult in resonant systems, since it is not clear what the contribution of the load is and what can be related to the changed resonator.

Furthermore a resonator is sensitive towards disturbance forces. The actuation force is small, such that the disturbance forces are more significant. A constant opposing force is evidently resulting in a reduced amplitude. For a sudden disturbance force the system is more robust due to the relative high vibrating mass, but lost energy still has to be restored.

**Off resonant**

Working off resonant means the impedance of the system is per definition higher than working resonant. This means the amplitude will only reach the same value as off-resonant if the excitation voltage is increased. This can be seen in Figure 7 as well. The result for a piezo actuator is high dielectric losses and high resistive losses due to the large current that will flow. There is thus a high energy requirement.

Despite the high losses, the big advantage of the off resonant system is the possibility to use any frequency in the range of the actuator, instead of a fixed resonance frequency. With a function generator it is easy to adjust this frequency anytime.

Furthermore the high force of the actuator gives a lot of opportunities. There is no build up time required thus it is possible to make relative large steps in amplitude. This gives then again the possibility to control the vibration better, since it can be switched on, switched off or adjusted when desired. Furthermore the process is easy to monitor. A decrease in amplitude can be directly related to a load, since the losses in the actuator remain the same. This load sensitivity of the system is thus much less.

To summarize, the differences of both principles are listed again in Table 2.

<table>
<thead>
<tr>
<th><strong>Table 2: Comparison resonance vs. off-resonance drive</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Resonant</strong></td>
</tr>
<tr>
<td>Low voltages</td>
</tr>
<tr>
<td>Low energy dissipation</td>
</tr>
<tr>
<td>Mechanical dissipation dominant</td>
</tr>
<tr>
<td>Mass required to store energy</td>
</tr>
<tr>
<td>Build up time required</td>
</tr>
<tr>
<td>Single frequency</td>
</tr>
<tr>
<td>Use of PLL</td>
</tr>
<tr>
<td>Low controllability</td>
</tr>
<tr>
<td>High load sensitivity</td>
</tr>
<tr>
<td>Hard to monitor load</td>
</tr>
</tbody>
</table>
For the TTS an important criterion is the variable frequency. Based on the comparison of Table 2 one might have the impression the resonant principle is thereby already excluded. However it is possible to design a tunable resonant transducer. For future designs this option should be considered. However for the TTS controllability of the different parameters, like the motion amplitude is essential to find the parametric relations in the bond process. Furthermore a high load resistance is preferred. Concerning these requirements the off-resonant principle is advantageous over the resonant principle. The high energy demands relating to this off-resonant drive are for this experimental design less important as long as it is still feasible to supply the required power. Therefore the choice is made for an off-resonant actuator.

### 3.1.2 Actuator type

An off-resonant actuator type has to be selected, meeting the requirements:

- Drive frequency of 50-200kHz
- Motion amplitude of 1µm

For conventional actuators like voice coil actuators and stepper motors, these specifications are not obtainable. Therefore the response time is too slow [2]. Miniaturizing this type of motors leads to problems of efficiency [3]. Due to the complex coil structure a voice coil motor has a minimal size. Making it smaller results in winding problems and Joule heating [4].

More suitable are the relatively new smart actuators. This type of actuator makes use of the material property to convert an input signal to a force or displacement. One of these smart actuators is the piezo actuator. In Appendix C the piezoelectric principle and the applicability of these principles in an actuator are extensively discussed. Due to the high stiffness and fast response it a piezo actuator is suitable for short stroke high dynamical applications. Based on the catalogues of different piezo actuator suppliers [5][6] this high bandwidth is verified. These suppliers sell actuators with resonance frequencies > 500 kHz.

Drawbacks of the piezo actuator are the high aging rate and the large hysteresis. Although there is no need for accurate positioning, the energy loss relating to the hysteresis should be inspected. Aging will be a point of attention in case the actuator will be commercialized and lifetime becomes important.

### 3.1.3 Dimensioning actuator

A piezo actuator can be ordered in different sizes, shapes and actuation directions. Furthermore it is possible to have a piezo actuator operate in diverse modes: linear, shear or bending. Figure 8 will show some examples how the capillary can be driven using these different actuation modes.
To make a good decision, the performance of the different type of actuator should be analyzed. A trade off in actuator type is made based on existing actuators. From the catalogue of PI (Physik Instrumente) [5] the tube and bender actuators turn out to be unsuitable for the design. Both actuators are used for relative high displacements of more than 10 µm and have large dimensions, which results in low stiffnesses and eigenfrequencies. These actuators are clearly not suitable for high dynamical purposes.

For the shear and linear actuators a closer look is taken into the different properties. In Table 3 the smallest actuators are listed to make the comparison. From here the choice is made for the smallest linear piezo actuator (P-022.30). In the next paragraphs some properties of the actuators are worked out leading to this decision.

Table 3: comparison shear and linear piezo actuators [7]

<table>
<thead>
<tr>
<th>Order nr</th>
<th>Units</th>
<th>P-112.01</th>
<th>P-112.03</th>
<th>P-022.30</th>
<th>P-033.30</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuation mode</td>
<td>Shear</td>
<td>Shear</td>
<td>Linear</td>
<td>Linear</td>
<td></td>
</tr>
<tr>
<td>Displacement [µm]</td>
<td>1</td>
<td>3</td>
<td>2.2</td>
<td>2.2</td>
<td></td>
</tr>
<tr>
<td>Nominal voltage [V]</td>
<td>±250</td>
<td>±250</td>
<td>100</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Capacitance [nF]</td>
<td>0.5</td>
<td>1.5</td>
<td>25</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Cross section [mm x mm]</td>
<td>3x3</td>
<td>3x3</td>
<td>2x2</td>
<td>3x3</td>
<td></td>
</tr>
<tr>
<td>Length [mm]</td>
<td>3.5</td>
<td>5.5</td>
<td>2</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Blocking force [N]</td>
<td>20</td>
<td>20</td>
<td>&gt;120</td>
<td>&gt;300</td>
<td></td>
</tr>
<tr>
<td>Derived stiffness in actuation direction [kN/mm]</td>
<td>20</td>
<td>7</td>
<td>60</td>
<td>130</td>
<td></td>
</tr>
<tr>
<td>Resonant frequency [kHz]</td>
<td>330</td>
<td>210</td>
<td>&gt;300</td>
<td>&gt;300</td>
<td></td>
</tr>
<tr>
<td>Max operational temperature ['C]</td>
<td>85</td>
<td>85</td>
<td>150</td>
<td>150</td>
<td></td>
</tr>
<tr>
<td>Number of cycles [-]</td>
<td>&gt;10⁶</td>
<td>&gt;10⁹</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Preload [MPa]</td>
<td>-</td>
<td>-</td>
<td>15</td>
<td>15</td>
<td></td>
</tr>
</tbody>
</table>

**Stroke**

Based on Table 3 the linear actuator has a higher displacement per volt even though the shear piezos have a larger length. For 1 µm a linear piezo requires 100 V, where the
The smallest shear piezo needs a voltage range of 500 V. This is a high specification for an amplifier and on top of that it poses the danger of electric shock.

It is also possible not to use the full stroke. However in case of the shear actuator a larger variant is necessary to still get a µm amplitude. Scaling up the geometry results in lower stiffnesses and thus lower eigenfrequencies. Knowing that the smallest shear piezo already has a lower stiffness and eigenfrequency than the linear one, makes this modification not a good decision.

A remarkable detail is that according to the material properties of Lead Zirconate Titanate (PZT), a standard piezo material, larger displacement can be obtained with the same electric field. This can be concluded from the fact that the shear component of the piezo stress coefficient can be up to 3 times higher than for the linear direction [8]. However it turned out from PI that the manufacturing process plays a role in this. The linear actuators are stacked monolithically. For that reason the electric fields that are obtained in this configuration are higher than in shear piezos. Shear piezos are made using another process resulting in a lower maximum electrical field. However due to the thicker layers the voltage required to get the same motional amplitude is higher in the shear piezos.

**Blocking force/stiffness**
The blocking force of the linear piezo is much higher than the one of the shear piezo. The smallest linear piezo actuator has a blocking force of 120N. This is more than a factor 100 larger than the axial load. This load can be approximated as a fraction of the bond force, which is typically 400 mN, see Appendix A. With a blocking force of 120 N the axial movement is barely disturbed. Based on the blocking force and the stroke a stiffness is derived according to:

\[
k = \frac{F_b}{x}
\]

(3.2)

A high stiffness is desired for high frequent operation. The linear piezo turns out to be much stiffer, which is no surprise with a higher blocking force and a smaller geometry.

**Temperature resistance**
Another important property is the temperature resistance. The linear piezos are able to operate at 150˚C, where the shear actuators can only resist a temperature of 85˚C. This temperature should be related to the internal heat generation in the piezos. The heat in the piezo scales linear with the capacitance and quadratic with the voltage, according to equation (3.3), where the power converted to heat (P) is calculated as a function of loss factor (tan(δ)), frequency (f), capacitance (C) and peak peak voltage (V\text{pp}).

\[
P = \tan(\delta) \cdot f \cdot C \cdot V_{pp}^2
\]

(3.3)
The linear piezo has a much higher capacitance than the shear piezo, mainly because of the high amount of layers. However the linear piezo operates at lower voltages. It yields that the heat generation is still almost a factor 10 higher than in the shear piezo. Besides the internal heat generation the system also has to deal with the heat coming from the oven that can reach temperatures of 300˚C. Then the linear piezos provide a better solution although the operational temperature of 150˚C is low in comparison to the environmental temperature.

**Lifetime**

From the requirement it was not necessary to have a high duty cycle for the TTS. Thereby it is recognized that lifetime will not be much of an issue for any of these piezo actuators.

In case of commercialization it is recommended to do an endurance test, since the lifetime of a piezo actuator is hard to predict. It strongly depends on the (combination of) environmental conditions, like temperature, humidity and method of loading. If e.g. the operating temperature is exceeded this will not directly breakdown the actuator. This will only happen at the Curie temperature, where the piezo depolarize. Even so, the piezo loses some of its lifetime.

The specified number of cycles is thus chosen to be on the safe side by the manufacturer. Taking this number equals only 83 minutes at continuous duty at 200 kHz. Currently a test is done in Italy, where this 2x2x2mm PICMA actuator already made 5 billion \(5 \times 10^{12}\) cycles during 538 days excited with 20 V pk-pk at 100kHz [9]. During the test the temperature was elevated several days to 150 degrees. In the first billion cycles, the amplitude of motion was only decreased by 10%. Besides that the actuator was increased by 10% of this amplitude. Based on this, the lifetime has thus more to do with degradation of the actuator then dysfunction of it.

**Preload**

As a final point the required preload for the linear piezo actuator is mentioned. This can be a large drawback. Creating this preload using a spring can cause a decrease of amplitude. The preload spring is then working parallel to the internal stiffness of the piezo. To avoid this, a construction should be designed that provides a constant force.

**Conclusion**

Assuming a force preload can be realized and an amplifier can be found that can drive the high capacitance the linear actuator provides the best solution. Since it will require a lot of power and expensive equipment to drive a high load, the choice is made to try to meet the objectives with the small actuator (P-022.30). This will be positive for the electrical, but negative for the mechanical bandwidth. If necessary or desired this trade-
off can later on be restored by choosing a larger piezo actuator. Advantageous is that all PICMA actuators are of the same length, which makes it easy to interchange them.

3.1.4 Power supply

Piezo material acts electrically like a capacitor when operating. This means a piezo actuator will be a large load for a drive system at higher frequencies. For the chosen actuator of 25 nF an amplifier has to be found that can drive it up to a maximum frequency ($f_{bw}$) 200 kHz. The maximum operating voltage is specified as 100V. Dynamically the amplitude ($V_{max}$) can be 50 V, when driving the piezo symmetrically around the 50 V. Then the required slew rate can be calculated according to (3.4).

\[
\frac{dV_{max}}{dt} = 2 \cdot \pi \cdot f_{bw} \cdot V_{max} = 63 \cdot \left[ \frac{V}{\mu s} \right] \tag{3.4}
\]

The maximum current ($I_{max}$) required corresponding to this slew rate of 63 V/\mu s is:

\[
I_{max} = C \cdot \frac{dV_{max}}{dt} = 25 \cdot 10^{-9} \cdot 63 = 1.6[A] \tag{3.5}
\]

The required power will then be:

\[
P_p = \frac{1}{2} \cdot V_{max} \cdot I_{max} = \frac{1}{2} \cdot 50 \cdot 1.6 = 40W \tag{3.6}
\]

A series of amplifiers that meet these high specifications are the hybrid power amplifiers of Apex. In Table 4 the main specifications of this type of amplifier are listed. With this amplifier the voltage span and slew rate are fulfilled and it is even possible to drive multiple piezos on one amplifier looking at the maximum current. Furthermore this amplifier has a very high gain bandwidth product of 10 MHz.

Table 4: MP108[10]

<table>
<thead>
<tr>
<th>Model</th>
<th>Vpp Min</th>
<th>Vpp Max</th>
<th>Iout Continuous Max</th>
<th>Slew rate (V/\mu s)</th>
<th>Standby Current (mA)</th>
<th>Gain Bandwidth Product (MHz)</th>
<th>Maximum power dissipation (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MP108</td>
<td>30</td>
<td>200</td>
<td>10</td>
<td>170</td>
<td>65</td>
<td>10</td>
<td>100</td>
</tr>
</tbody>
</table>

To amplify a 10 Volt control signal to the required 50 V an amplification of 5 is necessary, reducing the no load bandwidth to 2 MHz. The drop of bandwidth due to a capacitive load on the output should be obtained by an electrical simulation model. Apex made a software tool to design its amplifiers [11]. Without any manipulation techniques the closure frequency equals 1333 kHz for 25 nF. This closure frequency is the frequency where the open loop gain becomes less than 5. For 50 nF this will be 1000...
kHz. However in order to damp the peaking of the amplifier a capacitor can be added in the feedback loop. This will reduce the bandwidth to 650 kHz as can be seen in Appendix G.1. This bandwidth is large enough to be used for the TTS.

The heat dissipation will be a larger point of attention. The amplifier is specified to dissipate maximally 100 W. In case of continuous drive of this capacitive load a fan can be added as a precaution to prevent overheating. However it is likely that not the amplifier, but the piezo actuators are the critical factor.

3.1.5 Dissipation of the actuator

In case the load of the opamp is indeed purely capacitive like it was assumed electrically, then the required power of 40 W is fully dissipated in the amplifier. However other elements dissipate or use energy as well.

The largest contributor to dissipation of the actuator is the energy loss related to the electromechanical conversion in the piezo defined by the dielectric loss factor \( \tan(\delta) \) of the piezo. For the present actuator this factor is specified as 20\% [7], when using it in its full stroke. A supply of 40W means a dissipation in each piezo in case of continuous duty of:

\[
Q = \tan(\delta) \cdot P_p = 8 \text{ W}
\] (3.7)

In contrast to the amplifier it is a lot harder to dissipate this amount of heat from a 2x2x2mm piezo stack. In case there is no proper heat transfer it takes 1.6 s before the maximum temperature of the actuator is reached according to (3.8) and (3.9). Thereby a heat capacitance of the actuator was based on a specific heat capacity of PZT of 350 J·kg\(^{-1}\)·K\(^{-1}\)[7].

\[
C_T = c_p \cdot m = 350 \cdot 62e^{-6} = 21mJ/K
\] (3.8)

\[
t = \frac{\Delta T \cdot C}{Q} = \frac{130 \cdot 21e^{-3}}{1.7} = 0.3s
\] (3.9)

The convection of these small surfaces is minimal, thus conduction to a heat sink is required to get rid of the heat. In [12] a thermal model is made based on the geometry worked out in the next chapters. This model showed a maximum possible duty cycle of 10%. For testing this is sufficient.

As a safety precaution to make sure the temperature is not exceeding the 150°C a temperature sensor should be build-in close to the piezos.
3.1.6 Dissipation in bonding

Other dissipating elements are related to the bonding motion. If a moving mass of 1 g is assumed, the average power can be calculated using the drive frequency.

\[
P = m \cdot v = m \cdot \frac{1}{\sqrt{2}} \cdot 2 \cdot \pi \cdot f \cdot A
\]

\[
P = 1e^{-3} \cdot \frac{1}{\sqrt{2}} \cdot 2 \cdot \pi \cdot 200e^3 \cdot 2e^{-6} = 2mW
\]

Then there is the energy that has to be transferred to the bond partially coming from the ultrasonics and partially from the temperature. A way to estimate this is based on the present bonding settings.

\[
E_{bond} = E_{mec h} + E_{therm}
\]

The mechanical energy transferred to the bond can be approximated by the mechanical friction during bonding. This friction occurs when the transducer is pressed down by the bond force \(F_z\). According to [13] an average bond force is 300 mN. From [14] it is known that these frictional contacts at small scales can only be described by involving the molecular forces that strongly depend on material and geometry. However for the order of magnitude it is sufficient to assume a standard friction coefficient \(\mu\) of 0.2.

\[
E_{mec h} = F_z \cdot v_{RMS} \cdot t = \mu \cdot F_z \cdot \frac{1}{\sqrt{2}} \cdot 2 \cdot \pi \cdot f \cdot A \cdot t_{bond}
\]

\[
E_{mech} = 0.2 \cdot 400e^{-3} \cdot \frac{1}{\sqrt{2}} \cdot 2 \cdot \pi \cdot 65e^3 \cdot 1e^{-6} \cdot 4e^{-3} = 92\mu J
\]

The thermal energy involved in the bonding process is assumed to be equal to heat up the gold ball with a diameter \(d\) to the set bond temperature \(T_{bond}\). The heat capacitance \(c_0\) of gold is 128[W·kg⁻¹·K]. In practice the die should be heated up as well to make a bond. But again to get an estimate the equation (3.15) is sufficient.

\[
E_{therm} = c_0 \cdot (T_{bond} - T_0) \cdot \rho \cdot \frac{4}{3} \cdot \left(\frac{d}{2}\right)^3
\]

\[
E_{therm} = 128 \cdot (300 - 20) \cdot 19320 \cdot \frac{4}{3} \cdot \left(\frac{60e^{-6}}{2}\right)^3 = 25\mu J
\]

The required power would then be:

\[
P = \frac{E_{bond}}{t_{bond}} = \frac{92e^{-6} + 25e^{-6}}{4e^{-3}} = 29mW
\]
It is clear all these energies relating to the bonding motion are vanished compared to the power dissipated by the actuator, showing the poor efficiency of the TTS.

3.2 Sensor design

Monitoring the process is crucial for this experimental setup. Otherwise it is not possible to capture all process parameters. Most of these parameters are not hard to get a hand on as long as they relate to an electrical quantity. Thus power, current, voltage and frequency can be easily obtained. The bond force generated by the Phicom is calibrated according to the current in its voice coil actuator, thus does not require more attention. The temperature of the process is measured as well by the Phicom.

The sensor that still has to be included in the design is a vibration sensor. For model validation the vibration should be measured on several spots, for example on the capillary tip. Therefore it is required to have a non contact sensor, which can measure on all these locations. A laser vibrometer is then a logical choice. In Appendix G.6 the specifications of the used vibrometer are listed.

For commercialization an integrated sensing option has to be included in the TTS. A suitable choice would be a capacitive sensor. Another interesting option is to make use of self sensing. It is possible to use an element of the piezo stack to sense the displacement, since the electrical current and mechanical displacement are linearly coupled in a piezo-actuator. The derived signal is however small and contains relative much noise. Therefore the signal has to be filtered and amplified to be used. This is a straightforward problem and is still under investigation[15].

With the selected sensing options, there is no need to take these sensors into account in the mechanical design, except for the fact that the laser should be able to reach the moving capillary.

3.3 Control

As mentioned in Chapter 2 the important bond parameters have to be controlled. However there was no urgent need for closed loop control, since the system is stable and predictable. By manually adjusting the bond parameters the desired output can be obtained as well.

Important in the control of the piezo actuator is to use charge control or voltage control. Main advantage of the charge control is that one does not suffer from mechanical hysteresis anymore, since the current is directly related to the amplitude as described in §C.3.3. The energy losses relating to the hysteresis are still present. These losses appearing in the relation between the electrical displacement and the electrical field and do not change by the way of controlling. What does change when selecting another
control type is the way the system behaves electrically and mechanically. In §C.3.3 it is shown that using charge control will create a mechanically stiffer system, but also the capacitance will increase in comparison to voltage control where the capacitance decreases.

Making a charge controller requires more effort than a voltage controller, since feedback loops are required to get a constant charge on the output. Also the output is noisier than in case of a voltage amplifier. Furthermore the phase loss is larger due to these feedback loops. For closed loop control a voltage source can therefore be more desirable. For open loop however charge control is more ideal. Therefore simultaneous to the realization of the TTS a charge amplifier for the system is developed [15]. As long as this amplifier is still under investigation a voltage controller is used. When both controllers are operational, they will be compared on their performance. Based on this comparison the most suitable amplifier is selected for the TTS.

In the future the possibility of closed loop control can be investigated to create an automatically controlled system. However controlling a mechanical system at frequencies up to 200 kHz requires fast control techniques with sampling frequencies higher than 1 Mhz. Therefore a suitable choice is to use analog electronics. This method gives the opportunity to use the full available voltage range, instead of just ones and zeros. Nonetheless, to maintain a high resolution signal to noise ratio’s are very important.

3.4 Mechanical architecture

Decisions are made towards the actuation, sensing and controlling of the TTS. However the design is still should still be filled in. This paragraph deals with the architectural choices of the TTS. Since high frequencies are required, the system should be very minimalistic in order to not excite undesirable resonance frequencies. In fact it is only the capillary, the weld tool, which has to move. Important is thus the choice of tool (3.4.1). Furthermore it has to be decided whether to drive this capillary linearly or rotational (3.4.2) and finally in what configuration this chosen motion can be obtained (3.4.3).

3.4.1 Capillary

A requirement is to use an industrial standard capillary as weld tool. Still there are many variants in size and shapes of the capillary, especially of its cone. In the present transducer the capillary that is used is an 11.1 mm capillary, see G.3 for the specification. Important parameter is the length of this tool, since enough reach is necessary to prevent the US transducer from touching the work holder. Furthermore 6 mm space is required for the spark unit as shown in Figure 9. This spark unit is an electrode which sparks to the wire to form a ball at tail of the gold wire as can be read in
Appendix A.1. To make sure the wire is sparked and not the transducer the criterion of 6 mm is used.

![Symmetric positioned capillary with 6 mm distance from spark unit to transducer](image)

Figure 9: Symmetric positioned capillary with 6 mm distance from spark unit to transducer

The current transducer is driving the capillary asymmetrically. In a light weighted system one might experience undesirable bending and torsional modes of the system. This effect can be reduced by choosing a longer capillary, which will be driven in the center of mass. The disadvantage of this longer capillary is the increase of mass. With the specified free length at least a standard 15.9 mm capillary is required for a symmetric drive. This means the mass will increase with a factor 1.4. The resonance frequency will thereby drop a factor 1.2 maximally.

### 3.4.2 Rotation versus translation

There are various ways to drive a capillary by means of a piezo. It turned out from the actuator choice that the linear actuator was having the better specifications. Although it is not obvious, this actuator can still be used for a rotational drive.

An advantage of a rotating system can be the transmission ratio in amplitude. With less force, thus less dissipation, the same stroke can be obtained as for the linear actuation. If the rotating point equals the center of mass, this force is three times less:

\[
\begin{align*}
F_{\text{lin}} &= m \cdot \ddot{x} \\
F_{\text{rot}} &= \frac{1}{12} m \cdot L \cdot \dot{\theta} \\
J &= \frac{m \cdot L^2}{12} \\
\frac{F_{\text{rot}}}{F_{\text{lin}}} &= \frac{\frac{m \cdot L \cdot \dot{\theta}}{6}}{m \cdot \ddot{x}} = \frac{L}{6 \cdot \tan^{-1} \left( \frac{2 \cdot x}{L} \right)} \frac{1}{x} < \frac{1}{3}
\end{align*}
\]

(3.18)  (3.19)  (3.20)  (3.21)

There are however some drawbacks as well. It is required to have a rotational hinge when only actuating the system at one spot. This hinge can introduce all kind of unwanted dynamics up till the frequency where it decouples in horizontal direction. Making the hinge weak in this direction will solve the disturbance problem, but enables
the capillary to make a linear movement as well in combination with a rotation. Controlling a system with these combined modes is not straightforward and is therefore not preferable.

![Diagram of rotation principle with hinge](image)

**Figure 10: Rotation principle with hinge**

To avoid a hinge one can choose for a two point actuation shown in Figure 11. Placing these actuators next to each other requires the least clamping space and will result in the largest rotation and amplitude at the tip.

This motional amplitude can be described by equation (3.22), where $L_u$ is the length of the capillary to the center of rotation and $L_{cor}$ is the distance of the piezo actuator to the center of rotation. The mechanical amplitude of the actuator is given by $A$.

$$x = \frac{L_u}{L_{cor}} \cdot A$$  \hspace{1cm} (3.22)

With two 2x2x2mm piezos placed against each other and a length of 8 mm of the capillary, the amplitude at the tip will then be:

$$\frac{8\text{mm}}{1\text{mm}} \cdot 1\ \mu\text{m/mm} = 8\mu\text{m}$$  \hspace{1cm} (3.23)

Practically this loading case will create a moment on both piezos, resulting in a non optimal stroke and possible delaminating of the actuators. Therefore it is required to separate both actuation points along the capillary.

However it is given that the end of the capillary should still be free. This means the actuators should move upwards such that the center of rotation does not match the center of gravity anymore resulting in a decrease of bandwidth. However an interesting aspect of this type of actuation type is the presence of over actuation. This can be used to actively damp modes of the capillary.
A possibility to maintain symmetry and still use a two point actuation is by actuating the capillary in vertical direction on a relative long stiff beam to avoid high moments on the piezo. Introducing this beam will also add some inertia, such that the rotational advantage is decreasing. Another point of attention is that the absence of a rotational hinge will give this system the possibility to make a linear movement as well when the actuators are not exactly symmetrically placed or driven. In the previous situation this could happen as well, but now it will create a downward movement and can damage the die.

Choosing for a linear driven system might require higher forces, it has some big advantages as well. A linear drive can be realized by only one point of actuation and maintaining symmetry. There are no hinges required and one only has actuation forces in movement direction. The functioning of such is system is thereby better guaranteed and therefore this concept is chosen.
3.4.3 Push-pull

In the schematic drawings of 3.4.2 the force arrows represent dynamical forces. To maintain contact with the capillary and to avoid tensile stress an opposing force is required. This can either be done by a preload spring or by a second piezo actuator.

**Preload spring**

In case of a preload spring the design will be according to Figure 14. In here the piezo actuator is drawn in blue.

\[ \Delta s = \frac{k_p}{k_p + k_s} s \]  \hspace{1cm} (3.24)

It is thus desirable to have a spring with a low stiffness such that the stroke is maintained and no high frequent dynamics will originate from it.

The selected actuator was specified with a blocking force \((F_b)\) of \(>120\text{N}\) and a derived stiffness \((k_p)\) of \(>60\text{kN/mm}\) (Table 3, 3.1.3). To stay away from the desired frequency range the resonance frequency of the system on the preload spring should stay below
50 kHz. In case the system on the piezo stiffness is placed at a 200 kHz frequency, the stiffness $k_s$ should be equal to 3.75kN/mm according to (3.25). This stiffness corresponds to a loss of stroke of 6% according to (3.24).

$$k_s < \left( \frac{f_s}{f_p} \right)^2 k_p = \left( \frac{50}{200} \right)^2 60 = 3.75 \text{kN/mm}$$ \hspace{1cm} (3.25)

In this calculation higher harmonics of the preload spring are not taken into account. For these higher resonances the preload spring stiffness should be decreased even further, to make sure that the system is properly decoupled from this preload spring.

Considering the stiffness this concept is still feasible. However the problem of it is the introduced added mass, which is causing a drop in resonance frequency. Requiring a low stiffness is resulting in a spring with a relative large stroke. This large spring is unambiguously adding mass to the system. The added mass will be even larger due to the interface that has to be made to apply the preload on the small holder of the capillary.

At high frequencies there is also the problem that the assumed fixed world, where the actuator is pushing against, is in fact a bounded geometry. Thereby internal modes of this geometry will be excited as well and these vibrations will be seen back in the actuation.

Finally it is difficult to constrain this asymmetric concept in the not actuated directions. Especially in vertical direction the system has to be constrained, since the touchdown of the ball on the die will generate disturbance forces in that direction. These orthogonal vibrations can cause delaminating of the multilayer piezo actuators as well if the moving part is only held between the piezo and a weak spring.

**Second piezo**

The option of a second piezo actuator provides a better solution. By having both actuators moving with an equal and opposite amplitude, the intended vibration is realized. This is also called bipolar antagonistic drive [6] as drawn in Figure 15.

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**Figure 15: Bipolar antagonistic system**

---

24 Architecture
This way of driving does not suffer from additional mass and stiffness due to a preload spring. Hence, the springs are not participating in the vibration of the capillary. In fact, the stiffness of the system is not reduced, but doubled by adding the second piezo, resulting in a factor $\sqrt{2}$ higher natural frequency. The drawback is that electrical capacitance is doubled, reducing the electrical bandwidth with the same factor. However this can be compensated if necessary with an additional amplifier.

The problem of constraining the orthogonal movements is solved, since the moving mass is not free anymore at one end, but is enclosed by rigid bodies. These bodies, further referred to as counter masses, are much easier to keep in place.

Also the vibrations coming from the framework are isolated from the system by separating the vibrating system from the framework. In Figure 15 the fixed world is therefore removed. This isolation of can be done since the system has an internal force balance, due to the symmetrical actuation. Whatever the internal vibration is, the global position is maintained.

Secondly this separation creates the space that is necessary to easily preload the system. For this preload spring the same story holds as explained before. A weak spring with a high force is desired. Only in this case the spring does not contribute to the moving mass of the capillary anymore, which gives more design freedom.

In Table 5 the properties of the two drive concepts are summarized. From here it turns out that the bipolar antagonistic drive concept is advantageous.

<table>
<thead>
<tr>
<th>Preload spring concept</th>
<th>Bipolar antagonistic drive concept</th>
</tr>
</thead>
<tbody>
<tr>
<td>Decrease of stroke by preload spring</td>
<td>Preload spring not affecting vibrating system</td>
</tr>
<tr>
<td>Hard to constrain orthogonal movements</td>
<td>Easy to constrain</td>
</tr>
<tr>
<td>Interface required for preload spring</td>
<td>Easy to attach preload spring</td>
</tr>
<tr>
<td>Vibrations originating from framework</td>
<td>No vibration originating from framework</td>
</tr>
<tr>
<td></td>
<td>Increased mechanical resonance frequency</td>
</tr>
<tr>
<td></td>
<td>Decreased electrical resonance frequency</td>
</tr>
</tbody>
</table>

### 3.5 Conclusion

The architecture of the TTS is based on a bipolar antagonistic principle, driven off-resonant by two piezo actuators in counter phase. The multilayered linear piezos actuators turned out to give the best performance, by having a bandwidth of $>500$ kHz and a stroke of $2.2 \mu$m. By choosing for symmetry and isolation of the vibrating system a mechanical system is realized that should be capable to operate up to 200 kHz. Monitoring of this motion will be done by a laser vibrometer, to measure at different locations, e.g. the capillary tip. Drawback of the concept is the high dissipation. For
commercial applications this can be a bottleneck, but for the TTS, which can operate at lower duty cycles, this is not a problem.
4. System analysis

In this chapter a model of the designed system will be introduced. With this model the system can be further dimensioned. In the first place only the mechanical behaviour is considered. Thereby the assumption is made that the electrical part is not influencing the mechanics of the system. From C.3 it is known that mechanically this is indeed true for voltage controlled piezo actuators.

4.1 First calculation

As made clear before the goal is to reach 2 µm displacement at the capillary tip up till 200 kHz in a push-pull configuration. The masses and stiffnesses of the different elements from the concept have to be optimised further to realise this goal. To make some first calculations the parameters in Table 6 are assumed.

<table>
<thead>
<tr>
<th>Element</th>
<th>Dimensions [mm]</th>
<th>material</th>
<th>E-modulus [GPa]</th>
<th>Approximated stiffness in actuation direction [kN/mm]</th>
<th>Density [kg/m³]</th>
<th>Mass [mg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capillary</td>
<td>15.88x1.58</td>
<td>Aluminum oxide (Al₂O₃)</td>
<td>300</td>
<td>700</td>
<td>3720</td>
<td>100</td>
</tr>
<tr>
<td>Capillary holder</td>
<td>3x3x3</td>
<td>Aluminum</td>
<td>71</td>
<td>210</td>
<td>2770</td>
<td>75</td>
</tr>
<tr>
<td>Counter mass</td>
<td>6x5x5</td>
<td>Steel</td>
<td>200</td>
<td>2000</td>
<td>7800</td>
<td>1200</td>
</tr>
<tr>
<td>Piezo actuator</td>
<td>2x2x2</td>
<td>Lead zirconate titanate (PZT)</td>
<td>48.3</td>
<td>96</td>
<td>7800</td>
<td>62</td>
</tr>
</tbody>
</table>

Table 6: Dimensions

First of all one has to be sure that the vibration of the piezo actuator will mainly excite the capillary and not the counter mass. Since 2 µm is the requirement almost 10% of the stroke of 2.2 µm, as specified in Table 3: comparison shear and linear piezo actuators Table 3, §3.1.3, can be lost in exciting the counter masses. This percentage is also determining the ratio of masses of the counter mass (m_{cm}) and the capillary with holder (m_{ch}).

\[
\frac{2m_{cm}}{m_{ch} + 2m_{cm}} > 0.9 \quad (4.1)
\]

Taking the mass of the capillary with holder as specified, the counter mass should be >1g. Based on this mass the counter mass is dimensioned as 6x5x5 mm.

The easiest way to determine this eigenfrequency corresponding to the push pull movement is to assume a simple mass spring system, where the counter masses are assumed to be fixed. Then the moving mass is a concentrated mass consisting of the
capillary, its holder and the equivalent moving mass of the two piezos, which is 1/3 of
the mass.

The stiffness is the stiffness of two piezos in parallel, neglecting the stiffness of the
holder with capillary.

\[ f = \frac{1}{2\pi} \sqrt{\frac{2 \cdot k_{\text{piezo}}}{m_{\text{holder}} + m_{\text{capillary}} + \frac{2}{3} m_{\text{piezo}}}} \]  \hspace{1cm} (4.2)

This equation will result in an eigenfrequency of 150 kHz. Then the system will decouple.
The amplitude will decrease according to a -2 slope indicating a decrease of 40 dB per
frequency decade. Once decoupled it requires more and more energy to still actuate the
system. Therefore the challenge is to get this frequency as close as possible towards the
desired 200 kHz.

\subsection*{4.2 Basic model}

For the case the counter masses are not assumed as fixed a new model is set up in
Matlab. Still this new model is kept quite simple and understandable according to the
following schematic. Based on this schematic a set of equations can be written down.

![Figure 16: system model for TTS](image)

\begin{table}[h]
\centering
\begin{tabular}{|c|c|}
\hline
\textbf{m} & \textbf{Explanation} \\
\hline
\textbf{m}_1 & Counter mass + half the piezo \\
\textbf{m}_2 & Capillary + capillary holder + piezo \\
\textbf{m}_3 & Counter mass + half the piezo \\
\textbf{k}_1 & Piezo stiffness \\
\textbf{k}_2 & Piezo stiffness \\
\hline
\end{tabular}
\caption{Explanation table}
\end{table}

For each of the masses holds that there is an internal force balance, which is written
down in a set of equations in (4.3). Then in equation (4.4) the derived equations are
expressed in a matrix representation with mass matrix (M) and stiffness matrix (K).
\[ \begin{align*}
m_1 \ddot{x}_1 - k_1 (x_2 - x_1) &= F \\
m_2 \ddot{x}_2 + k_1 (x_2 - x_1) - k_2 (x_3 - x_2) &= -2F \\
m_3 \ddot{x}_3 + k_2 (x_3 - x_2) &= F \end{align*} \] (4.3)

\[ M \ddot{x} + K x = F \]

\[ \begin{bmatrix} m_1 & 0 & 0 \\
0 & m_2 & 0 \\
0 & 0 & m_3 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\
\ddot{x}_2 \\
\ddot{x}_3 \end{bmatrix} + \begin{bmatrix} k_1 & -k_1 & 0 \\
-k_1 & k_1 + k_2 & -k_2 \\
0 & -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\
 x_2 \\
x_3 \end{bmatrix} = \begin{bmatrix} F \\
-2F \\
F \end{bmatrix} \] (4.4)

### 4.3 Eigenmodes

The eigenfrequencies \( \omega \) of this undamped system can be found by the equation:

\[ \det(K - \omega^2 M) = 0 \] (4.5)

Then the eigenvector \( x_r \) belonging to the eigenfrequency \( \omega_r \) is then be obtained by solving:

\[ (K - \omega_r^2 M) x_r = 0 \] (4.6)

This system has three concentrated masses corresponding to three nodes. Therefore one will find three eigenfrequencies. In case the system is indeed symmetric as pictured, the following holds:

\[ \begin{align*}
m_{cm} &= m_1 = m_2 \\
m_{ch} &= m_2 \\
k &= k_1 = k_2 \end{align*} \]

The eigenfrequencies corresponding to this system are then described as:

\[ \omega_1 = 0 \]

This is the rigid body motion since the system is not fixed to the world.
\[ \omega_2 = \sqrt{\frac{k}{m_{cm}}} \]  
In this motion the end masses move in opposite direction. Because of the equal masses, the displacement is equal and mass m stand still. Therefore the middle mass can be seen as fixed world and the frequency corresponds to the frequency of a single mass spring system.

\[ \omega_3 = \sqrt{k \cdot \frac{(2m_{cm} + m_{ch})}{m_{ch} \cdot m_{cm}}} \]  
This motion corresponds to the motion where the middle mass moves in opposite direction to the counter masses. This is thus the push pull movement. In case of high counter masses, the \( m_{cm} \) vanishes and the equation simplifies to equation (4.2) of §4.1.

### 4.4 Frequency response

To see the frequency response function (FRF) of the system it is useful to make a state space notation of the mechanical model according to E.1.3.

In case the system is excited exactly according to the third modal shape, it is clear that only one resonance peak will appear. However when there is a slight asymmetry in the system the second mode shape will be visible as well. In Figure 17 these resonances can be seen at 42 and 152 kHz. This third mode is indeed close to the calculated frequency of 150 kHz of 4.1.

Furthermore it is seen that the response will not drop below the dc level up to 215 kHz, which is enough to meet the criterion of 200 kHz.
Figure 17: Axial displacement in [m] over force in [N] for a slight asymmetric excitation showing a usable frequency range up to 215 kHz

### 4.5 Opportunities for bandwidth optimization

From the previous paragraphs it appeared that the specified frequency range was obtained, with the notion that third eigenmode was located in the set domain. In order to shift this peak outside this domain the stiffness of the piezos should be increased or one or multiple masses have to be lowered. Practically this is however difficult as can be seen with help of (4.7):

\[
\omega_3 = \sqrt{\frac{k \cdot (2 \cdot m_{cm} + m_{ch})}{m_{ch} \cdot m_{cm}}}
\]  

(4.7)

- Decreasing the counter mass (M) will not help too much, since it always has to be larger than the small mass in order to maintain the high amplitude of the capillary.

- Decreasing the load (m) will help, but it is hard to realize this. To lift the eigenfrequency from 152 kHz to 200 kHz the mass has to be reduced by a factor \( \left( \frac{200}{152} \right)^2 = 1.64 \).

- The capillary is chosen based on its dimensions and cannot be changed. The holder should enclose the capillary and can therefore not bring the necessary mass decrease.

- The remaining option is to increase the stiffness of the piezo. In the specifications of the piezo actuators in 3.1.3 it was possible to get the multilayer actuator of 2 mm with different surfaces; 2x2, 3x3 and 5x5. Choosing the 3x3 piezo results in a resonance frequency of 202 kHz. However since the
capacitance will scale also linear with the area, the required current is more than
two times higher. This is not desirable concerning the thermal consequences.

4.6 Conclusion
Based on the first calculations it turns out that the third resonance peak did not reached
the specified 200 kHz. However the frequency response showed a usable range up to
215 kHz. However it is important to realize three important aspects towards this model
that influence the dynamical response:

• For a better description of the actual frequency response of the TTS a finite
element model has to be made with multiple elements.

• In the current frequency responses the amplitude was related to the middle
point of the drive system. In practice the response of the capillary tip is most
important. It is expected that the capillary will not move in a similar way as the
holder, due to its internal stiffness. Therefore a finite element model is required.

• The calculations done so far are based on the material properties. Since the
piezo is a multilayered, it is likely that its stiffness will be slightly less.

Based on the first calculations done, the concept showed feasible to proceed to a real
design. To be on the safe side it is desirable to make this design such that the 2x2x2
piezo is interchangeable with the 3x3x2 piezo in order to lift the bandwidth in a later
stage.
5. Mechanical design

In Chapter 3 and 4 the first architectural choices and calculations were made. In this chapter the TTS will be further worked out to a mechanical design. This includes the design of the different components present in the system. The manufacturing and assembly of these components is worked out in Appendix D. The drawings for the different components can be found in Appendix H.

In Figure 18 the setup design is presented. The numbered items correspond to the following parts:

1. Piezo actuator (3.1.3)
2. Capillary holder (5.2)
3. Preload spring (5.5)
4. Framework (5.3)
5. Counter mass (5.1)
6. Capillary (3.4.1)

Figure 18: TTS assembly

5.1 Counter masses

In chapter 4 the counter mass was assumed to be a steel block of 6x5x5mm in order to get a nominal 2 μm stroke. However to still have some margin of error the steel masses were increased by 20% to become 6x6x5 mm, corresponding to 1.4g. From Figure 19 it can be seen that with these counter masses a fraction of 0.92 is obtained using the basic model of paragraph 4.2.
With the masses set to 5x6x6 mm, no internal resonances will occur within the specified frequency range. The first eigenfrequency will occur at 465 kHz, according to equation (5.1). By using these ‘small’ masses, it is prevented that vibrations attenuate through the complete structure.

$$\frac{1}{2\pi} \sqrt{\frac{12E}{\rho L^2}} = 465kHz$$  

(5.1)

To keep these masses into place weak leaf springs are used. These leaf springs are compliant in actuation direction to minimize the influence on the motion, but have a high stiffness in the orthogonal directions.

### 5.2 Capillary holder

To drive the capillary on the piezo actuators a good contact is necessary. Therefore the capillary is clamped inside a holder such that its position is fixed with respect to the actuators. Important consideration in this holder design is that it will have a low mass, since this will contribute to the driven load. Therefore this part is made of aluminum, which is almost three times lighter than steel. This difference of material requires assembly of the part in the framework. This has the advantage that this relative vulnerable moving part can be replaced in case it is damaged. Replacing the complete structure would be much more expensive.
The holder is having a three line contact with the capillary, to properly align the tool in the center of the holder. In Appendix E.3 the stiffness of these single contacts was found to be 371 kN/mm using Hertz model. The outer surfaces of the holder match the actuation surface of the largest selected piezo actuator. This way it is ensured that the pressure on the piezo is equal over its surface. Thereby it follows that a cubic holder of 3x3x3 mm is chosen to hold the capillary shown in Figure 20.

In the present transducer the capillary is clamped by means of a tightening screw. This will however introduces additional mass and asymmetry of the holder. Another option is to press fit the capillary in the holder. The disadvantage of this option is that the press fitted parts cannot be disassembled again. The chosen solution is to use the preload force not only for preloading the piezo, but as well to clamp the tool in between the two parts of the holder. The implication is that all parts should be assembled together in one time. In Appendix D.1. the assembly steps are explained.

As long there is no preload the two parts of the holder should just like the counter masses been held in position. According to the same philosophy leaf spring were applied. However in comparison to the counter masses, the stiffness of these springs is reducing the effective force of the actuator. Therefore a relative low stiffness is required.

### 5.3 Framework

By separating the vibrating system from its framework, makes it unnecessary to thoroughly analyze the dynamics of this part. Of course it should still have to cope with the accelerations during positioning. However during bonding the framework will only move downwards. Since this direction is orthogonal to the bonding movement, this may not cause problems.
More important is that the framework will fit in the Phicom to perform the bond tests. However due to limited space, it was required to remove the camera, to still be able of moving the TTS in the bond area. The spark unit on the other side could not be removed. Thereby it was chosen to have a C-frame instead of an O-frame.

Furthermore the framework should have the possibility to be mounted to the interface of the Phicom. Currently the ultrasonic transducer is mounted to the Phicom by means of a flange with three screws. The same interface will be used to mount the TTS.

From this interface the other surfaces of the framework were defined. Orthogonality and flatness of these surfaces are important for a properly aligned assembly in order to actuate the system indeed in the desired direction.

### 5.4 Preload

In the discussion on actuators of 3.1 it was mentioned that the linear piezo actuator requires preload during dynamical operation to avoid tensile stresses. Recommended is to have a preload corresponding to at least half the blocking force e.g. >60 N. With this preload the piezo is compressed with half of its maximal stroke. As a result no tensile stresses will occur when the piezo oscillates at full stroke around this working point.

Two methods to preload a piezo is active or passive preloading. Active preloading is done by generating an offset by means of a dc voltage. Passive preloading uses a (constant) mechanical force on the piezo. In a rigid frame only active preloading would work. Having moving masses on leaf springs gives the opportunity to use passive preloading by means of springs. The relative large counter masses will generate enough space to apply a force locally and push the system together. In contrary to a rigid frame, temperature is not affecting this amount of preload, which is an advantage especially for a thermosonic process.

As explained in 3.4.3 a weak spring with a high force is desired. As revealed before the easiest way to obtain this force is by a relative weak compression spring, which can be placed in line of actuation as can be seen in Figure 22. Other options were disc springs.
or the already implemented leaf springs. However disc springs are in general stiffer and used for higher forces. The leaf springs will require a too large displacement resulting in high internal stresses and a shift of the nominal position of the system. Finally a preload spring with a stiffness of 40.4 N/mm was selected (Appendix G.2). The consequence for the response of the TTS is discussed in the next paragraph.

![Figure 22: Preload spring and dowel assembly](image)

Applying the preload is done by fastening a M3 in the threaded hole of the dowel. Further screwing is pushing the spring against one of the counter masses. Important to realize is to apply the preload simultaneously from both sides to avoid high stresses on the leaf springs.

### 5.5 Design criteria decoupling springs

In order to have a pseudo-floating system all springs should be already decoupled at the desired frequency range of 50-200 kHz. The axial stiffness ($k_x$) of the leaf springs corresponds to the s-curved bending stiffness, which is a function of the elastic modulus of the material ($E$), the area moment of inertia ($I$) and the length of the spring ($l$) according to equation (5.2). The area moment of inertia is determined by the width ($w$) and thickness ($t$) of the leaf spring according to equation (5.3).

\[
\begin{align*}
    k_x &= \frac{12EI}{l^3} \quad \text{(5.2)} \\
    I &= \frac{wt^3}{12} \quad \text{(5.3)}
\end{align*}
\]

In case of touchdown the vertical stiffness is also relevant. This stiffness is determined in the shear direction of the leaf springs according to equation (5.5)[16].

\[
    k_y = E \cdot t \cdot \frac{w^3}{4 \cdot l^3 + 3 \cdot w^2 \cdot l} \quad \text{(5.4)}
\]
An overview is given of the different springs in Table 8:

**Table 8: Overview of spring stiffnesses**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>( K_{pl} )</th>
<th>( K_{cm} )</th>
<th>( K_{ch} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description</td>
<td>Preload compression spring</td>
<td>Leaf spring supporting counter mass</td>
<td>Leaf spring supporting capillary holder</td>
</tr>
<tr>
<td>Quantity (q)</td>
<td>2</td>
<td>2x2</td>
<td>2x2</td>
</tr>
<tr>
<td>Length (L in mm)</td>
<td>7.6</td>
<td>5.45</td>
<td>2.5</td>
</tr>
<tr>
<td>Width (w in mm)</td>
<td>4 (diameter)</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>Thickness (t in mm)</td>
<td>0.1 (wire)</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>E-modulus (E in GPa)</td>
<td>-</td>
<td>200</td>
<td>71</td>
</tr>
<tr>
<td>Axial stiffness (( k_x ) in N/m)</td>
<td>40.4</td>
<td>6.2</td>
<td>13.6</td>
</tr>
<tr>
<td>Vertical stiffness (( k_y ) in kN/mm)</td>
<td>-</td>
<td>2.4</td>
<td>1.5</td>
</tr>
</tbody>
</table>

The masses are except for the counter mass similar as in Chapter 4.

**Table 9: Overview of masses**

<table>
<thead>
<tr>
<th>Description</th>
<th>( m_p )</th>
<th>( m_{cap} )</th>
<th>( m_{cm} )</th>
<th>( m_{ch} )</th>
<th>( m_t )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description</td>
<td>Piezo actuator</td>
<td>Capillary</td>
<td>Counter mass</td>
<td>Capillary holder</td>
<td>Total mass</td>
</tr>
<tr>
<td>Mass [mg]</td>
<td>64</td>
<td>100</td>
<td>1400</td>
<td>73</td>
<td>3101</td>
</tr>
</tbody>
</table>

The vibrating system will decouple from all these leaf springs at 1.14 kHz, which is far enough below the 50 kHz to avoid troublesome resonances.

\[
\frac{1}{2\pi} \sqrt{\frac{2k_{pl} + 4k_{cm} + 4k_{ch}}{m_t}} = 1.14 \text{ kHz}
\]  

(5.5)

In case of accelerating the complete transducer on an XY stage this stiffness is preventing the masses to drift away. The amplitude will be 3.9 \( \mu \)m in case of a desirable forward acceleration of 200 m/s\(^2\). However during bonding the transducer will stand still and the vibration is already diminished during the downward movement.

More important is to check whether the leaf springs can have enough deflection before plastic deformation or, even worse, fracture takes place. In Figure 23 the free body diagram is drawn for a single leaf spring. For the particular loading case moment (M) is equal to (5.6). This is as well the critical moment that occurs on both ends of the spring.

\[
M = \frac{1}{2} FL
\]  

(5.6)
The stress ($\sigma$) at the edges can be described by the moment, the eccentricity to the centerline ($y$) and the area moment of inertia ($I$).

$$\sigma = \frac{My}{I} \quad (5.7)$$

In table the resulting critical forces and related tip displacements are listed for both types of leaf springs.

Table 10: Leaf spring limits

<table>
<thead>
<tr>
<th></th>
<th>$k_{cm}$</th>
<th>$k_{ch}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield strength [Mpa][17]</td>
<td>$R_{p0.2}$</td>
<td>260</td>
</tr>
<tr>
<td>Maximum force [N]</td>
<td>$F_{\text{max}}$</td>
<td>0.8</td>
</tr>
<tr>
<td>Maximum displacement [mm]</td>
<td>$\delta_{\text{max}}$</td>
<td>0.13</td>
</tr>
</tbody>
</table>

During operation or acceleration these figures provide enough freedom. However during assembly care should be taken that the preload is applied symmetrically. The difference in force can be maximal 10N:

$$\Delta F = \delta_{\text{max}} \cdot (4k_{cm} + 4k_{ch}) = 0.13 \cdot 4(6.2 + 13.6) = 10N \quad (5.8)$$

### 5.6 Conclusion

The design of the TTS is finalized as seen in Figure 24. It shows the monolithically manufactured C-frame with leaf spring guided counter masses. In this frame the remaining components i.e. capillary, capillary holder and piezo actuators were assembled. For positioning of these components an assembly tool was used described in Appendix D. By using preload springs the vibrating system was enclosed. The complete vibrating system was designed to decouple at 1.1 kHz from all supporting springs to become a dynamically isolated system. The realized TTS is able to be build in the Phicom as shown in Appendix I.
Figure 24: Realized TTS
6. Finite element model

In the conclusion of chapter 4 it was suggested to make a finite element model (FEM) of the setup to get a better notion of its behaviour. It is not useful to make this model of the complete TTS, only the truncated part of Figure 25 needs to be modelled in order to investigate the dynamical behaviour of the capillary. It was decided to make a model in Matlab, to quickly change the parameters of the model and to get fast results. Furthermore an own model leads to a better understanding of the system.

![Figure 25: Modeled part of design](image)

6.1 Setting up the model

6.1.1 Mechanical

The mechanical model should have at least one degree of freedom (DOF) in actuated direction. Since the design is made symmetric this one DOF model is indeed a realistic approach. For the elements in line with the actuation (drive train) this requires 1 DOF rod elements. The capillary should be modelled with 2 DOF beam elements, because of its bending behaviour. The drive train and capillary are both divided in 50 elements, consisting an elementary stiffness and mass matrix. Increasing this number even further will finally lead to a better geometrical description. However the rounding errors occurring in Matlab will get higher.

The connection of the capillary to the holder is based on the Hertz contact calculation, done in Appendix E.3, resulting in a stiffness of 371 kN/mm. With this stiffness the capillary is connected along the length of the holder on both sides as shown schematically in Figure 26. Note that the internal stiffness of the different parts is not visualised. On the basis of this description of Figure 26 the mechanical finite element model is realised. The global mass matrix (M) and global stiffness matrix (K) are build up by assembling the corresponding elementary matrices according to the finite element principles described in Appendix E.1.
6.1.2 Electromechanical

The mechanical finite element description has to be incorporated in an electromechanical model to include the piezoelectric properties of the system. Fundaments for making this finite element piezoelectric model are the constitutional laws for piezoelectricity [18] written down in equation (6.1) and equation (6.2). In equation (6.1) one can retrieve the original mechanical stress strain relation, but now completed with an electromechanical coupling factor $d$. This factor relates strain to an electrical field. Equation (6.2) gives the electrical description of a piezo element. The relation between the electrical displacement and the electrical field is not only described by the dielectric coefficient, but again as well by the coupling factor $d$, which in this case relates stress to an electrical field.

\[
S = s_E \cdot T + d^T \cdot E \quad (6.1)
\]

\[
D = d \cdot T + \varepsilon_T \cdot E \quad (6.2)
\]

- $S$ in [-] 6x1 strain vector
- $T$ in [N/m$^2$] 6x1 stress vector
- $E$ in [V/m] 3x1 electric field vector
- $D$ in [C/m$^2$] 3x1 electric displacement vector
- $s_e$ in [m/V] or [C/N] 3x6 compliance matrix with constant electrical field
- $d$ in [m/V] or [C/N] 3x6 piezo stress coefficient matrix
- $\varepsilon_T$ in [F/m] 3x3 dielectric coefficient matrix with constant stress

In C.3 these equations are rewritten to a voltage controlled format. Subsequently a finite element model is derived in Appendix E.2 and written down in state space format as shown in equation (6.3). Also a charge controlled expression is worked out in both appendices in case this type of control will be used in a later stadium.

\[
\begin{bmatrix}
\dot{X} \\
\dot{\dot{X}}
\end{bmatrix} =
\begin{bmatrix}
0 & I \\
-M^{-1}K_{xx} & 0
\end{bmatrix}
\begin{bmatrix}
X \\
\dot{X}
\end{bmatrix} +
\begin{bmatrix}
0 & 0 \\
M^{-1} \cdot d_{33} \cdot K_{xx} \cdot T
\end{bmatrix}
\begin{bmatrix}
F \\
V
\end{bmatrix}
\]

\[
\begin{bmatrix}
\dot{X} \\
\dot{Q}
\end{bmatrix} =
\begin{bmatrix}
I & 0 \\
d_{33} \cdot K_{xx} \cdot T^T & 0
\end{bmatrix}
\begin{bmatrix}
X \\
\dot{X}
\end{bmatrix} +
\begin{bmatrix}
0 & 0 \\
0 & K_{pp}
\end{bmatrix}
\begin{bmatrix}
F \\
V
\end{bmatrix}
\] (6.3)
6.1.3 Damping

In the previous paragraphs no notion is made towards a damping term. Due to damping the poles of the system can shift slightly and the amplitude of the resonance frequency will reduce. For the designed system the damping is assumed to be very small. Mechanically there are no elements like sliding contacts and electrically the (leakage) resistance is very high (>MΩ). These damping factors will thus not contribute to complete different dynamics. Since damping is usually hard to predict a simple way to construct a damping matrix is by the method of Rayleigh. Thereby the matrix is a factorized combination of the stiffness and mass matrix, according to:

\[ C_{xx} = \alpha M + \beta K_{xx} \quad (6.4) \]

These factors \( \alpha \) and \( \beta \) can be modified in order to stroke with the reality. For now this damping matrix is taken with very small values, only to obtain a numerically stable model.

6.2 Frequency response

The resulting frequency response function (FRF) of the designed system is presented in Figure 27. To draw some similarities with the response function of Chapter 4 (Figure 17), the response is taken at the centre of the system, i.e. the middle of the capillary holder. Where the first calculation resulted in an eigenfrequency of 152 kHz, now this resonance peak is shifted to 167 kHz. Furthermore the frequency where the -154 dB line (2µm/100V) is crossed is shifted to 216 kHz. The reason for this shift lays in the fact that
capillary decouples already at 46 kHz (bottom part) and 52 kHz (upper part). The reduced moving mass is resulting in an increased eigenfrequency.

![Graph](image1.png)

**Figure 27:** Axial displacement in [m] over voltage in [V] of the center of the capillary holder

To answer the question whether this decoupling is harmful one should look at the response of the capillary tip. This is after all the spot where the bond is made. From Figure 28 it can be seen that this capillary end is indeed decoupled at 46 kHz. However the amplitude remains above the -154 dB for a much wider ranger. The anti-resonance at 249 kHz of the holder corresponds to a resonance at the tip. However the resonance of the upper part of the capillary at 54 kHz is causing then again an anti-resonance at the tip. Since this is on the edge of the specified range it is not a problem.

![Graph](image2.png)

**Figure 28:** Axial displacement in [m] over voltage in [V] of the capillary tip

The first appearing mode of the system is not visible in previous figures due to the limited frequency range. In this first mode the vibrating system resonates on its springs.
From Figure 29 this frequency is found at almost 1 kHz, close to the approximated 1.1 kHz of 5.5.

![Frequency vs Voltage Plot]

**Figure 29: Axial displacement in [m] over voltage in [V] of the capillary tip indicating the effect of spring connections**

### 6.3 Mode shapes

To get a better understanding of the resonance frequencies seen in 6.2 the modes shapes of the mentioned frequencies are visualised in Figure 30. In here the actual movement is drawn in red. The movement of the drive train is thus indicated by the dots. To visualise this longitudinal waveform of the drive train even better the thin blue line is showing the orthogonal projection of this movement. Note that all mode shapes are individually scaled.

<table>
<thead>
<tr>
<th>Waveform</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>33 kHz</td>
<td>Counter masses are resonating, where the centre stays still</td>
</tr>
<tr>
<td>46 kHz</td>
<td>Bottom part of capillary is resonating</td>
</tr>
<tr>
<td>52 kHz</td>
<td>Upper part of capillary is resonating</td>
</tr>
</tbody>
</table>
Waveform Description

<table>
<thead>
<tr>
<th>Waveform</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push pull mode, where the holder with the capillary is resonating on the counter masses</td>
<td></td>
</tr>
<tr>
<td>Second resonance mode of the bottom part of the capillary</td>
<td></td>
</tr>
</tbody>
</table>

Figure 30: Important mode shapes

6.4 Model sensitivity

The model as presented in the previous paragraphs is made based on a number of assumptions and calculations. For example only axial movement is taken into account, because of the symmetry in the system. In this paragraph the sensitivity of the model is looked at by investigating the influence of the Hertz contact, hysteresis, start-up behavior, bond load and finally orthogonal vibrations.

6.4.1 Sensitivity of Hertz contact

The response of the capillary tip is the most important. However it has a larger uncertainty then the response of the drive train, due to the clamping in the holder. In Appendix E.3 it was already noticed that the contact pressure was just above the critical contact pressure, leading to local plastic deformation. In this way a good contact is established. Because of the plastic deformation Hertz theory is not valid anymore. The contact surface increases leading to a larger stiffness, whereas the elasticity of the aluminium holder decreases leading to less stiffness. In case the contact stiffness approaches infinity it can be seen that the capillary modes shift to higher eigenfrequencies according to Figure 31. However with a minimal exceedance of the contact pressure it is likely the FRF does not deviate significantly from the calculated 371 MN/m.
6.4.2 Influence of hysteresis

The hysteresis is causing the piezo actuators to not operate synchronous and opposite. This hysteresis effect is shown in Appendix F.3 in a time cycle. From the complementing spectrum plot it turns out that the energy content is almost completely located in a single frequency. Thus there is barely a excitation of other frequencies and, hence, the hysteresis can be seen as a pure phase loss.

However, this phase loss is non linear with the voltage. When linearised, it can be seen as a mechanical damper or electrical resistance. It dissipates energy from the system, making the piezo no longer an ideal capacitor. The linearised damper can be enclosed in the damping matrix $C_{xx}$. The chosen actuators were specified to have a loss of 20% coming from this hysteresis effect. The actual current will thus be 20% higher than the calculated current, when driving it in its full stroke.

Due to the hysteresis the actuators enclosed in the TTS, will excite other modes of the system as well. This will diminish the amplitude of the desired motion. To see the effect of this Figure 32 shows the difference in response when one piezo is at maximum voltage, while the other is at 80% of its maximum voltage. From the figure it is seen that the counter mass mode at 33 kHz is now excited as well. Furthermore the amplitude drops with 1 dB, which is 12% loss. To avoid this effect of hysteresis charge control can be used, see Appendix C.3.
6.4.1 Start-up behaviour

Switching on the control signal of a single sine in one step causing the excitation of multiple frequencies. From Figure 33 and Figure 34 it is seen that a step function of 200 kHz is exciting the eigenfrequency at 167 kHz as well. The amplitude spectrum clearly shows two peaks at these frequencies. The time response is showing these two frequencies as well, clearly visible by the beat frequency of 33 kHz, which is the difference between these. Furthermore it is seen the amplitude peaking up to 4 µm, indicating the eigenfrequency excitation.

To purely excite a single frequency the signal has to be build up for instance by an exponential function. In the figures an exponential function is used with a time constant τ of 0.13 ms. This time constant is a compromise between a fast build-up time and a pure sinusoidal signal. With this exponential function the time response shows a much better response converging to a 2 µm amplitude. Furthermore the power spectrum of Figure 34 is shows only one clear peak at 200 kHz. Due to the build up, the energy level of this spectrum is less than the one of the step function response.
6.4.2 Sensitivity investigation capillary offset

Although the system is configured with the intention that it will move in one direction, still it is possible that especially at higher frequencies torsional modes will start to play a role as well. To get some insight in the response of the system without constraining these movements the model has to be adjusted. The way to do that is replacing the 1d rod elements by 2d beam elements. In contrast to the beam elements of the capillary these elements still require an axial movement, resulting in a elementary matrix of 3 by 3; translation in x, translation in y and a rotation in the xy plane. Again the connection to the capillary is modeled by breaking up the global matrix and introducing a Hertz stiffness contact.
New in the model are the boundary conditions for the vertical (y) direction, coming from the vertical stiffness of the leaf springs. For the counter masses and the capillary holder these stiffnesses are respectively 2.4 kN/mm and 1.5 kN/mm according to 5.5.

**Axial response**

This model is used to see the response in axial direction (X) and in vertical direction (Y). In case of symmetric positioning of the capillary, the axial response using this 2D model is similar to the 1D model. This is not surprising, since the bending of the capillary was already included in the 1D model. However in practice there might be a slight offset in position. In Figure 35 and Figure 36 it is seen the response changes when the capillary is moved downwards. The resonance peaks of the capillary move away from each other since the lower part becomes longer and the upper part shorter. Furthermore there is a slight shift in frequency of the push pull mode. This shift can be related to less decoupled mass of the upper part of the capillary.

![Figure 35: Amplitude of holder in [m] over voltage in [V] for a symmetrically positioned capillary and a capillary with 1 mm downward offset](image)

![Figure 36: Amplitude of holder in [m] over voltage in [V] for a symmetrically positioned capillary and a capillary with 2 mm downward offset](image)

**Vertical response**

A second option of this model is to see the response in Y-direction. From Figure 37 it can be seen that the orthogonal Y movements are as expected not significant as long as there is no contact. The Y-movement is ±100dB less than the axial movement even with an offset of the capillary of 1 mm. In Appendix E.4 the corresponding vertical modes of the TTS can be found.
Figure 37: Amplitude in [m] over voltage in [V] for actuation direction (X) and vertical direction (Y) at the center of the holder

6.5 Conclusion

With the use of an electromechanical finite element model, the response of the TTS is predicted. It showed good similarities with the first model of Chapter 4. Important observation was the shifted push pull mode to 167 kHz, due to the introduction of the capillary modes at 46 and 52 kHz. The usable frequency range was limited to 216 kHz by the model. Important note is that the model is based on a linear approach. Effects such as hysteresis are therefore not incorporated in the model.

However based on linear assumptions a sensitivity analysis is done towards this hysteresis, but also start up effects, 2d behavior and the influence of the contact are investigated. Main conclusion is that asymmetry in actuation or geometry, results in excitation of the counter mass mode at 22 kHz and will decrease the effective stroke of the actuator. This happens as well when a ramped signal is used to drive the actuators.
7. Measurements

The simulation performed in Chapter 6 has to be verified on the TTS. These measurements will be done in this chapter. First the measurement set-up is discussed in §7.1. Then in Appendix F.3 and F.4 the performance of the actuators and the amplifier are tested. This equipment is then used in the set-up. The frequency responses and time responses of the set up will be measured in §7.2 and §7.3 respectively. These results will be compared with the simulations done before, such that any discrepancies can be declared.

7.1 Measurement set-up

Figure 38 shows a schematic representation of the used measurement setup. An AC excitation voltage is amplified, where the amplified signal powers the TTS. With a laser vibrometer and a network analyzer the TTS is evaluated. In Appendix F all testing equipment is listed including a complete wiring diagram is included. Also the measurements of the amplifier and actuators can be found in here.

![Measurement set-up diagram]

7.2 Frequency response

7.2.1 Measurement locations

In order to do a system identification the amplitude will be measured at different spots. Of course the vibration at the capillary tip will be crucial. However in order to find the main resonances of the TTS a complete list of measurements is done according to Table 11. By the first ten measurements the modes in line of actuation are revealed. Then the capillary movement is described in more detail by measurements 11 to 17. The

Figure 38: Measurement set-up
remaining measurements are out of plane measurements. In order to quantify the rotations belonging to the differential measurements a column is added to obtain the intermediate distance.

<table>
<thead>
<tr>
<th>Absolute</th>
<th>Differential</th>
<th>Intermediate distance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>6: 15 mm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7: 4 mm</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Along capillary</td>
<td>Out of plane</td>
<td>22: 13 mm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>24: 11 mm</td>
</tr>
</tbody>
</table>

### Table 11: Locations of FRF measurements

#### 7.2.2 Results

All transfer functions are obtained from the input voltage of the actuators to the velocity of the measured spot. The input voltage delivered by the amplifier equals 1.1 V RMS. By means of integration of the velocity, the position signal was acquired. This indirect displacement measurement proves to have a better signal to noise ratio and less phase loss due to the higher bandwidth of the velocity decoder, which is 1.5MHz according to Appendix G.6. It was chosen to represent the displacement in absolute scale to get a direct idea whether the 2 µm stroke is achieved in the complete frequency range at a maximum voltage of 100 V<sub>pp</sub>. This would then correspond to 20 nm/V. However Appendix C.4.1 demonstrates the non-linear behavior of piezo actuators. Both hysteresis and stroke increase exponentially with the applied voltage. Thus it yields that small signal excitations like these will result in a relative smaller stroke than its linear approximation.
**Capillary holder**

The first measurement is obtained from just above the capillary holder. The resulting graph is displayed in Figure 39 together with the modeled transfer function of §6.2 of the holder. It can be seen directly that the off resonant level of the measurement does not correspond to the modeled 20 nm/V. It shows a amplitude level of only 3 nm/V. In §7.4.1 this will be evaluated. Furthermore it can be seen that the modeled eigenmodes of the TTS are measured as well, but in some cases these are shifted in frequency. There is the suspension decoupling frequency at 1.1 kHz, the capillary mode at 46 kHz and the push pull mode at 120 kHz.

Then there is a resonance peak at 21 kHz, corresponding to the second mode, where the counter masses move in opposite direction. The presence of this mode indicates a non-perfect symmetric system or asynchronous actuation. This peak is however small compared to the push pull mode, which is clearly the largest with an amplification of 12 and a value of 14 nm/V.

Interesting to see is that the modeled capillary modes are merging to one mode, since these where located very close to each other. This also leads to one anti-resonance at 52 kHz, where the desired stroke of the holder is not obtained anymore.

![Figure 39: Amplitude in [nm] over voltage in [V] of capillary holder (meas. 1)](image)

**Capillary tip**

More important is again the transfer function of capillary tip (meas. 2). The zero in the response of the holder corresponds to a pole of the capillary tip. The resultant resonance peak has a value of 118nm/V, which is an amplification of 24. As a consequence of the disappearance of this zero, Figure 40 shows a response that stays just above 5 nm/V up to 250 kHz. Thereby the off-resonant level clearly differs from the amplitude level of the holder, which was 3nm/V. This deviation will be thoroughly discussed in §7.4.
Counter mass
Final response of Figure 41 is of the counter mass (meas. 9). Based on this response two important observations can be done.

In the first place it is verified that the peak at 21 kHz is indeed the counter mass mode. The anti-resonance of the previous responses at this frequency corresponds to large peaking of the amplitude of the counter mass. Based on the presence of this peak it can be concluded that there is no perfect push pull motion. Since the symmetric model does not contain peaking at this frequency, the measured response is compared to an asynchronously driven model. This model is realized by exciting one piezo actuator at 80% of its usual voltage. The modeled resonance peak is located at 33 kHz, showing again a shift to a lower frequency.

Another interesting observation is that the off-resonant level is at 0.45 nm/V. Based on the 3nm/V amplitude of the holder it is found that 13% of the stroke of the actuator is used for actuating the counter mass. Thereby it can be concluded there is a slight asymmetry in the system, asynchronous actuation or a combination of both. Otherwise the counter masses are only excited at 10% of the actuator amplitude.
7.3 Time response

To verify the mechanical amplitude of the frequency responses of the previous paragraph and to investigate whether the motional amplitude of 1 µm can be obtained, a time domain measurement is performed. To excite only one single frequency an exponentially build-up signal is created according to §6.4.1. This excitation signal is then generated by an arbitrary waveform generator of a USB scope, specified in Appendix G.5.

A typical response of the capillary tip is shown in Figure 42. At 100 kHz, 35 V excitation a stroke of 1µm is obtained. According the frequency response of Figure 40 the phase lag of the displacement is already 235°. This does not match the phase lag of Figure 42. The cause for this dissimilarity is the use of a displacement decoder, which is not capable of measuring the phase adequately at these frequencies. In Appendix F.5 the differences of velocity decoder and displacement decoder are further examined.
For different voltages and frequencies these time measurements are performed. Prior to these measurements the TTS was assembled again. This changed the dynamical response of §7.2 slightly as can be seen in Appendix F.6. To summarize the results of the time measurements the mechanical amplitude as a function of the input voltage are depicted for three distinct frequencies, i.e. 5kHz, 100kHz and 200 kHz in Figure 43. Since the power supply was limited to 70V, a trend line is added to retrieve the 100 V responses as well.

Figure 42: 100 kHz, $70V_{pp}$ excitation response

Figure 43: Stroke in [nm] as function of $V_{pp}$ in [V] for 5kHz, 100 kHz and 200kHz
For 5 kHz the measured off resonant level of amplitude 5 nm/V is indeed measured in the time response as well. However the specified 2 µm stroke is not obtained according to the quadratic fit of Figure 43. Instead only half of the stroke is found.

Exciting the TTS with 100 kHz will yield the specified 2 µm stroke at maximum voltage. It was already seen in Figure 40 that the amplitude at this frequency is a factor 1.6 higher than the off resonant level. The reason for having a more rapid amplitude increase than the 5 kHz curve is due to the shift of the push pull mode to a lower voltage lifting the amplitude at 100 kHz. This shift occurs due to the loss of stiffness at higher voltages as can be seen in Appendix C.4.1.

At 200 kHz excitation the maximum stroke is with only 430 nm more than a factor 4 smaller than the aimed 2 µm. This decrease in stroke is caused by the decoupling mode, which was shifted to a lower frequency in the frequency response during testing according to Appendix F.6. This was causing the presence of a -2 slope at 200 kHz. Increasing the voltage normally leads to an exponentially increased stroke, but since it also shifts the decoupling frequency, the 200 kHz response was placed at an even lower level cancelling the exponential growth.

### 7.4 Comparison of measurement and model

#### 7.4.1 Amplitude comparison

**Off resonant stroke**

Based on amplitude measured by the time response in the previous paragraph it seen that the off-resonant stroke did not reach the expected and modeled 2 µm/100V, but only half. First cause for this loss can be attributed to decreased stroke of the actuators. From Appendix F.3 these strokes turned out to be 1.7 and 1.8 µm. The other verified cause is the actuation of the counter masses. From §7.2.2 this fraction turned out to be 13% of the actuators stroke. However at higher voltages the hysteresis is larger as shown in Appendix F.3, where the power factor of the actuator is decreased to 80% at full stroke at 1kHz. This hysteresis is causing the actuators to not operate synchronous, resulting in excitation of the counter masses.

**Capillary movement**

Another remarkable deviation was found in the frequency response of the capillary tip of §7.2.2. Based on the model this tip should have the same amplitude as the holder, which was 3nm/V. However from Figure 40 it turned out that this amplitude was over 5 nm/V. To explain this, the amplitude was measured along the length of the capillary from top (meas. 11) till tip (meas. 17). The result of Figure 44 shows indeed the variation in amplitude as a function of the position for the off-resonant frequency of 5kHz. The amplitude increases from 2 nm/V at the top to 5 nm/V at the tip.
This variance can only be the result of a combined rotational and translational movement of the capillary as shown in Figure 45. Since this rotation is already present at low frequencies, a plausible reason for this to happen is the eccentricity of the actuators towards the center of rotation ($\varepsilon$). As long as the inertia forces in the capillary stay below the actuation forces, this rotation is observable.

![Figure 45: Capillary movement](image)

### 7.4.2 Modal comparison

Although quite some similarities were found in 7.2.2, there are still some discrepancies between the model and the measurement. In Table 12 the difference in resonance frequencies are written down. It can be seen that the capillary mode is located at the modeled frequency. The other modes deviate from the model. In this paragraph these differences will be explained.
### Table 12: Resonance frequency comparison of model and measurement

<table>
<thead>
<tr>
<th>Mode</th>
<th>Measured frequency (kHz)</th>
<th>Theoretical frequency (kHz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suspension decoupling mode</td>
<td>0.96</td>
<td>1.12</td>
</tr>
<tr>
<td>Counter mass mode</td>
<td>21.4</td>
<td>33.0</td>
</tr>
<tr>
<td>Capillary mode</td>
<td>46.5</td>
<td>45.7/51.9</td>
</tr>
<tr>
<td>Push pull mode</td>
<td>120</td>
<td>168</td>
</tr>
</tbody>
</table>

**Suspension decoupling mode**

The suspension decoupling mode has a decreased frequency of 15% compared to its theoretical equivalent. A plausible explanation for this decrease is the added moving mass of all springs. However this decrease requires a mass increase of 27%, i.e. 0.8g, where the heaviest springs, the preload springs, weight only 0.35g. The moving mass is thus less than the suggested mass increase. This yields that the stiffness of the springs has to be less than expected. Inaccuracy in the process of wire sparking can cause local weak spots in the leaf springs explaining the observed behavior.

**Counter mass mode and push pull mode**

The decrease in frequency of the counter mass mode and push pull mode is even larger with respectively 65% and 71% of the modeled values. This is a serious decrease, which cannot be explained by an uncertainty of mass. More likely it is that the TTS is more compliant than modeled. The stiffness of the piezo actuators was derived from the measured resonance frequency in Appendix C.3.4. Therefore it is not likely its stiffness will differ a lot at low voltages used for the frequency response. More suspicious are the holder with the capillary, since these were not modeled in great detail.

- The capillary was modeled as infinitely stiff in radial direction. Based on the deformation result of Ansys, the stiffness of this part of the holder is 333 kN/mm according to Figure 46.
- The capillary holder was modeled as a solid block of aluminum split in two parts. However the v-groove in the holder makes this part more compliant than modeled. Annoys shows this stiffness to be 142 kN/mm according to Figure 47.
The second part of the holder has no v-groove and is loaded in line of actuation. It has therefore a high stiffness of 887 kN/mm. Adding these three derived “springs” in series with the other unchanged “springs” leads to an equivalent spring of 26 kN/mm as seen in Figure 48.

In case the holder is again modeled as a solid block of aluminum with a stiffness of 210 kN/mm, the equivalent stiffness of this series of springs yields 31 kN/mm.

For the counter mass and the push pull mode this yields a decrease of a factor 1.2 in stiffness and 1.1 in eigenfrequency compared to the modeled frequency. This decrease is a part of the explanation, but still its contribution is too small to explain the complete shift.

Furthermore it was concluded in §7.4.1 that the actuators were misaligned causing a rotation. Thereby the resonance frequencies are affected as well. However it yields that the axial resonance frequency will increase since the axial movement is counteracted by the appearing rotational stiffness.
A final explanation for the shift is that the assumed line contact of the holder modeled by a Hertz stiffness is not a proper line contact. In case of an undefined contact it is hard to estimate the stiffness. An idea is to use a holder in which the capillary can be press fitted or glued to see if these holder show improved results.

### 7.5 Temperature

The temperature step responses of one of the piezo actuator and its adjacent counter mass are presented in Figure 49. The measurement was done at 20 kHz, 70 V_{pp}. The corresponding power level is derived from the measured RMS current and voltage over the actuator. The temperature increase is 31K and 21K for respectively the piezo actuator and the counter mass by a power of 2.6W. From this response the following time constants are derived: 24s for the actuator and 83s for the counter mass.

![Temperature response of piezo actuator and its adjacent counter mass](image)

To derive the thermal resistance of the piezo actuator in the TTS, the power loss in the actuator should be known. From the power triangle of Appendix F.3 a phase angle (φ) was determined as 83° for 70V_{pp}. According to equation (7.1) the dissipation power will be 0.32W resulting in a thermal resistance of 97K/W by equation (7.2).

\[
Q = \cos(\phi) \cdot P = \cos(83^\circ) \cdot 2.6 = 0.32W \quad (7.1)
\]

\[
R_T = \frac{\Delta T}{Q} = \frac{31}{0.32} = \frac{97K}{W} \quad (7.2)
\]

In case of 0.8 W dissipation, corresponding to a 10% duty drive at 200 kHz, according to §3.1.5, the temperature will rise by 78 K, using the calculated resistance. However the internal temperature is a factor 1.5 higher according to [12]. Thereby the maximum duty cycle is limited to 13% to stay below the maximum operational temperature of 150°C. However it should be noted that at elevated temperatures the physical properties of the actuator are changed as well. Therefore it is recommended to use a smaller duty cycle.
and not push the limit.

With the thermal constant of 24s, the temperature does not vary significant during duty, such that previous calculation holds. A bond of 4ms at 200 kHz excitation (8W), will give a temperature rise of only 1.6K according to equation (7.3).

\[
\Delta T = \left(1 - e^{-\frac{t}{\tau}}\right) \cdot Q \cdot R_T = \left(1 - e^{-\frac{4e^{-3}}{24}}\right) \cdot 776 = 1.6K
\]  

(7.3)
8. Conclusions and recommendations

8.1 Conclusions

With success an ultrasonic transducer test setup (TTS) is developed, which is able to generate an ultrasonic welding motion at the tip of the weld tool i.e. capillary of at least 400 nm motional amplitude in the frequency range of 50-200 kHz. Thereby the requirement of 1 µm amplitude as specified in paragraph 1.3 is not met. However by having this amplitude, the TTS is theoretically still capable of making a wire bond, since this level corresponds to the standard motional amplitude of present US transducer.

The TTS is based on a symmetric bipolar antagonistic drive. This way of actuation requires two piezo actuators that are controlled by a synchronous, but opposite signal, creating a push pull motion as explained in chapter 3. By this push pull design it is guaranteed that the TTS is easy to manufacture, to monitor, but most of all it is predictable. This is due to the stand-alone character of the vibrating parts, which is realized by isolating these vibrating parts using leaf springs.

This predictability gave the opportunity to develop an electro-mechanical FEM model. With this model the modal behavior of the TTS was visualized. It turned out to be a good tool to tune the resonance frequencies and define the geometry of the TTS. Based on the model it was chosen to place the characteristic push-pull mode in the specified frequency range at 170 kHz. This results in a non flat response. However the amplitude of motion never drops below 1 µm per 100 Volt, since there are no anti-resonances present in this range.

In Figure 40 the measured response is depicted, together with its modeled equivalent. Important observations are first of all the negative shift in frequency by a factor 1.5 of the modes related to the axial drive train at 33 kHz and 168kHz and secondly the decrease of amplitude by a factor 4.
The observed frequency shift corresponds to a stiffness loss of a factor 2. By a more detailed analysis of the holder, 10% of the shift could be declared. Other causes are not yet found, although a presumption is made towards the contact stiffness between the capillary and its holder.

The decreased amplitude level was mainly the effect of the non linearity of the actuators. At higher voltages the piezo actuator becomes more compliant and is having an exponentially increased stroke. At full excitation the off resonant stroke was determined as 1 µm, which is no more than half of the specified actuator stroke of 2.2 µm.

The cause of this loss was in the first place attributed to the actuators that were measured apart from the TTS and showed a maximum stroke of only 1.7 µm, due to degradation during the test phase. Furthermore a significant part of the stroke was lost in exciting the counter masses. At 1 V these losses were measured to be 13%, however with the increased hysteresis at higher voltages this percentage will grow.

Final conclusion was found in the thermal behavior of the TTS. The measured temperature increase during actuation turned out to be less than initially calculated. A maximum theoretical duty cycle of 10% at 200 kHz full stroke was specified to be thermally safe. The thermal measurement indicates a safe duty cycle of maximal 13%. When using this type of drive commercially, advanced cooling methods are thus required to lift the duty cycle to 21%, corresponding to 48k operation. For the TTS this duty cycle is sufficient.

**8.2 Recommendations**

- In order to obtain the same resulting mechanical amplitude it is useful to flatten the dynamical response, by means of control. This can either be done by feedback control, or inversed plant feed forward control. The latter one is easier
to implement, but its robustness is questionable due the found non linearity’s. In order to still use this method, learning feed forward control can be implemented, due to the repeatability of the TTS. Feedback control deals better with these non linearity’s, but is quite challenging due to the high mechanical bandwidth.

- To verify the behavior of the TTS under loading conditions, the sensitivity to this loading should be investigated. Due to the high force it is not likely that the amplitude of the actuators will diminish. However the connection of the capillary to the holder can be tested this way.

- Another option is to replace the present capillary holder by a press fit holder. With this holder an explanation might be found for the loss of stiffness and the frequency shift, since the contact stiffness of these holders are better predictable. A drawback of this solution is that the fact this press fitted capillary cannot be disassembled once it should be replaced.

- It is advisable to electrically isolate the piezo actuators. Although the contact surfaces are already are isolated, still the non-isolated surfaces can cause sparking to the metal structure.

- Recommended for driving the TTS is to use a charge controlled amplifier instead of voltage controlled one. By this charge control the mechanical hysteresis is avoided, resulting in a predictable and linear response between set point and displacement. By avoiding the hysteresis the two piezos will move synchronously, which increase the stroke of the capillary. Especially at higher voltages, where the hysteresis becomes more significant, this solution is more preferable. Note that the energy loss related to the hysteresis will be still present. Besides this positive effect on the amplitude the charge control is causing a decrease of amplitude as well, since the actuator to become mechanically stiffer, due to electromechanical coupling. Positive of this increased stiffness is the improved bandwidth.

- Interesting for future use is to apply the option of self-sensing in the TTS, to measure the motional amplitude of the vibration. This should be compared with the laser vibrometer to verify this sensor option. This sensor option can be used to monitor the bonding or can even be used as feedback.
9. References


[29]. F&K. [Online]
Appendix A. Traditional bonding

A.1 Bond cycle

To present a more detailed view on this bonding process, Figure 51 shows the complete bond cycle of the PhiCom to form a single wire bond. For every wire these steps are repeated. With the current 36 k products per hour, two wires connections are made in 100 ms. The actual bond time is typically 4 ms per bond, yielding a duty cycle of the transducer of 16% for 2 wires products.

Figure 51: Wire bond process

0. Positioning of die

First the die has to be brought in position. To do this a motor is pulling the product carrier a small forward by using an index pin, to the next point where a die is located. A brake is creating pressure on the lead frame to make sure that the position of the lead frame has no overshoot. Using a camera system and recognition patterns, the exact location of the die can be determined.
1. Ball bond

The first connection that has to be made is the ball bond, where the free air ball (FAB) located at the tail of the wire has to be welded to the die at the registered position. The wire is brought in position by moving the XY-stage on which the instrument is mounted. Together with the XY movement the downward motion starts. This Z movement, done by a voice coil, is making the touch down of the ball onto the die pad. During this trajectory the transducer already starts to vibrate gradually. This is called (pre-ignition). The reason for doing this is that one can save some time that it takes to build up the vibrations during the bonding. After the touch down the ball is welded to the die pad. Due to the combination of heat, pressure due to the Z-coil and the ultrasonic energy the ball deforms and diffusion of metals takes place at the weld surface. When the welding is done a rigid connection is the result.

2. Wire loop

From the die pad the wire has to make a loop to the connection point of its lead frame. This point is called the wedge pad. The wire is not directly moved towards its destination, but an actual loop is made to avoid high stresses in the bonds. This movement is again a combination of the motion of the XY-table together with the Z movement.

3. Wedge bond

The bond on the wedge pad is different compared to the ball bond. Instead of a preformed ball that is welded, a part of the wire is squeezed between wedge pad and capillary. Again the ultrasonic movements are used to create the weld. The sharp edge of the capillary smears the wedge towards the side and creating already a small cut in the wire, which makes it easier to break it.

4. Break the wire

The transducer is moved upwards for few millimetres. Then a wire clamp located above the transducer is clamping the wire. When moving the whole device together further upwards the stress in the wire is causing it to break at the end of the wedge. The result is a part of the wire that is still sticking out of the capillary, in other words the tail to spark the next ball at.

5. Bring home

The last step of this connection is to bring the transducer back in the original spark position to start the next bond.
6. Sparking

As soon as the transducer is back in its original position a new gold ball is formed at the end of the wire, the tail. The process of sparking does this FAB formation. An electrode is moved towards the tail of the wire, which is sticking out of the capillary for a few millimetres. When the distance between both becomes small enough, the break down voltage is reached. At that point the electrode is discharged and the heat caused by the spark is melting the wire locally. The surface tension will deform the free end of the wire into a ball shape. Depending on spark time and provided energy the ball will get its final dimensions.

Due to the air flow parallel to the wire, the ball is retracted until the ball touches the capillary and is kept there under tension.

7. Repeat cycle

For the second connection that has to be created steps 1 till 6 can be repeated in the same sequence. When all connections are made, the next die has to be brought in position, which means step 0 should be included then as well.

The resulting product is depicted in Figure 52. It shows the gold wire loops connecting the die with the lead frame.

![Figure 52: Bonded product](image)

A.2 Variation in process

All commercial wire bonders have basically the same architecture, since they all use the principle of an US transducer. There are only a few variations or choices in the process, mostly depending on the type of product.
<table>
<thead>
<tr>
<th>Process settings</th>
<th>Typical values</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Frequency</strong></td>
<td>65 kHz</td>
</tr>
<tr>
<td><strong>Temperature</strong></td>
<td>310 °C</td>
</tr>
<tr>
<td><strong>Bond force</strong></td>
<td>300 mN</td>
</tr>
<tr>
<td><strong>Amplitude</strong></td>
<td>300 nm</td>
</tr>
<tr>
<td><strong>Bond time</strong></td>
<td>5 ms</td>
</tr>
<tr>
<td><strong>Current</strong></td>
<td>50 mA</td>
</tr>
<tr>
<td><strong>Voltage</strong></td>
<td>500 mV</td>
</tr>
<tr>
<td><strong>Wire</strong></td>
<td>Au, 25 µm</td>
</tr>
</tbody>
</table>

- **Frequency**: 65 kHz
  - Discussed in paragraph 1.1
- **Temperature**: 310 °C
  - It is common to use elevated temperatures during bonding. A large advantage is the increased process speed. However there are also drawbacks like an increased susceptibility to contaminants, migration of atoms and thermal loading/aging of the machine.
- **Bond force**: 300 mN
  - It improves the connection when the bond is pressed on the target surface. Logically if this force becomes too large the bond is squeezed too much and becomes weak and the bond pad is damaged. This holds as well for the touch down on the surface.
- **Amplitude**: 300 nm
  - A higher amplitude is related to more energy. However the bond should not be pulled off its contact. When contact is made, the amplitude should be seized by the internal deformation of the bond. As a comparison a ball bond is typically 50 µm.
- **Bond time**: 5 ms
  - It is clear that more energy is brought in the bond when the bond time is increased. This leads to higher process times.
- **Current**: 50 mA
  - The current is directly related to the amplitude of the bond process, but is not controlled.
- **Voltage**: 500 mV
  - The voltage is the controlled parameter. Increasing the voltage increases the current and amplitude.
- **Wire**: Au, 25 µm
  - The wire thickness and material can be varied. For longer distances thicker wire can be used. Common used materials are aluminium, copper and gold, where gold is often preferred because of its excellent conductivity and the resistance to oxidation. For the other materials inert gas is necessary to avoid a thick
Traditionally bonding oxidation layer\cite{19}.

- **Capillary**  
  \[ \text{Al}_2\text{O}_3, 11.1 \text{ mm} \]  
  Depending on the ball size and material, different geometries of capillaries are used. Especially the tip of the capillary can be dimensioned in various ways by for example modifying hole radius, outside diameter and the chamfer dimensions. To withstand the operating conditions, the capillary is made of a ceramic (e.g. Aluminum oxide), optimised for high temperatures and long lifetime. In Appendix G.3 a datasheet of the capillary is shown.

- **Bond type**  
  Common bonds are the ball bond and wedge bond. For ball bonds a specific problem can occur in the neck of ball, thus the transition between ball and wire. This relative weak part undergoes a lot of heat and forces during the welding and is therefore critical. Wedge bonds have their weak spot at the heel of the bond. This is where the weld ends and the wire starts. Another important issue is that the direction of the wire after a wedge bond is in the direction of the wedge, to avoid shear stresses in the weld. These shear stresses will occur when the two wedge bonds connecting one wire are slightly misaligned.

### A.3 Bond measurement

One of the measurements done during bonding of a transistor by the current 65 kHz process can be seen in Figure 53. The transistor requires two wire connections. Therefore 4 bonds have to be made. In chronological way these are a ball, wedge, ball and another wedge bond. Figure 54 shows a close up of the ball bond showing the resonating system.
Figure 53: Bond cycle of a transistor showing four bonds for two wires

Figure 54: Ball bond measurement showing the buildup of current and mechanical amplitude
Appendix B. Challenges for future design

B.1 High frequency bonding

One might think that the energy transferred to the wire is causing it to melt locally to form a bond. In fact there is only a local temperature rise in the material of around 150 degrees. The energy put in the ball is in the first place deforming the material which leads to strain hardening. Then microscopic slip planes shear across each other. This scrubbing removes the oxide layer and thereby provides clean new surfaces. At that moment diffusion of atoms takes place, forming the intermetallic compound between the ball and the bond pad.

Recent investigations [20][13][1] have shown that high frequency (>65 kHz) wire bonding has some important advantages with respect to traditional bonding. Main reason for this is that thermal energy is annealing the material, which makes it softer, whereas ultrasonic energy has a hardening effect as explained before. This makes that bonding with high frequencies is more efficient, since there is less deformation and thus frictional energy loss within the material [21]. This increase in efficiency has a positive effect on:

- The bond time, which is reduced because the actual diffusion starts earlier and the maximum amplitude is reached sooner.
- The intermetallic layer, where a larger percentage of the surface is welded and thus has increased bond strength.

Other important reasons for choosing high frequency are:

- Lower temperatures can be used when frequency/ ultrasonic energy is increased
- Amplitude of ultrasonic movement can be decreased, which can make fine pitch bonds and require only ramp up time.
- Less critical lead clamping, because the weld can resist higher stresses
- Easier bonding on different surfaces, which can be related to the more efficient energy transfer
- Improved ball shape / uniformity of ball and less based on experiments of Kulicke and & Soffa
- Less deviation in bond quality, also based on the same experiments.
Besides advantages one can also find disadvantages for high frequency:

- Decrease of process window, since the sensitivity to the bond parameters has increased, because the process is more relying on the ultrasonic energy. Therefore a very tight process control is necessary for both settings as equipment and material.

- Operating at higher frequencies demands a robust system and a more sophisticated ultrasonic generator, because more parasitic resonances are present at these frequencies.

### B.2 Decreasing mass

The currently developed Phicom, should be capable of reaching a speed of 36kunits per hour. However the mass of the transducer including its yoke, turned out to be a critical factor already. In the horizontal directions the moving mass is of course dominated by the complete xy-table, with all its instruments. For the z-movement, the inertia of this transducer assembly is the limiting factor. The inertia of the complete yoke is 2100 gcm$^2$ [22]. With this inertia, the first troublesome eigenfrequency is located at 800 Hz according to the measurement of Figure 55. This frequency corresponds to the US transducer rotating on its clamping membrane.

The spark down and spark up movement (par. A.1) are the critical trajectories is the bond cycle for the z-movement, since 5 mm has to be traveled in less than 5 ms for 36 k operation [23]. Based on [23] accelerations of 1.1km/s$^2$ are required to meet this requirement of 6 ms, where currently only half of this acceleration is used in practice. For 48 k operation the spark down time should be decreased even further. The electrical currents required for these accelerations will exceed the 10 A. At the moment these are 3.2 A. It is questionable whether these currents can demagnetize the magnets and if the coil needs to have additional cooling.
Figure 55: Encoder displacement in [m] over Voice Coil Force in [N] for current transducer yoke [22]
Appendix C. Piezo fundamentals

C.1 Introduction

The word piezo is derived from the Greek word for pressure. Jacques and Pierre Curie found out that a quartz crystal was generating an electric potential when a pressure was applied. This is called the direct piezoelectric effect. Later on one discovered that the effect was reversible. The material deformed when an electric field was applied. This is known as the converse piezoelectric effect. Besides quartz other materials are discovered during the years that have this same property:

- Natural crystals like Rochelle salt
- Natural textures such as bones and wood
- Synthetic piezo ceramics such as Lead Zirconate Titanate (PZT) and Lead Lanthanum Zirconate Titanate (PLZT)
- Synthetic polymers such as Polyvinyl difluoride (PVDF)

The reason for these materials having this piezoelectric effect is related to their molecular structure. The piezo molecules are charged, because of the asymmetry of positive and negative ions inside the molecule. As an example a PZT molecule is shown in Figure 56. Above the Curie temperature this molecules is still isotropic and do not possess the piezoelectric properties (the perovskite structure, Figure 56,1). When the material cools down below the Curie temperature the structure is changed in a tetragonal (ferroelectric) structure (Figure 56,2). At that point the asymmetry is creating a dipole of the molecule.

Figure 56: PZT molecule (1) before poling (2) after poling[5]
These dipoles do not directly lead to a macroscopic piezoelectric effect, since the molecules are not oriented all in the same direction. They are randomly oriented in groups, so called Weiss domains. To gain the piezoelectric properties, the dipoles, have to be aligned. In some materials this happens naturally, but like in PZT poling is necessary to orient the Weiss domains. During poling the material is held in a strong electrical field (>2kV/mm) under elevated temperatures.

After poling, the Weiss domains rotate slightly such that the material becomes neutrally charged again. Still the major direction is maintained. At this point a mechanical stress or electrical field can disturb the macroscopic charge balance of the piezo material. This is causing the piezo crystal to deform in case of an electrical field (Converse piezoelectric effect, Figure 57) or to be charged in case of a pressure (Direct piezoelectric effect, Figure 57).

![Piezoelectric Effect](image.png)

Figure 57: Piezoelectric effect [24]

### C.2 Constitutive laws

The piezoelectric relations can be described as well in mathematical form. The constitutional laws for a piezo material are in strain-charge format according to the IEEE standard on piezoelectricity [18].
\[ S = s_E \cdot T + d_{33}^T \cdot E \]  
\[ D = d_{33} \cdot T + \varepsilon_T \cdot E \]

where the following variables are used:

- **S** in [-] 6x1 strain vector
- **T** in [N/m²] 6x1 stress vector
- **E** in [V/m] 3x1 electric field vector
- **D** in [C/m²] 3x1 electric displacement vector

And the following constants:

- **s_E** in [m²/N] 6x6 compliance matrix with constant electrical field
- **d** in [m/V] or [C/N] 3x6 piezo stress coefficient matrix
- **\( \varepsilon_T \)** in [F/m] 3x3 dielectric coefficient matrix with constant stress

For PIC255 used for modelling the transducer the constants are [7]:

\[
s_E = \begin{bmatrix}
16.1 & -5.7 & -7.4 & 0 & 0 & 0 \\
-5.7 & 16.1 & -7.4 & 0 & 0 & 0 \\
-7.4 & -7.4 & 20.7 & 0 & 0 & 0 \\
0 & 0 & 0 & 44.9 & 0 & 0 \\
0 & 0 & 0 & 0 & 44.9 & 0 \\
0 & 0 & 0 & 0 & 0 & 43.2
\end{bmatrix} \cdot 10^{-12} \left[ \frac{m^2}{N} \right] \quad (C.3)
\]

\[
d = \begin{bmatrix}
0 & 0 & 0 & 0 & 550 & 0 \\
0 & 0 & 0 & 550 & 0 & 0 \\
-180 & -180 & 400 & 0 & 0 & 0
\end{bmatrix} \cdot 10^{-12} \left[ \frac{C}{N} \right] \quad (C.4)
\]

\[
\frac{\varepsilon_T}{\varepsilon_0} = \begin{bmatrix}
1650 & 0 & 0 \\
0 & 1750 & 0 \\
0 & 0 & 1750
\end{bmatrix}, \varepsilon_0 = 8.854 \cdot 10^{-12} \left[ \frac{F}{m} \right] \quad (C.5)
\]

The selected actuator is a multilayer linear actuator. That means they are actuating in the same direction as they are poled. The poling direction is always indicated by the index number 3 [18]. Thus linear actuation in this direction will be indicated by 33.

Due to the stacking of the small layers of around 50 \( \mu \)m of the multilayered actuator, the actuation can be considered as purely linear. Therefore the equations (C.1) and (C.2) can be used in one dimension only resulting in the simplified scalar equations of (C.6) and (C.7)

\[ S = s_E \cdot T + d_{33}^T \cdot E \]  
\[ D = d_{33} \cdot T + \varepsilon_T \cdot E \]

80 Piezo fundamentals
C.3  Piezo-electrical behaviour

In the previous paragraph the constitutional laws are given that describe the physical behaviour of a piezo element. This paragraph will elaborate further on how these laws express themselves in the usage of piezo material in an actuator. For a better understanding the equations are dimensioned by surface area (A) and thickness (dx) resulting in the following variables.

- **Force F in [N]**: \( f = \frac{T}{A} \)  \( (C.8) \)
- **Charge q in [C]**: \( q = D \cdot A \)  \( (C.9) \)
- **Displacement x in [m]**: \( x = S \cdot dx \)  \( (C.10) \)
- **Voltage V in [V]**: \( V = \frac{E}{dx} \)  \( (C.11) \)

Then the constitutional laws can be rewritten to (C.18) and (C.19):

\[
x = \frac{A}{s_E \cdot dx} \cdot F + d_{33}^T \cdot V \quad \text{(C.12)}
\]
\[
q = d_{33} \cdot F + \frac{dx}{A \cdot \epsilon_T} \cdot V \quad \text{(C.13)}
\]

By introducing the stiffness \( k \) in [N/m] and capacitance \( C \) in [F] as constants leads to expressions (C.20) and (C.21). These are the finite descriptions of piezo elements.

\[
x = \frac{1}{k} \cdot F + d_{33} \cdot V \quad \text{(C.14)}
\]
\[
q = d_{33} \cdot F + C \cdot V \quad \text{(C.15)}
\]

C.3.1 Voltage control

Looking at the electrical domain (C.15) it can be seen that a piezo can be roughly modelled as a capacitor. However, the interaction with the mechanical domain is generating a charge as well, even if no external mechanical force is applied. The force can be obtained from rewriting (C.14) in (C.16). Inserting this equation back in (C.15) gives the expression for the charge \( q \) according to (C.17).

\[
F = k \cdot x - k \cdot d_{33}^T \cdot V \quad \text{(C.16)}
\]
\[
q = (k \cdot d_{33}^2 + C)V + d_{33} \cdot k \cdot x \quad \text{(C.17)}
\]

Figure 58 shows how the charge is build up in case a voltage step is applied. Besides the direct charge voltage relation, the second branch shows the induced counteracting charge. In this figure the external displacement is assumed to be constant and is not depicted.
The level of coupling between the direct electrical charge and the mechanically induced charge can be described by introducing the piezo electric coupling factor $k_f^2$ (C.18). This factor $k_f^2$ is typically 0.4-0.7[2].

$$
k_f^2 = \frac{\varepsilon_{33}}{k \cdot C}
$$

(C.18)

This leads to a charge on the piezo element according to (C.19)

$$
q_p = C(1 - k_f^2) \cdot V
$$

(C.19)

The induced charge on the electrodes of the piezo element is brought on the electrodes by an induced current ($I_{ind}$) and is causing an induced voltage ($V_{ind}$). Thereby the current on the piezo ($I_p$) and the voltage ($V_p$) are described by respectively (C.20) and (C.21)

$$
I_p = I - I_{ind} = C \frac{dV}{dt} - \frac{dq_{ind}}{dt} = \frac{dq_p}{dt}
$$

(C.20)

$$
V_p = V - V_{ind} = V - \frac{q_{ind}}{C} = \frac{q_p}{C}
$$

(C.21)

The induced voltage related to the piezo element makes that a piezo is often represented as a capacitor in series with a voltage source as in Figure 59.

![Figure 58: Build-up of charge](image)

Figure 58: Build-up of charge

It should be realized that this induced voltage is a reaction on a voltage change. This reaction is as fast as the response of the mechanical system. From (C.22) the response time ($t$) is approximately:

$$
t = \frac{1}{3f_0}
$$

(C.22)
This means with a 600 kHz resonance frequency, only the first µs an induced voltage will be present.

In case of a voltage controlled system, the voltage delivered by the amplifier (V) is increased during this time to make sure that the voltage over the piezo (Vp) is constant and the error $\varepsilon$ disappears, according to Figure 60. A fast amplifier having a high gain (G) and typical slew rates in the order of 100V/µs-1000V/µs. Since this is faster than the mechanics, the error cannot build up, such that hardly any difference is seen between V and Vp.

![Voltage controlled amplifier](image)

As a consequence of this control, the current trough the amplifier will increase, due to the compensated induced current. The current will thus be described according to:

$$I = I_p + I_{ind} = C \left( \frac{dV_p}{dt} + \frac{dV_{ind}}{dt} \right)$$ \hspace{1cm} (9.23)

### C.3.2 Dynamical response with voltage control

The previous model of Figure 58 was suggesting a direct relation between the voltage step and the displacement x to get a first understanding of the piezoelectric effect. To use the model dynamically the mechanical response should be incorporated in the model. Introducing damping (c) and mass (m) the displacement of the piezo as a function of the input voltage can be described as in (C.24).

$$\frac{x}{V} = d_{33} \cdot \frac{cs + k}{ms^2 + cs + k}$$ \hspace{1cm} (C.24)

Including this function in the diagram leads to Figure 61. The electrical model of Figure 59 is adapted to include the dynamical behavior. In Figure 62 (l) the mechanical branch is added replacing the voltage source. Figure 62 (r) shows an electrical equivalent model by replacing the mass by a second capacitor, the spring by an inductor and finally the damper by a resistor.
Figure 61: Charge build-up including dynamics of actuator

Figure 62: Electromechanical representation of piezo actuator (left) and equivalent electrical model of piezo actuator (right)

At low frequencies the mechanics are still following the applied voltage, or in other words the phase delay is zero. This can be seen in Figure 63. In case of voltage control it was shown that this induced voltage was compensated by a higher current. Indeed Figure 64 shows the graph including this induced current on top of a current graph without this induced current.

In case the piezo is approaching the resonance frequency, the amplitude builds up as can be seen in Figure 63. Due to the risen amplitude the internal force in the actuator will increase as well. By the relation of the piezo stress coefficient, an even higher induced charge is obtained resulting in the peaking of the current.

Just after the resonance frequency the phase is shifted 180 degrees, implying an induced current which contributes to the applied voltage. As a response the amplifier is decreasing its current. Then the mechanics are decoupled, there is no amplitude and induced current anymore, resulting the overlapping graphs at these frequencies.
C.3.3 Current control

Starting again with formula’s (C.14) and (C.15) and rewriting these to find an expression for the force and the voltage, (C.25) and (C.26) are found.

\[ F = k \cdot x - k \cdot d_{33}^T \cdot V \]  
\[ V = -\frac{d_{33}}{C} \cdot F + \frac{1}{C} q \]  

(C.25)  
(C.26)

In case there is no external force the relation between \( q \) and \( V \) is given by C again. However now the mechanical behavior is modified, since the voltage is not constraint. Substituting (C.26) in (C.25) leads with the introduction of the piezoelectric coupling factor \( k \) to (C.27). In Figure 65 the relation of the force and displacement in case of current control is visualized.

\[ x = \frac{1}{k} \left( 1 - k_f^2 \right) \cdot F + \frac{d_{33}}{C} \cdot q \]  

(C.27)

Figure 63: Stroke in [m] over voltage in [V]  
Figure 64: Current in [A] over voltage [V]

Figure 65: Build up of amplitude
By this additional stiffness the mechanical response in case of charge control results in a factor \(1 - k_f^2\) lower amplitude and a factor \(\frac{1}{1-k_f^2}\) higher eigenfrequency then in case of voltage control. In Figure 66 these two mechanical responses are shown.

![Figure 66: Amplitude in [m] over force in [N] for a voltage controlled actuator versus a charge controlled actuator](image)

In case of current control not \(V_p\), but the current trough the piezo \((I_p)\) is kept constant by the (charge) amplifier. Thereby the voltage over the amplifier \((V)\) should be increased to compensate the induced voltage \((V_{ind})\)

\[
V = V_p + V_{ind} = \frac{1}{C} \int_0^t I_p + I_{ind}
\]

(C.28)

![Figure 67: Voltage in [V] over charge [C]](image)
Mechanically charge control is preferred when high bandwidths are required. However for precision system voltage control is favorable, because charge control has larger errors due to the amplifiers feedback.

### C.3.4 Practical based model of actuator

The model derived in the previous paragraph is based on the material properties of the piezo material (PIC252) of the selected actuator. One can however also base himself on the specified properties of the actuator. After all the actuator is not a homogeneous block of piezo material. Besides ceramic layers, also silver/palladium (AgPd) thick film electrodes are stacked in the actuator. These electrodes are around 10 µm [7] in a layer of around 50 µm[25]. The active length of the actuator is thus smaller than the nominal length. The consequence is that the dynamical behavior will differ from the material properties based model.

The specifications of the 2x2x2mm actuator were listed in Table 3 of 3.1.2. From actual measurements it is known that this type of actuator has its first resonance at 600 kHz. Assuming the mass will be similar to the theoretical mass, all global variables for the model can be obtained from the specifications resulting in Table 13. The elemental properties are depending on the number of elements (N). Important to realize is that the piezoelectric coupling factor is still related to the properties of the piezo material and should therefore not be recalculated according to the global specifications.

<table>
<thead>
<tr>
<th>Capacitance (C)</th>
<th>25 nF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stiffness (K)</td>
<td>90kN/mm</td>
</tr>
<tr>
<td>Mass (M)</td>
<td>62 mg</td>
</tr>
<tr>
<td>Piezo stress coefficient (D)</td>
<td>2.2µm/100V</td>
</tr>
<tr>
<td>Piezoelectric coupling (k)</td>
<td>0.71</td>
</tr>
</tbody>
</table>

Table 13: Specifications practical model

In Figure 68 a comparison is done between the currently derived model and the model based on material specifications. This last mentioned model is based on 55 elements to make sure that the specified amplitude is indeed obtained. This amplitude is thus below its resonance for both models -153 dB, which corresponds to 2.2µm/100V. From the figure it becomes clear that stacking the layers with electrodes is not influencing the dynamical behavior. The resonance shifts from 622 kHz to 600 kHz. For modeling the complete mechanical system used in Chapter 5 this practical model will be incorporated.
C.4 **Practical issues piezo actuators**

C.4.1 **Hysteresis and non-linearity**

The previous equations showed linear relations between the different variables. In practice a phenomenon called hysteresis occurs in the conversion of voltage to displacement. Apparently there is a delay before the Weiss domains orient when applying an electrical field. In Figure 69 a typical hysteresis curve is shown. In case of current control this hysteresis effect is not visible in the displacement, since there is a linear relation between current and displacement. However the energy losses related to the hysteresis effect are still present. This loss of energy of typically 5-20% at full stroke is depending on the type of material. ‘Hard’ PZT actuators, which is harder to polarize, having the lower losses, but less stroke, where the ‘soft’ PZT actuators can reach higher strokes, but is having higher losses.

Another remarkable observation is the non-linearity of stroke versus drive voltage seen in Figure 69 as well. The stiffness of the piezo is clearly higher for small voltages than for high voltages. A drive voltage of 50 V results in a stroke of 11µm, where 100 V lead to 25 volt, which is an increase of 13% in compliance.
C.4.2 Depolarization

In the previous paragraph the operating curve of the actuator was shown. However to have a complete view on the operational area the butterfly curve of Figure 70 is displayed. This curve shows the limits of the piezo actuator.

When the material is subjected to a (large) negative voltage, the piezo elements will depolarise at $P_1$. By increasing this negative voltage much further to several kV/mm will repole the piezo in the other direction. When operating the piezo in the left halve depolarization takes place at $P_2$. When driving a piezo dynamically a voltage offset is thus necessary.
C.4.1 Delaminating of piezo actuator

At higher frequencies where inertia of the elements starts to play a role these forces appear, during the moment the piezo is contracting again after being extended. These forces can delaminate the layers of the piezo actuator.

To use the actuator dynamically without damaging the actuator a preload force is used. This force is compressing the actuator with half its stroke. Thereby it is made sure that the actuator will not exceed its nominal length and will always be subjected to compressive forces. An exception is when the actuator starts to resonate. Then the mechanical amplitude is larger than the specified (off-resonant) amplitude, resulting in tensile forces and possible breakdown of the actuator. When applying a preload it is desirable to have a constant force. A force delivered by a spring will introduce a parallel stiffness to the actuator. This will decrease the stroke of the actuator.
Appendix D. Assembly and manufacturing of TTS

D.1 Assembly

The design up till now consists of five different parts namely the framework, the capillary holder, the capillary and the two piezo actuators. All these parts should be assembled manually. The design is based on symmetrical actuation. This means that all components should have proper tolerances. But even more important is to position them correctly. Therefore dowel pins will be used to position the holder with respect to the framework. To prevent over constraining the contact, one hole is a slotted hole. Once in position a screw can tighten the two smooth surfaces against each other.

To align the piezos properly in all directions requires more creativeness. The dimensions of these parts are so small an assembly tool is required. This tool shown in Figure 71 can be mounted onto the framework. Then the piezos can be shifted into position by pushing it against a small ridge.

![Figure 71: Assembly tool](image)

This tool can be used as well for adjusting the length of the capillary which is sticking out. At the place the capillary is located a threaded hole is made. With a bolt the depth of the hole can be adjusted and thereby the length of the capillary sticking out changes too.

When the piezos and capillary are placed satisfactorily, the preload has to be applied to enclose the vibrating system. With a 0.5 mm pitch of the M3 it requires 3 turns to reach the preload of 60 N. Based on the force limits of the leaf spring (5.8) it is required to apply this load slowly starting with steps of maximum 6N, which is almost 1/3 of a turn.
D.2 Manufacturing

During the design of the setup it has been of constant attention that it can be made without difficult manufacturing processes. Complex (3d) geometries are avoided saving time and money. The required processes for the completion of this setup are:

- Milling - the contours of the framework
- Turning - the dowels for the preload spring
- Drilling - the framework and the capillary holder
- Wire sparking - the counter masses and the capillary holder

After manufacturing and assembly the setup looks as shown in Figure 72. Other photos containing a zoomed out view can be seen in 0.
Appendix E. Finite element models

E.1 Introduction to FEM

E.1.1 Type of element
The model for the transducer will be based on Finite Element Modelling (FEM). This means the complete structure is divided into small elements with certain elemental properties. Since we have to deal with mechanical and electrical behaviour, both types of properties have to be assigned to the designated elements.

But first the type of element has to be selected. An element is described by a number of nodes. For linear movement 1d elements with 2 (end) nodes are usually sufficient, but for 3d movements, where volumes are involved, 3d elements with multiple nodes are required. Every node has certain degrees of freedom (DOF’s) for example a translation in x, and a rotation around the z-axis.

The DOF’s of the nodes are described by shape functions, where a local coordinate system is used. For the simplest case of an element that can only move in one direction, a rod element, the shape functions can be described as:

\[ N = \begin{bmatrix} \phi_1(q) \\ \phi_2(q) \end{bmatrix} = \begin{bmatrix} \frac{x}{l} \\ 1 - \frac{x}{l} \end{bmatrix} \]  

(E.1)

with degrees of freedom q:

\[ q = \begin{bmatrix} u_1(t) \\ u_2(t) \end{bmatrix} \]  

(E.2)

Then:

\[ u_n(x, t) = N(x) \cdot q(t) \]  

(E.3)

In case the element has to bend as well around an axis, this set of degrees of freedom is not sufficient. The shape functions become already more complex, because the element is described by 2 translations and two rotations.

E.1.2 Stiffness and mass matrix
Mechanically a structure can be decomposed into a number of masses, connected with springs and dampers. The shape functions are necessary to assign a stiffness and mass
Finite element models

to the DOF’s of all nodes of a certain element. The stiffness for a single element is then defined as

$$K = \int_0^l EA \cdot \left( \frac{dN^T}{dx} \right) \left( \frac{dN}{dx} \right) dx$$  \hspace{1cm} (E. 4)

and the mass is:

$$M = \int_0^l \rho Al \cdot N^T N \, dx$$  \hspace{1cm} (E. 5)

Solving these integrals for a simple 1 DOF element one will find a stiffness and mass matrix:

$$K = \frac{EA}{l} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}$$  \hspace{1cm} (E. 6)

$$M = \frac{\rho Al}{6} \begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}$$  \hspace{1cm} (E. 7)

For a beam element the mass and stiffness matrices are already not unique anymore. The reason for this is that there is not just elongation that takes place, but as well bending and shear. When omitting the shear one finds the expression of an Euler Bernoulli beam that can only bend in one direction:

$$K = \frac{E l}{l^3} \begin{bmatrix} 12 & 6l & -12 & 6l \\ 6l & 4l^2 & -6l & 2l^2 \\ -12 & -6l & 12 & -6l \\ 6l & 2l^2 & -6l & 4l^2 \end{bmatrix}$$  \hspace{1cm} (E. 8)

$$M = \frac{\rho Al}{6} \begin{bmatrix} 156 & 22l & 54 & -13l \\ 22l & 4l^2 & 13l & -3l^2 \\ 54 & 13l & 156 & -22l \\ -13l & -3l^2 & -22l & 4l^2 \end{bmatrix}$$  \hspace{1cm} (E. 9)

If shear is involved the cross section of the beam is not perpendicular to the neutral axis, but is corrected for the shear stress in the beam. The beam is therefore undefined and the stiffness matrix will look as follows:

$$K = K_1 + K_2 = \frac{E l}{l^3} \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & -1 \\ 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 1 \end{bmatrix} + \frac{\mu GA}{4l} \begin{bmatrix} 4 & 2l & -4 & 2l \\ 2l & l^2 & -2l & l^2 \\ -4 & -2l & 4 & -2l \\ 2l & l^2 & -2l & l^2 \end{bmatrix}$$  \hspace{1cm} (E. 10)

Here $\mu$ is a geometrical factor that is a measure for the shear contribution. For solid rods this value is around 0.88[26], depending on the Poisson ratio. However for slender thin
beams this effect of shearing is negligible and one can make use of the Euler Bernoulli beam.

In case of the extended model, where the drive train consists of 3 degrees of freedom: z, x and θ to create both bending and elongation, the previous matrices should be merged to obtain a stiffness and mass matrix. The stiffness matrix will then look like:

\[
\begin{bmatrix}
\frac{EA}{l} & 0 & 0 & -\frac{EA}{l} & 0 & 0 \\
0 & \frac{12EI}{l^3} & \frac{6EI}{l^2} & 0 & -\frac{12EI}{l^3} & \frac{6EI}{l^2} \\
0 & \frac{6EI}{l^2} & \frac{4EI}{l} & 0 & -\frac{6EI}{l^2} & \frac{2EI}{l} \\
-\frac{EA}{l} & 0 & 0 & \frac{EA}{l} & 0 & 0 \\
0 & -\frac{12EI}{l^3} & -\frac{6EI}{l^2} & 0 & \frac{12EI}{l^3} & -\frac{6EI}{l^2} \\
0 & \frac{6EI}{l^2} & \frac{2EI}{l} & 0 & -\frac{6EI}{l^2} & \frac{4EI}{l}
\end{bmatrix}
\]  
(E.11)

The various types of elements are the building blocks for a FEM model. To model the desired structure the different elements have to be assembled in the right way by connecting corresponding nodes to each other.

### E.1.3 State space model

To see the frequency response function (FRF) of the system it is useful to make a state space notation. The state space notation can always be reduced to the following format

\[
x(t)' = Ax(t) + Bu(t) \\
y(t) = Cx(t) + Du(t)
\]  
(E.12) (E.13)

For a mechanical structure with mass matrix M and stiffness matrix K this way the notation makes it possible to have only first order differential equations:

\[
x' = \dot{x} \\
\ddot{x}' = F - M^{-1}Kx
\]

In this case these states thus represent a position and velocity vector:

\[
\begin{bmatrix}
x' \\
\dot{x}'
\end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -M^{-1}K & 0 \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \end{bmatrix} + \begin{bmatrix} 0 \\ F \end{bmatrix}
\]  
(E.14)

The C matrix will then give the desired position or velocity of one of the nodes. For a three mass system as in Chapter 4 the position of the middle mass is obtained by the following C matrix:
In case of a direct feed trough the contribution of the D matrix should be added.

E.2 Electromechanical finite element modelling

To make a finite element description, the finite descriptions of a piezo element of C.3 are used:

\[ x = k_{xx} \cdot F + d_{33}^T \cdot V \] (E. 15)

\[ q = d_{33} \cdot F + k_{pp} \cdot V \] (E. 16)

To use the model in case of voltage controlled or charge controlled actuation the derived formats of C.3 will be used. To complete the equation for dynamic finite element modelling, the inertia force term and the damping term are added as well. This could be derived as well from the Hamilton principle as done in [27]. For the charge controlled format the matrix equation will then be:

\[ F = M \cdot \ddot{X} + \frac{1}{(1 - k^2)} K_{xx} \cdot X - d_{33} \cdot K_{xx} \cdot K_{pp}^{-1} \cdot Q \] (E. 17)

\[ V = -d_{33} \cdot K_{xx} \cdot K_{pp}^{-1} \cdot X + K_{pp} \cdot Q \] (E. 18)

Where matrices M, K_{xx}, K_{pp} are respectively the assembled mass, stiffness and capacitance matrices. These equations can be written in state space form as well to have only first order differential equations like is done in E.1.3:

\[
\begin{bmatrix}
\dot{X} \\
\dot{\dot{X}}
\end{bmatrix}
= 
\begin{bmatrix}
0 & 1 \\
-\frac{1}{(1 - k^2)} M^{-1} K_{xx} & 0
\end{bmatrix}
\begin{bmatrix}
X \\
\dot{X}
\end{bmatrix}
+ 
\begin{bmatrix}
0 & 0 \\
M^{-1} & M^{-1} \cdot d_{33} \cdot K_{xx} \cdot K_{pp}^{-1}
\end{bmatrix}
\begin{bmatrix}
F \\
Q
\end{bmatrix}
\]

\[
\begin{bmatrix}
X \\
V
\end{bmatrix}
= 
\begin{bmatrix}
I & 0 \\
-d_{33} \cdot K_{xx} \cdot K_{pp}^{-1} & 0
\end{bmatrix}
\begin{bmatrix}
X \\
\dot{\dot{X}}
\end{bmatrix}
+ 
\begin{bmatrix}
0 & 0 \\
0 & K_{pp}
\end{bmatrix}
\begin{bmatrix}
F \\
Q
\end{bmatrix}
\]

In the previous equation it was assumed the dimensions of electrical and mechanical matrices were equal. However there is no reason for having the same dimensions. It is possible to apply the electrical properties only on one direction (33) and having a mechanical system with more degrees of freedom. Furthermore it is possible to have a nodal mechanical description of the system, but an elemental electrical description, e.g. displacement of a point and voltage over an element.

Then it is necessary to use a transformation matrix T like in (E. 20) for example:
$$T = \begin{bmatrix} 1 & 0 & 0 \\ -1 & 1 & 0 \\ 0 & -1 & 1 \\ 0 & 0 & -1 \end{bmatrix} \quad (E.20)$$

Finally the state space system will be:

$$\begin{bmatrix} \dot{X} \\ \dot{\dot{X}} \end{bmatrix} = \begin{bmatrix} 0 & I \\ -\frac{1}{(1-k^2)}M^{-1}K_{xx} & 0 \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{\dot{X}} \end{bmatrix} + \begin{bmatrix} 0 \\ M^{-1} \cdot d_{33} \cdot K_{xx} \cdot T \cdot K_{pp}^{-1} \end{bmatrix} \begin{bmatrix} F \\ Q \end{bmatrix} \quad (E.21)$$

$$\begin{bmatrix} \dot{X} \\ \dot{\dot{X}} \end{bmatrix} = \begin{bmatrix} 0 & I \\ -d_{33} \cdot K_{xx} \cdot K_{pp}^{-1} \cdot T^T & 0 \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{\ddot{X}} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \begin{bmatrix} F \\ Q \end{bmatrix} \quad (E.22)$$

For the voltage controlled state space the same kind of derivation can be done leading to equation (E.22)

$$\begin{bmatrix} \dot{X} \\ \dot{\dot{X}} \end{bmatrix} = \begin{bmatrix} 0 & I \\ -M^{-1}K_{xx} & 0 \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{\dot{X}} \end{bmatrix} + \begin{bmatrix} 0 \\ M^{-1} \cdot d_{33} \cdot K \cdot T \end{bmatrix} \begin{bmatrix} F \\ Q \end{bmatrix} \quad (E.22)$$

### E.3 Hertz contact stiffness

The contact stiffness of the capillary in its holder is crucial for the dynamics of this tool. The response of the holder itself can be calculated using FEM without much uncertainty. However contact stiffnesses are usually harder to predict. By choosing a line contact a good approximation is taking Hertz theory to predict this stiffness between the body’s capillary (1) and holder(2). The properties of the hertz contact are listed in Table 14.

<table>
<thead>
<tr>
<th>Property</th>
<th>Abbreviation</th>
<th>Capillary (1)</th>
<th>Holder (2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elasticity modulus [GPa]</td>
<td>E</td>
<td>300</td>
<td>71</td>
</tr>
<tr>
<td>Radius [mm]</td>
<td>R</td>
<td>0.79</td>
<td>∞</td>
</tr>
<tr>
<td>Poisson ratio [-]</td>
<td>ν</td>
<td>0.30</td>
<td>0.33</td>
</tr>
<tr>
<td>Yield strength [Mpa]</td>
<td>R_{p0.2}</td>
<td>400</td>
<td>460</td>
</tr>
<tr>
<td>Force [N]</td>
<td>F</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td>Length of contact [mm]</td>
<td>L</td>
<td>3</td>
<td></td>
</tr>
</tbody>
</table>

**Table 14: properties Hertz contact**

The effective E modulus ($E'$) and effective radius are given by respectively (E. 23) and (E. 24).

$$\frac{1}{E'} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \quad (E.23)$$
\[
\frac{1}{R'} = \frac{1}{R_1} + \frac{1}{R_2}
\]  
(E. 24)

The half width length (b) is half the width of the contact area determined by (E. 25).

\[
b = \left(\frac{4}{\pi}\right)^{\frac{1}{2}} \left(\frac{F^'}{L}\right)^{\frac{1}{2}} \left(\frac{R'}{E'}\right)^{\frac{1}{2}}
\]  
(E. 25)

With the known contact surface the mean pressure (P_m) is calculated in (E. 26). This pressure distribution has a local maximum P_{max} in the middle of the contact according to (E. 27).

\[
P_m = \frac{F}{2 \cdot b \cdot L}
\]  
(E. 26)

\[
P_{max} = \frac{4}{\pi} \cdot P_m
\]  
(E. 27)

It is found out that the maximum shear stress reaches it maximum at circa 0.786b\[28\] beneath the contact surface. Based on this the critical shear stress (\(\tau_c\)) and critical pressure (\(P_c\)) before plastic deformation will occur are given by the expressions (E. 28) and (E. 29)

\[
\tau_c = 0.5 \cdot R_{p0,2} = 0.304 p_{max} = 0.387 p_c
\]  
(E. 28)

\[
P_c = \frac{0.5}{0.387} \cdot R_{p0,2}
\]  
(E. 29)

The indentation (\(\delta\)) of the capillary in the aluminium holder is given by (E. 30).

\[
\delta = \frac{2F}{\pi L} \cdot \frac{1 - \nu^2}{E_1} \cdot \left(\ln \left(\frac{R_1}{b}\right) - \frac{1}{2}\right)
\]  
(E. 30)

The nonlinear stiffness (k) of this contact can finally be obtained by the deriving the force with respect to the indentation. The resulting formula is given in (E. 31)

\[
k = F' (\delta) = \frac{L}{\frac{2}{\pi} \cdot \frac{1 - \nu^2}{E_1} \cdot \left(\ln \left(\frac{4 \cdot R_1}{b}\right) - 1\right)}
\]  
(E. 31)

The results for the given contact are listed in Table 15
<table>
<thead>
<tr>
<th>Property</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Half width length [µm]</td>
<td>b</td>
<td>17.7</td>
</tr>
<tr>
<td>Mean pressure [Mpa]</td>
<td>P_m</td>
<td>565</td>
</tr>
<tr>
<td>Maximum pressure [Mpa]</td>
<td>P_max</td>
<td>719</td>
</tr>
<tr>
<td>Critical shear stress [Mpa]</td>
<td>τ_c</td>
<td>200</td>
</tr>
<tr>
<td>Critical mean contact pressure [Mpa]</td>
<td>P_c</td>
<td>516</td>
</tr>
<tr>
<td>Indentation [µm]</td>
<td>δ</td>
<td>18.1</td>
</tr>
<tr>
<td>Stiffness [kN/mm]</td>
<td>k</td>
<td>371</td>
</tr>
</tbody>
</table>

Table 15: Results Hertz contact

Since the mean pressure exceeds the critical mean contact pressure there will be some local plastic deformation. The Hertz calculations are only valid for elastic deformation. However the exceedance of the critical pressure is so small the stiffness of 371 kN/mm can still be used for the model. In 6.4.1 the influence of a finite contact stiffness can be seen.

### E.4 Vertical modes of TTS

<table>
<thead>
<tr>
<th>Waveform</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>9 [kHz]</td>
<td>Counter masses move in opposite direction on its springs</td>
</tr>
<tr>
<td>10 [kHz]</td>
<td>Counter masses move in same direction on its springs</td>
</tr>
<tr>
<td>28 [kHz]</td>
<td>Rotating counter masses move in opposite direction on its</td>
</tr>
<tr>
<td>159 [kHz]</td>
<td>Capillary holder moves on its springs</td>
</tr>
</tbody>
</table>

Figure 73: Modes in vertical direction
# Appendix F. Measurement set-up

## F.1 Testing equipment

To perform the desired measurements the following equipment is used:

<table>
<thead>
<tr>
<th>Mechanical</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>• TTS</td>
<td>Experimental transducer test set-up</td>
</tr>
<tr>
<td>• Vibration isolating table</td>
<td>Since amplitude will be measured with a non contact measurement, all disturbing vibrations are reduced by this table</td>
</tr>
<tr>
<td>• Interface block</td>
<td>To create an interface with the table a robust block is used for this interconnection. For easy assembly this block contains a hole at the location of the preload spring bolt.</td>
</tr>
<tr>
<td>• Piezo clamping device</td>
<td>The amplitude of a single piezo is measured in this device, since this instrument has one fixed wall, making it easier to obtain the real stroke of the actuator.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Electrical</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>• Voltage amplifier (Appendix G.1)</td>
<td>Amplifies the control signal with a factor 5 to a maximum of $200V_{pp}$</td>
</tr>
<tr>
<td>• 2 Power supplies</td>
<td>Required to create a symmetric positive and negative supply of the amplifier. Maximum deliverable voltage per amplifier is 70 V.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Signal generation</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>• Network analyzer(G.4)</td>
<td>Used for finding the frequency responses of the TTS. It should be able to operate at 50-200kHz</td>
</tr>
<tr>
<td>• Splitter</td>
<td>A 50 Ω splitter is used to feedback the output signal to the reference channel</td>
</tr>
<tr>
<td>• Adapter</td>
<td>The adapter is converting the high Ohmic input to a 50 Ω input</td>
</tr>
<tr>
<td>• Function generator (Appendix G.5.1)</td>
<td>This instrument generates the control signal that drives the TTS</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Sensor</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>• Current sensor(G.7)</td>
<td>Measures the current by a Hall sensor</td>
</tr>
<tr>
<td>• Differential laser vibrometer (Appendix G.6)</td>
<td>Measure either velocity or displacement in single or differential mode</td>
</tr>
</tbody>
</table>
- Thermistor(0): Measures the temperature by a change of resistance
- Multimeter: Used for verifying output voltages of the different instrument and measures the capacitance of the piezo
- Vernier: Used for verifying the position of the capillary and the compression of the springs

**Signal processing**
- Scope: Reads out the signal coming from the voltage probe, current probe and laser
- Computer: Used to interface the setup to
- Processing software: Matlab scripts are used to control the function generator and retrieve data from the scope and network analyzer

**Cables & accessories**
- Gpib to USB: To connect network analyser to computer
- Coaxial cables: Used for all control and measured signals
- Banana cables: Used for powering the amplifier
- T end: Used to split the signals
- 50 Ohm Terminator: Used to get a 50 Ohm input

---

**F.2 Schematic design measurement set-up**

![Schematic diagram]

---

**Table 16: Schematic design measurement set-up**
F.3 Verification piezo actuator

Before the measurement setup is built according to 7.1, first the functioning of both actuators has to be ensured. The piezo actuators are measured in a piezo clamping device, schematically drawn in Figure 74. The two differential laser beams are exposed orthogonal to the contact surfaces of the piezo actuator. This setup is used since it was not obtainable in the experimental setup to get so close to the actuator.

![Figure 74: Schematic drawing piezo clamping device](image)

As can be seen from Figure 74 a preload is used as well in this device in the form of a M2 nylon bolt, decreasing the amplitude with approximately 4% according to:

$$\frac{k_n}{k_p} = \frac{E_n \cdot A_n}{E_p \cdot A_p} = \frac{3e^9 \cdot \frac{1}{4} \cdot \pi \cdot 1.7^2}{45e^9 \cdot 4} = 4\%$$

(9.29)

For measuring the amplitude a frequency of 1 kHz is used. This is still before the first eigenfrequency of the TTS. The piezo is actuated at different voltages resulting in the hysteresis plot of Figure 75. The larger the voltage, the larger the hysteresis loop will be. This hysteresis is also observable in the electrical domain as expected. The current is not perfectly 90 degrees shifted with respect to the voltage as would be for a pure capacitor. Instead power is dissipated as can be seen from Figure 76. The higher the voltage, the larger the smaller the power factor is. For 50 V actuation the dissipation is increased to 10%. The stroke at 50 V of 1.6 µm does not match the specified stroke of 2.2 µm±20% taking into account amplitude loss due to the preload.

The two piezo actuators selected for the TTS were able to reach an amplitude of respectively 830 nm and 886 nm at 50 V. In Figure 77 the hysteresis curves of the first actuator is shown together with its quadratic fit to determine the maximum amplitude. For this actuator a power spectrum is made according to Figure 78. This spectrum clarifies that hysteresis is basically a phase delay, more than a distortion, since there are no other clear frequencies present besides the excitation frequency. Only some higher harmonics are visible.
F.4 **Verification amplifier**

\[
\frac{V_{out}}{V_{in}} = -\frac{10k}{2k} = -5 = 14\text{dB}
\]

Figure 79: Electrical scheme voltage amplifier
Up to 200 kHz the amplifier still has a flat response of almost $13.8\pm0.05$ dB, which corresponds to almost a factor 5 as shown in Figure 80. The phase delay at 200 kHz equals 12.7 °. Note that for clarity of the plot, the phase is in this figure is not inverted.

In case the two piezos are connected to the amplifier the response will be according to Figure 81. The resonance frequency of the piezo actuator is clearly visible at 550 kHz due to the induced voltage. At 200 kHz the phase delay is 11° and the magnitude increase 0.7 dB due to the upcoming resonance peak.

**Figure 80: Amplification amplifier measured at 112 mV**

**Figure 81: Amplification of amplifier loaded with two piezo actuators**

### F.5 Verification decoder

Figure 82 and Figure 83 showing the same measurement processed by respectively a displacement decoder and a velocity decoder. Important differences are the much higher noise level of the displacement decoder. The explanation for this is that a
displacement decoder counts the number of phase crossings of the incoming interfered laser signal to determine the travelled distance. This method is very susceptible to drop-outs of the signal. The velocity decoder retrieves the frequency of the interfered signal, such that a signal with drop-outs can still be processed.

Another difference can be seen in the phase delay. Although the displacement decoder has a bandwidth of 350 kHz the phase delay is almost 4 degree/kHz, making this decoder unsuitable for phase measurements at these frequencies. The magnitude response is despite the noise still a valid measurement.

F.6 Robustness TTS

During testing of the performance of the TTS, it was a number of times assembled and disassembled. This was causing the dynamical response to change slightly. Other causes affecting the response were the degradation of the piezo actuators and the positioning of the sensor. The system response as measured directly after the time measurements of 7.3 is shown in Figure 84. There are two important changes. First there is an additional resonance peak appearing at 17 kHz. It is possible this peak corresponds to the upper parts of the capillary and is shifted was not clearly visible in previous responses since it was canceled by the counter mass mode. However this hypothesis is not confirmed yet. Secondary it is seen the TTS is decoupling at a lower frequency. This results in an amplitude of only 2nm/V
Figure 84: Dynamical response in [nm/V] of TTS measured after testing
Appendix G. Technical documentation

G.1 Amplifier

### STABILITY FOR CAPACITIVE LOADS

<table>
<thead>
<tr>
<th>Model</th>
<th>MP108.1000</th>
<th>Nota/More</th>
<th>Pin</th>
<th>12 kOhms</th>
<th>Estimated Closure Frequency</th>
<th>649.262 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rcl</td>
<td>0</td>
<td>Ohms</td>
<td>Cn</td>
<td>5 nF</td>
<td>Estimated maximum bandwidth</td>
<td>74,9894 kHz</td>
</tr>
<tr>
<td>Cload</td>
<td>0.05 uF</td>
<td></td>
<td>Cr</td>
<td>0 pF</td>
<td>Estimated Closure Rate</td>
<td>22.8 dB/decade</td>
</tr>
<tr>
<td>Rm</td>
<td>2 kOhms</td>
<td></td>
<td>Rr</td>
<td>0 Ohms</td>
<td>Estimated Phase Margin</td>
<td>23.52 Degrees</td>
</tr>
<tr>
<td>Rf</td>
<td>10 kOhms</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**R-C Pole Calculator:**

- 100 kOhms
- 3.13 nF
- 1000 kHz
- 100 nF
- 0.05 uF

**Total Rout:** 5 Ohms

**Pole Zout/Cload:** 636.6184 kHz

**I/Beta (UC):** 10.6 dB

**Noise Gain:** 7.6 dB

**Pole Noise Gain:** 26.5265 kHz

**Zero Noise Gain:** 11.103 kHz

**Pole C/Drift:** 159,154 kHz

**Zero P/IC:** 2.8212 kHz

**Zero R/IC:** 3.1839 kHz

---

**Bode Plot**

**Phase Shift**

---

Page Down for Plots.
### G.2 Preload spring

<table>
<thead>
<tr>
<th>Material</th>
<th>Stainless steel DIN17224-1.4310</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ends</td>
<td>squared and ground</td>
</tr>
<tr>
<td>Diameter (center to center) [mm]</td>
<td>$D_m$</td>
</tr>
<tr>
<td>Wire diameter [mm]</td>
<td>$D$</td>
</tr>
<tr>
<td>Number of windings [-]</td>
<td>$N_w$</td>
</tr>
<tr>
<td>Zero length [mm]</td>
<td>$L_0$</td>
</tr>
<tr>
<td>Stiffness [N/mm]</td>
<td>$C$</td>
</tr>
<tr>
<td>Maximum stroke [mm]</td>
<td>$S_n$</td>
</tr>
<tr>
<td>Force at maximum stroke [N]</td>
<td>$F_n$</td>
</tr>
</tbody>
</table>
**G.3 Capillary**

Capillary used in Phicom: 1572N-13S-437GM
Capillary used in test setup: 1572N-13S-625GM

---

**CERAMIC CAPILLARIES**

**1/16 inch Diameter**

- Ø0.020 / .76mm ±.005 / .06mm
- Ø0.024 / 1.56mm ±.005 / .06mm
- Ø0.028 / 2.2mm ±.005 / .06mm
- 0.375 / 9.6mm
- 0.437 / 11.7mm
- 0.475 / 12.1mm

**1/8 inch Diameter**

- Ø0.030 / .76mm ±.005 / .06mm
- Ø0.040 / 1.28mm ±.005 / .06mm
- 0.375 / 9.6mm
- 0.437 / 11.7mm

**TUNGSTEN CARBIDE CAPILLARIES**

**1/16 inch Diameter**

- Ø0.034 / 1.07mm ±.005 / .06mm

**1/8 inch Diameter**

- Ø0.043 / 1.1mm ±.005 / .06mm

---

**Non-standard lengths and tighter tolerances available at additional cost. Non-standards lengths may alter the back hole size. Some series are standard with tighter tolerances. Dimensions in inches unless otherwise specified.**
The 1574 series features a flat face design and a 90° double inside chamfer for surfaces with good bondability. The 90° double inside chamfer provides excellent 2nd bondalling and a taller, more compact ball bond.

The 1574N may be specified for a 120° inside chamfer for surfaces with poor 1st bond bondability.

For 120° IC Angle, Specify 1574N Series

Specify: Series - Dash Number - Length+Finish - Options
Example: 1574-18.457G-3M-3D

Note: For Tungsten Carbide material, specify 1174 & 1174N series (1/16 in. diameter only).
For 1/8 in. diameter ceramic, specify 1574 & 1574N series.

<table>
<thead>
<tr>
<th>SERIES &amp; DASH NO.</th>
<th>H* in/µm ±0.001/2.5</th>
<th>IC in/µm (ref)</th>
<th>B** in/µm ±0.002/5</th>
<th>OR*** in/µm ±0.003/8</th>
<th>T (30° CONE) in/µm ±0.003/8</th>
<th>T (20° CONE) in/µm ±0.003/8</th>
<th>SUGGESTED WIRE DIAMETER in/µm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1574-10</td>
<td>0.001/25</td>
<td>0.002/5</td>
<td>0.004/26</td>
<td>0.005/64</td>
<td>0.006/1.65</td>
<td>0.006/1.74</td>
<td>0.005/12M to 0.009/20</td>
</tr>
<tr>
<td>1574-12</td>
<td>0.001/30</td>
<td>0.002/5</td>
<td>0.004/61</td>
<td>0.005/64</td>
<td>0.006/1.65</td>
<td>0.006/1.74</td>
<td>0.007/18 to 0.009/23</td>
</tr>
<tr>
<td>1574-13</td>
<td>0.001/33</td>
<td>0.003/8</td>
<td>0.009/48</td>
<td>0.005/64</td>
<td>0.006/1.65</td>
<td>0.006/1.74</td>
<td>0.008/20M to 0.010/25</td>
</tr>
<tr>
<td>1574-15M</td>
<td>0.001/38</td>
<td>0.003/8</td>
<td>0.021/53</td>
<td>0.005/69</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.009/22M to 0.011/28</td>
</tr>
<tr>
<td>1574-17</td>
<td>0.001/43</td>
<td>0.002/5</td>
<td>0.021/53</td>
<td>0.005/69</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.010/25M to 0.013/33</td>
</tr>
<tr>
<td>1574-17S</td>
<td>0.001/46</td>
<td>0.008/20</td>
<td>0.034/86</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.008/215</td>
</tr>
<tr>
<td>1174-18</td>
<td>0.001/46</td>
<td>0.005/14</td>
<td>0.029/74</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.009/215</td>
</tr>
<tr>
<td>1574-18S</td>
<td>0.001/46</td>
<td>0.002/5</td>
<td>0.022/59</td>
<td>0.005/69</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.008/215</td>
</tr>
<tr>
<td>1574-20</td>
<td>0.002/53</td>
<td>0.007/18</td>
<td>0.034/86</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.009/215</td>
</tr>
<tr>
<td>1574-20N</td>
<td>0.002/53</td>
<td>0.007/18</td>
<td>0.040/11</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.008/215</td>
</tr>
<tr>
<td>1574-22</td>
<td>0.002/56</td>
<td>0.006/15</td>
<td>0.029/74</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.009/215</td>
</tr>
<tr>
<td>1574-22M</td>
<td>0.002/56</td>
<td>0.005/9</td>
<td>0.029/74</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.010/25M to 0.013/38</td>
</tr>
<tr>
<td>1574-25</td>
<td>0.002/54</td>
<td>0.005/13</td>
<td>0.035/89</td>
<td>0.004/61</td>
<td>0.008/203</td>
<td>0.008/215</td>
<td>0.008/215</td>
</tr>
<tr>
<td>1574-30</td>
<td>0.003/76</td>
<td>0.016/25</td>
<td>0.056/127</td>
<td>0.005/140</td>
<td>0.016/419</td>
<td>0.017/437</td>
<td>0.020/51M</td>
</tr>
<tr>
<td>1574-35</td>
<td>0.005/99</td>
<td>0.008/20</td>
<td>0.057/130</td>
<td>0.005/140</td>
<td>0.016/419</td>
<td>0.017/437</td>
<td>0.017/437</td>
</tr>
<tr>
<td>1574-35S</td>
<td>0.003/89</td>
<td>0.008/20</td>
<td>0.057/130</td>
<td>0.005/140</td>
<td>0.016/419</td>
<td>0.017/437</td>
<td>0.017/437</td>
</tr>
<tr>
<td>1574-40</td>
<td>0.004/102</td>
<td>0.016/25</td>
<td>0.066/162</td>
<td>0.005/145</td>
<td>0.016/419</td>
<td>0.017/442</td>
<td>0.017/442</td>
</tr>
<tr>
<td>1574-50</td>
<td>0.005/127</td>
<td>0.017/30</td>
<td>0.074/188</td>
<td>0.007/178</td>
<td>0.019/463</td>
<td>0.020/508</td>
<td>0.020/508</td>
</tr>
</tbody>
</table>

* For hole sizes less than 0.00011 and hole sizes 0.0025 through 0.0009, the tolerance is ±0.002/.0001.
** For B dimensions greater than 0.0304, the tolerance is ±0.003/.0001.
*** OR tolerance ±0.005 for OR less than or equal to ±0.003, for OR greater than ±0.003, tolerance is ±1.9%.
Tighter tolerances available at additional charges. Dimensions in inches unless otherwise noted.
G.4  **Network Analyser**

HP 4395A Network/Spectrum/Impedance Analyzer  

- Frequency range: 10 Hz to 500 MHz  
- Resolution: 1mHz  
- 115 dB dynamic range @ 10 Hz IFBW  
- +/-0.05 dB, +/-0.3 degree dynamic accuracy

G.5  **Scopes**

G.5.1  **Tiepie Handyscope 3**

<table>
<thead>
<tr>
<th>Resolution</th>
<th>Signal/noise ratio</th>
<th>levels</th>
<th>Maximum sample frequency</th>
<th>percentage proportion</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>48 dB</td>
<td>256</td>
<td>100 MHz</td>
<td>0.39%</td>
</tr>
<tr>
<td>12</td>
<td>72 dB</td>
<td>4096</td>
<td>50 MHz</td>
<td>0.024%</td>
</tr>
<tr>
<td>14</td>
<td>84 dB</td>
<td>16384</td>
<td>3.125 MHz</td>
<td>0.0061%</td>
</tr>
<tr>
<td>16</td>
<td>96 dB</td>
<td>65535</td>
<td>195 kHz</td>
<td>0.0015%</td>
</tr>
</tbody>
</table>

| Number of inputs | 2                  |
| Input voltage range | 0.2-80V           |
| Number of output   | 1                  |
| Maximum voltage output | 0±12V             |
| Memory             | 128 kSamples       |
| Arbitrary waveform generator | Yes, 14 bit resolution |

| USB connection | Yes |
| Trigger function | Yes |

G.5.2  **Tiepie Handyscope 4**

<table>
<thead>
<tr>
<th>Resolution</th>
<th>Signal/noise ratio</th>
<th>levels</th>
<th>Maximum sample frequency</th>
<th>percentage proportion</th>
</tr>
</thead>
<tbody>
<tr>
<td>12</td>
<td>72 dB</td>
<td>4096</td>
<td>50 MHz</td>
<td>0.024%</td>
</tr>
<tr>
<td>14</td>
<td>84 dB</td>
<td>16384</td>
<td>3.125 MHz</td>
<td>0.0061%</td>
</tr>
<tr>
<td>16</td>
<td>96 dB</td>
<td>65535</td>
<td>195.3125 kHz</td>
<td>0.0015%</td>
</tr>
</tbody>
</table>

| Number of inputs | 4                  |
| Maximum voltage | 0.2-80V           |
| Memory           | 128 kSamples       |
| USB connection   | Yes               |
| Trigger function | Yes               |
## G.6 Laser vibrometer

Polytec OFV-552 Controller

### G.6.1 Velocity decoder VD-02

#### 7.5.2 Velocity Decoder VD-02

<table>
<thead>
<tr>
<th>Measurement range</th>
<th>5</th>
<th>25</th>
<th>125</th>
<th>1000</th>
<th>( m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full scale (peak)</td>
<td>0.05</td>
<td>0.25</td>
<td>1.25</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Frequency range</td>
<td>( f_{nv} )</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td>( f_{nv} )</td>
<td>250</td>
<td>1500</td>
<td>1500</td>
<td>1500</td>
</tr>
<tr>
<td>Max. acceleration</td>
<td>8000</td>
<td>24000</td>
<td>120000</td>
<td>960000</td>
<td>g</td>
</tr>
<tr>
<td>Frequency response(^1)</td>
<td>0.5 Hz ... 20 Hz</td>
<td>±0.5</td>
<td>±0.5</td>
<td>±0.5</td>
<td>±0.5</td>
</tr>
<tr>
<td></td>
<td>20 Hz ... 100 kHz</td>
<td>±0.1</td>
<td>±0.1</td>
<td>±0.1</td>
<td>±0.1</td>
</tr>
<tr>
<td></td>
<td>100 kHz ... 250 kHz</td>
<td>+0.2/-1</td>
<td>±0.1</td>
<td>±0.1</td>
<td>±0.1</td>
</tr>
<tr>
<td></td>
<td>250 kHz ... 1.5 MHz</td>
<td>-</td>
<td>±0.5/-2</td>
<td>±0.5/-2</td>
<td>±0.5/-2</td>
</tr>
<tr>
<td>Resolution(^2)</td>
<td>frequency-dependent(^3)</td>
<td>0.05...0.2</td>
<td>0.1...1</td>
<td>0.3...3</td>
<td>2...5</td>
</tr>
<tr>
<td></td>
<td>typically(^4)</td>
<td>0.1</td>
<td>0.5</td>
<td>0.6</td>
<td>2.5</td>
</tr>
<tr>
<td>Frequency-dependent phase shift ( \rho_{d} ) (typ.)</td>
<td>-2.19</td>
<td>-0.45</td>
<td>-0.42</td>
<td>-0.36</td>
<td>°/kHz</td>
</tr>
<tr>
<td>Signal delay ( t_{d} ) (typ.)</td>
<td>6.08</td>
<td>1.25</td>
<td>1.17</td>
<td>1.01</td>
<td>μs</td>
</tr>
<tr>
<td>Calibration error(^6)</td>
<td>( T_{a} = (25 ± 3)°C )</td>
<td>±1</td>
<td>±1</td>
<td>±1</td>
<td>±1</td>
</tr>
<tr>
<td></td>
<td>( T_{a} = (77 ± 5)°F )</td>
<td>±1</td>
<td>±1</td>
<td>±1</td>
<td>±1</td>
</tr>
<tr>
<td></td>
<td>( T_{a} = +5°...+40°C )</td>
<td>±1.5</td>
<td>±2</td>
<td>±2.5</td>
<td>±2.5</td>
</tr>
<tr>
<td></td>
<td>( T_{a} = +41°F...+104°F )</td>
<td>±1.5</td>
<td>±2</td>
<td>±2.5</td>
<td>±2.5</td>
</tr>
<tr>
<td>Linearity error(^6)</td>
<td>1</td>
<td>1.5</td>
<td>1</td>
<td>1</td>
<td>%</td>
</tr>
<tr>
<td>Harmonic distortions</td>
<td>( &lt; -52 )</td>
<td>( &lt; -46 )</td>
<td>( &lt; -50 )</td>
<td>( &lt; -50 )</td>
<td>dB</td>
</tr>
<tr>
<td>Spurious signals (non-harmonic)(^7)</td>
<td>( &lt; -86 )</td>
<td>( &lt; -86 )</td>
<td>( &lt; -86 )</td>
<td>( &lt; -86 )</td>
<td>dBFS</td>
</tr>
</tbody>
</table>

\(^1\) The frequency response defines the frequency-dependent amplitude error, referred to the reference frequency of 1 kHz.

\(^2\) The noise-limited resolution is defined as the signal amplitude (rms) at which the signal-to-noise ratio is 6 dB with 1 Hz, spectral resolution, measured on 3M Scotchite Tape® (reflective film).

\(^3\) The attainable resolution is frequency-dependent and is specified for frequencies above 10 kHz.

\(^4\) The typical value refers to the center of the operating frequency range.

\(^5\) Conditions: sinusoidal vibration, \( f = 1 \text{kHz} \), amplitude 70% of full scale range, load resistance \( \geq 1 \text{N} \).

\(^6\) The linearity error is defined as the amplitude-dependent, relative deviation of the scaling factor, referred to the scaling factor under calibration conditions (refer to footnote\(^5\)).

\(^7\) The maximum amplitude of the distortion refers to the full scale. An exception of which is a single peak, generated by the optical sensor, in the frequency range 20...25 kHz, whose amplitude depends on the stand-off distance.
G.6.2 Displacement decoder

<table>
<thead>
<tr>
<th>Measurement range (μm/V)</th>
<th>Full scale (peak-to-peak) (μm)</th>
<th>Resolution (rounded) (nm)</th>
<th>Frequency range (kHz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>1</td>
<td>0.015</td>
<td>0...350</td>
</tr>
<tr>
<td>0.1</td>
<td>2</td>
<td>0.03</td>
<td>0...350</td>
</tr>
<tr>
<td>0.2</td>
<td>4</td>
<td>0.06</td>
<td>0...350</td>
</tr>
<tr>
<td>0.5</td>
<td>10</td>
<td>0.15</td>
<td>0...350</td>
</tr>
<tr>
<td>1</td>
<td>20</td>
<td>0.3</td>
<td>0...350</td>
</tr>
<tr>
<td>2</td>
<td>40</td>
<td>0.5</td>
<td>0...350</td>
</tr>
<tr>
<td>5</td>
<td>100</td>
<td>1.5</td>
<td>0...350</td>
</tr>
<tr>
<td>10</td>
<td>200</td>
<td>3</td>
<td>0...350</td>
</tr>
<tr>
<td>20</td>
<td>400</td>
<td>6</td>
<td>0...350</td>
</tr>
<tr>
<td>50</td>
<td>1000</td>
<td>15</td>
<td>0...350</td>
</tr>
<tr>
<td>100</td>
<td>2000</td>
<td>30</td>
<td>0...350</td>
</tr>
<tr>
<td>200</td>
<td>4000</td>
<td>60</td>
<td>0...350</td>
</tr>
<tr>
<td>500</td>
<td>10000</td>
<td>150</td>
<td>0...350</td>
</tr>
<tr>
<td>1000</td>
<td>20000</td>
<td>300</td>
<td>0...350</td>
</tr>
<tr>
<td>2000</td>
<td>40000</td>
<td>600</td>
<td>0...350</td>
</tr>
<tr>
<td>5000</td>
<td>100000</td>
<td>1500</td>
<td>0...350</td>
</tr>
</tbody>
</table>

1 The full scale values correspond to ±1 V (peak-to-peak) maximum output voltage.
2 The resolution corresponds to the quantization step of approx. 0.4 mV at the analog output.
3 When a suitable measurement range has been selected for the digital velocity decoder.

Amplitude Frequency Response

<table>
<thead>
<tr>
<th>Frequency response</th>
<th>Max. additional error with reference to f = 1kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 kHz...100 kHz</td>
<td>±0.08 dB</td>
</tr>
<tr>
<td>100 kHz...200 kHz</td>
<td>±0.1 dB</td>
</tr>
<tr>
<td>200 kHz...350 kHz</td>
<td>±0.2/-1 dB</td>
</tr>
</tbody>
</table>

Phase shift: typ. -3.96°/kHz
Delay: typ. 11μs
Noise-limited resolution: < 0.5 pm/√Hz
Calibration error: ±1% of the measurement value (Ta = 25°C...+40°C) conditions: sinusoidal vibration, f = 1 kHz, amplitude 70% of full scale, load resistance ≥ 1 MΩ

G.7 Current sensor

Tektronix A6302 with AM503B

- 2 Meter Cable
- 20 Amps Continuous / 50 Amps Peak
- Up to 0.15 in. Conductor Diameter
G.8  **Thermosistor**

Sensordata NTC THERMISTOR: TYPE SC30


- Range: –40°C to +105°C
- Resistance: 10 kOhm
- Thermal time constant: 5 sec.
- Accuracy: ±.1°C
Appendix H. Drawings

Following drawings are included:

- Framework
- Capillary holder
- Preload dowel
- Assembly tool
H.1 Framework
H.2  *Capillary holder*
H.3  

*Preload dowel*
H.4  Assembly tool
Appendix I. Photos

Figure 85: TTS mounted to test block

Figure 86: Front view TTS with symmetrically positioned counter masses, piezo actuators and capillary locked up by preload springs
Figure 87: TTS mounted in Phicom