High frequency control of a 1-DOF haptics system extended

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1 Preface

Since this is my bachelor final project my personal main goals are to learn a lot on the coarse of control engineering. The subject of the master-slave technique and the force-feedback attracted me the moment I read about it. Also the field of application (medical world combined with the technical) had my attention since I started my education. That is why I am looking forward to the upcoming 10 weeks to work on this project.
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2 Introduction

In August 2007 ing. Scholtes finished his traineeship report[3] focusing on the low frequent behavior and control of a 1-DOF haptic system. This master-slave setup (study object in this and of previous research[3]), designed by Ir.R.Hendrix (figure 1), together with a digital amplifier, specially made for this haptics setup, forms a study object for future multi-degree haptics systems. This survey is part of a study that started in 2005, attempting to design a next generation, haptic feedback enhanced, robot system for Minimally Invasive Surgery (MIS).

This report basically forms an extension of jesse Scholtes’ report[3], focusing on the high frequent behavior and control instead of the low frequent features, causing some similarities and references.

![Figure 1: 1-DOF setup](image)

2.1 Outline

As already mentioned in the introduction the intention of this report is to be a stepping-stone for future surveys in the field of haptics. During normal motion of the human hand a maximal frequency of about 10 Hz can be created in the medical equipment. When however for instance a surgeon hits a hard-contact surface (like bones), higher frequencies can occur (because of vibrations). In the next 3 chapters an outline is given of how to control and visualize this high frequency behavior:

**Chapter 1: Introduction**

The first chapter gives a brief introduction about the project and the goals
Chapter 2: System identification
The second chapter forms the main text of the report. First some measuring methods are discussed, for obtaining a FRF (frequency response function) and afterwards the FRF itself is measured and the controller is designed. Also the delay of the system and the error are identified. Finally a calibration for the force against voltage is done.

Chapter 3: Conclusions, recommendations and expectation
This chapter contains the conclusion and recommendations for the future, with optional subjects for future research in it.

2.2 Goals
Since previous research by [3] dealt with the low frequent element of the setup, the behaviour is totally known till 200 Hz. Some conclusions that may be drawn from his report are:

- both master and slave give (almost) the same response.
- the error signal stays within good boundaries +/-7e-3[rad]
- a digital amplifier gives the best experimental results
- an open loop measurement using a chirp signal and a noise signal gives the most reliable measurement data

The goal of this report is to discover the response of the setup if it reaches higher frequencies. Which behaviour is the same and what are the differences? Can the same controller be used? This leads to the following goals for this, ten weeks during, bachelor final project:

- motor-encoder response (FRF) in SI-units till 1000 Hz
- create a controller with a as high as possible crossover frequency
- determine the delay in the system
- verification of the voltage-force transmission
2.3 Test setup

The 1-dof haptics system consists of a master and a slave part, a digital amplifier, TU/e Dacs AQI data acquisition and a PC with Linux OS, Matlab with realtime workshop included. Important here is that master and slave side are the same and as is concluded(section 2.2) by previous research [3] react the same. The composition of the setup is given in figure (2). For further details of the composition of the arrangement is referred to previous research [3].

Figure 2: Test setup

- 1. Actuator; Maxon RE 35 [6];
- 2. Encoder; 30,000 increments/revolution, the encoders resolution is 2.094e-4 rad.
- 3. Force sensors; Kaman inductive position sensors) [5]; stiffness = 153-415 N/cm (rigid wall) ; accuracy = 10 - 15mN.
- 4. Pulleys and dyneema wire;
- 5. Segment part a and b;
- 6. Aluminium leafsprings;
- 7. End switches;
3 Analysis of the system

When talking about dynamical systems, it mostly concerns a simple system containing a mass, spring and a damper (figure 3). Theoretically by calculations using the body mass, stiffness of the spring and the damping constant the output of the system can be determined if the input is known. If however, as always in the real world, some frictions or disturbances exist (time-, temperature, frequency- and place dependent) the result cannot be predicted with modeling and has to be identified using FRF.

These real-time analysis are done using the equipment described earlier (section 2.3). In this chapter, by execution motor position experiments, the setup is analyzed. By making a frequency response function of the stable system the dynamics are studied. Afterwards a controller can be added, to be able to handle the system with regard to specified criteria regarding performances and stability. At last methods to upgrade the performance of the control are applied, like feedforward control to reduce the error. A model can be made to determine certain parameters of the system (like spring stifnesses).

![Figure 3: standard dynamical system](image)

3.1 Measuring methods

One of the goals of this report is to identify the high frequent behaviour of the system. The regular way of doing this is by making use of a frequency response function (FRF). A frequency response function gives the response of the system at different frequencies, and puts it in a Bode plot (amplitude against frequency and phase against frequency).

As study [3] also mentioned the important factor is to make a reliable measurement with which the analysis can be made. It concluded that a chirp signal was the best measuring method for the low frequent analysis. Because this is another approach all methods are given a second opinion.

- **Band-Limited white noise**

  A band-limited white noise block generates normally distributed random numbers that are suitable for use in continuous systems. Band-limited white noise produces an output at a specific sample rate, which is related to the correlation time of the noise. In practice, physical systems are never disturbed by white noise, although white noise is a useful theoretical approximation
when the noise disturbance has a correlation time that is very small relative to the natural band-
width of the system.

- **frequency sweep (chirp)**
  A chirp signal is a sign function with a certain amplitude and range in frequency. In a given
amount of time the frequency rises from its lower boundary to the maximum. survey [3] con-
cluded that for the low frequent part this was the best solution.

- **A constant velocity with noise**
  This method almost speaks for itself. A constant velocity is generated and a noise signal is
added. The noise signal can for instance be a random number block or a band-limited white
noise block. advantage of the Band-Limited white noise is that it produces output at a specific
sample rate, which is related to the correlation time of the noise. Additional to that the ampli-
tude is adaptive, so the "amount" of noise can be regulated.

For the low frequent analysis the “frequency sweep (chirp)” was the best solution. Since this is an-
other approach the different methods have to be weighed out. With one simple function the “frequency
sweep (chirp)” method is able to excitate the system over the full frequency range, but there are some
disadvantages: first of all, when the setup is excitated over it’s full frequency range, the table will res-
onate and if not it should be almost impossibly fixed.
Secondly as survey [3] stated, with a simulation time of 1000 seconds a frequency response for every
(150/1000) 0.15Hz is made ([3] chapter 2.1). This is necessary to get a good view on the low frequent
dynamics. If however instead of 150 Hz the range is set to a maximum of 2000Hz the measuring
time has to be increased drastically to get a decent FRF. Finally, since the system is excitated slowly
at low frequencies, the friction is dominant here. This also is a negative site effect. Advantage of the
“constant velocity with noise” method is that it cancels out stick-slip and static- or coulomb friction.
According to earlier research by [3] a constant velocity can’t be achieved because of the stroke of 35○,
however by setting a low velocity, a constant velocity can be approached. Since the amplitude is also
easy adaptive, the setup is easier to stabilize on a fixed position and isolate it from outside disturbances.

It may be clear that for the high frequent part of the system, because of the canceling out of the friction
terms, the adaptive amplitude and sample rate depended output of the white noise, a constant velocity
with Band-Limited white noise is chosen.

### 3.2 Frequency response function

The way how and in what steps a frequency response function has to be made is discussed later, but
first of all the following is observed:
3.2.1 Open Vs. Closed Loop

The best method for sending in the different frequencies is already examined, but there are also different ways of modeling. That is closed loop (C-1) and open loop (D-1). In the figures (4) and (5) the same setup is analyzed by the two different methods. Visible is that the closed loop analysis (for this controller) damps out the high frequent disturbances, because of the feedback loop, more than the Open Loop analysis. The result is a more smooth curve in the high frequent region of the figure. In the rest of the tests/report the closed loop analysis will be used because of this reason, if not it will be explicitly mentioned.

![Figure 4: FRF Master Open Loop](image)

3.2.2 Method of making a FRF

When making a FRF the purpose is to get an idea of the behaviour of the response of the plant $H(C-1)$. When dealing with closed loop systems the information of the plant can’t been read out of the exits of the system (since the exits contain more then just information of the plant). The common way (6) of making a FRF is now by defining the process sensitivity (output divided by the noise) and the sensitivity (input divided by the noise). By now dividing the process sensitivity through the sensitivity, the plant $H$ is obtained.

Now the basics of making a FRF are clear, the chosen method (section 2.1) can be applied to measure a FRF. Since for former research [3] a basic controller (for stability and robustness) are already made, these are can be adapted here. Since this survey [3] was located on the low frequent behaviour and tried to make it as stable and robust as possible a second order low pass filter was added to damp frequencies above 150 Hz. Because this report especially focusses on these high frequencies this filter is left out and the analysis is started. In the search of the high frequent dynamics of the system the
adapted controller to create a stable system consist of the following:

**Integrator:** An integrating action is used to compensate low frequent static friction.
- zero = 0.5 [Hz]

**Lead controller:** A phase lead is necessary at 30Hz, thus a lead action is added.
- zero: 10 [Hz]
- pole: 90 [Hz]

The use of this controller leads to the bode plot of the robust controller (Figure 7). A lot of the high frequent dynamics is already quite good visible. But as the amplitude of the system is set smaller, the
crossover frequency decreases. As this happens the systems (loosens) damps more and more detailes high frequent dynamics get visible in the FRF. As gets visible in the pictures (8) and (9) this is done two times. Ones for a amplitude of 0.2 times the robust controller’s gain and the second time 0.05 times the robust controller’s gain.

Figure 7: plant with robust controller
Figure 8: plant with 0.2*robust controller
What may be concluded from the pictures is that there are not a lot of differences between the three controllers applied, the eigenfrequencies become better visible around 200 Hz and 800 Hz, also around 350 Hz the system has an eigenfrequency. The eigenfrequency around 200 Hz could be explained by a difference in the stiffness of the springs (decoupling of motorpulley from segments A and B, contact stiffness) attached to the segments. Around 350 Hz another eigenfrequency acts, caused by the total decoupling of both segments from the motorpulley. This decoupling depends on the stiffness of the connection and by making a dynamical model of the system and fitting the eigenfrequencies, systemparameters (like the stiffness of the spring leaves) can be calculated. This is however not done yet and is recommended for future research. The third eigenfrequency around 800 Hz is caused by the decoupling of both segments from each other (aluminium leavesprings in section 2.3).

To check whether the results of the previously mentioned systems are reliable the coherence has to be plotted. The coherence is a function that shows in what level there is a linear relation between the input signal and the output signal. In literature [7] the limit of 0.8 and above is taken for a reliable signal. For merged signals like this signal $H=PS/S$ an indication of the form of the coherence can be given by multiplying $PS$ and $S$, but it can also be analyzed by the two signals separately.(appendix I)

The advantage of looking at the sensitivity and the processesensitivity’s coherence separate is that if rather good values for the separate coherence show, the product of both would also have given a good
result and thus the result is reliable. Secondly it is possible to analyze the two curves. Since the sensitivity equals \( \frac{1}{1+CH} \) and for very big frequencies the sensitivity will grow to 1. On the other hand, the processensitivity equals \( \frac{H}{1+CH} \). So for very big frequencies this function will approach H. As can be seen in the three figures of the coherence this is right because the eigenvalues from the function H can be spotted in the coherence of the processensitivity. The higher the crossover frequency of a system the better the eigenfrequencies are visible because the controller takes a bigger part in the fraction \( \frac{1}{1+CH} \).

Since there is not a lot of difference between the three controllers and the robust controller at first is the most stable, this will be the one to be used for further analysis.

### 3.3 Delay

In every system a certain delay acts, at least as big as the sample frequency of the system. This setup is sampled every \( \frac{1}{4000} \) [s], so the delay is always \( \frac{1}{4000} = 0.25 \) [ms] or more. The calculation of the delay will come clear on basis of picture (10). As explained in the figure a full rotation takes \( 360^\circ \). I will soon start to super-conduct. In a system with only the sample frequency as a delay this would match with 4 [KHz]. Divided by four would \( 90^\circ \) match with 1 [KHz]. In this setup however is at \( 90^\circ \) the frequency 400 [Hz] and thus at a full pitch this means 1600 [Hz]. The delay is then at \( 360^\circ \), 0.625 [ms] and 2.5 times as big as minimum possible. Reasons for delays in a dynamical system are caused by the data acquisition, sampling of the encoder, turning from analog to digital signals.
3.4 Controller

The total dynamics of the system are identified, the next step is to control the setup also in the high frequent area. The goal of the first FRF was to get an identification of the system. If for instance however the system gets operational uncontrolled, without damping the eigenfrequencies, during a high precision operation like eye-surgery, the system will not be stable. Concluding that a controller has to be applied to stabilize and make the system robust.

3.4.1 Feedback

How to get a system stable will be explained on the base of the next factors. If these margins are satisfied the cross-over frequency of the system will be stable and the total system well operative in the low frequent area. In the high frequent area it will be controllable till about the crossover frequency. In the nyquist figure 11 these margins are visualized. Additional to this it has to be pointed out that the further the line $L(j\omega)$ is removed from the point -1, so the bigger MM is, the more robust the system is.

- Gain Margin (GM);
The factor the gain can be increased with, before instability occurs. In literature [7] the GM is said to have a value of 6 dB or higher for stability.

- **Phase Margin (PM);**
  The angle the gain can be increased with, before instability occurs. The PM needs to have a phaselead of 30° or higher for stability.

- **Modulus Margin (MM);**
  The radius of the smallest circle with midpoint -1, touching the polar figure. Regularly the MM is said to have a value of 6 dB or lower for stability.

- **Crossover Frequency:**
  Frequency at the point the magnitude part of the Bode figure passes the 0 dB line, in this case try to set as high as possible.

In figure (12) the plant is ones more given. This time the figure is totally analyzed:

- **Till 10 Hz:**
  Till about 10 Hz the slope of the figure is -1 as can be seen in the picture. This is caused by static friction and can be written in the frequency domain as \( \frac{1}{m}\).

- **10-200 Hz:**
  Further on in the figure the slope changes to -2, which in the frequency domain corresponds to \( \frac{1}{m\omega^2} \). This -2 slope is caused by Newton’s second law \( F=m\cdot a \).
Figure 12: plant and controller*plant

- 200-2000 Hz;

The different -2 slopes correspond to the different masses the system has because of the decoupling of the system at the three eigenfrequencies. By further investigation it is possible to link the distance between the lines drawn in the figure to the difference between the uncoupled masses.

When looking at the right part of figure (12), it can be seen that first of all it is a lot smoother, because the first two eigenfrequencies are filtered out. Secondly the -1 slope at the beginning of the figure is for a big part gone, as is the unwanted static friction. Around 800 Hz the eigenfrequency is still present, but vanishes in the high frequent noise, because the crossover frequency of the controller is at 102.09 Hz. The characteristics of the controller:

- **Gain:**
  - zero: 0.0006

- **PD controller:** A PD controller decreases the phase-lag.
  - F: 628.319[Hz]
  - D: 62.8319[Hz]

- **Second order low pass filter:** A second order low pass filter is added to damp frequencies above a certain frequency.
  - pole: 500[Hz]
  - damping: 0.6

- **Notch filter:** A notch filter, filters away eigenfrequencies/resonance frequencies.
  - zero: 345[Hz]
  - damping: 0.05
• zero: 260 [Hz]
• damping: 0.1

Stability margins:
• Cross over frequency = 102.09 Hz
• Phase margin = 40°
• Gain margin = 8.5 dB
• Modulus margin = 5.7 dB

3.4.2 Feedforward

Feedforward actually means predicting the future of different derivatives of the position signal B-1. This can be done in many degrees, but it is mostly done till the second degree (acceleration). When a trajectory is given to the setup when only applying feedback control nothing is given in higher degree derivatives. When however as can be seen in figure (14) the higher degree derivatives are multiplied with a constant use will also been made of higher degree information. Determining the constants is done by first just taking the feedback control. Then the error is plotted against time and different
values are just filled out by trial and error. An example is given in (appendix 6), where the viscous friction is tuned during 1000 [s] and the best value seems to be 0. lowest friction for the coulomb friction and the inertia are 0.01 [-] and 0 [-] respectively.

![Figure 14: model feedforward](image)

### 3.5 Error

A clear way to identify whether a system is stable and robust is to take a look at the error signal (error is taken just before the controller as can be seen in the closed loop model in C-1). Obviously the error of the setup is wanted to be as close to zero as possible. When the system moves very slowly the error clearly decreases compared to an extreme acceleration. This means that the error depends on the trajectory (appendix 2) that is given in. When the trajectory is not altered, the error can be compared for different control methods to check which controller gives the lowest error. For instance the difference between feedforward- and feedback-control. The error of these two methods is given in figure (15).

### 3.6 Verification

In this section The relation between the force exerted on the setup and the voltage given out by the amplifier is investigated. Theoretically 8 [ampère/volt] is given out by the amplifier. The DC-motor gives 38.9 [mNm/ampère] with a ratio of 1:10 measured on a arm of 75 [mm][3]. This all leads to:

\[
\frac{8 \times 10^{-3} \times 10^{-1} \text{ampere} \times N \times m}{\text{volt} \times \text{ampere} \times \text{volt}} = 41.5 \text{[N Volt]}
\]

Out of this a theoretical graph can be made, given in the solid line in figure (16), the other figures are experimental tests to see whether the reality fits the theory. All are done by putting a certain voltage on the setup and checking with a steelyard what force is exerted on the arm. The results are put in the same figure as the experimental values and compared. The data from the experiments can be found in appendix 5.
Figure 15: error with and without feedforward

What may be concluded from these figures is that the experimental values are relatively close to the theoretical one. A reason for differences could be the changes in temperature. Conclusion is that the voltage to force transmission is well tuned.
Figure 16: verification

![Force-voltage verification graph](image-url)
4 Conclusion, Recommendations and expectations

Former research research [3] is continued for the high frequent part of the setup.

This lead to the conclusions:

- The response of the master and the slave is said to be the same
- closed loop measurement is best taking into account the high frequent noise.
- applying the earlier made controller, leaving out the high frequent filter, is a good way to identify the high frequent dynamics of the system.
- Eigenfrequencies are found at 200Hz, 350Hz and 800Hz using this controller.
- The system holds a delay of $1/1600 = 0.625 \text{ ms}$
- to get a robust and stable system a controller is designed at a crossover frequency of 102.09 Hz.
- 41.5 N matches 1 volt
- Feedforward is applied to reduce the error ($\pm 0.015 [\text{rad}]$)

Not all of the set goals are achieved, some research for the future stays open.

Recommendations for future research:

- motor-forcesensor FRF + controller;
  The goal was to make an identification of the system with position-feedback and with force-feedback. Because a lack of time the last one could not be implemented (neither could the controller),

- control scheme according to "Haptic control for Dummies" by Frederik Klomp ([1]);
  Also because time was running out this control scheme is left for future research.

- A dynamic model; When the masses decouple a model can be made for this. This model can be linked to the experiments and constants (like spring stiffness or damper constants) can be fitted.

- fix the model;
  The setup can be fixed better, to isolate it more from vibrations from surroundings.
5 Final remarks

Since this was my Bachelor final project my main goal was to learn a lot in the field of control engineering. That is a goal I, in my opinion, certainly achieved. On the other hand, I expected at the start of this project a little bit more real master-slave work (giving a input at the master, see how the slave reacts and feel the reaction back in the master) . I only did the work at one of the two setups (master). Further I am also a little disappointed that I could not finish my goals, since the real innovative part of the master-slave setup is the force-feedback and I did not really went on with that part.

Concluded, I really enjoyed working on the haptics setup, it took a lot of time and I did not finish the goals, but I learned a lot.

Finally I like to thank all people that coached me: Ir.R. Hendrix, ing. J. Scholtes and Prof.dr.ir.M. Steinbuch.
APPENDIX

A  appendix 1: FRF: coherence

Figure A-1: coherence robust controller process sensitivity and sensitivity
Figure A-2: coherence 0.2*robust controller process sensitivity and sensitivity
Figure A-3: 0.05*robust controller process sensitivity and sensitivity
B appendix 2: Reference trajectory

Figure B-1: Reference signal
Figure C-1: simulink closed loop model
D appendix 4: open loop model simulink

Figure D-1: Simulink open loop model
E appendix 5: calibration data

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Figure E-1: calibration table
Figure F-1: Feedforward tuning viscous friction
References


