The design of an experimental paperpath setup

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The design of an experimental paperpath setup

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Summary

The goal of the project described in this report was to construct an experimental setup of the paperpath of a copier. Although it was not possible to create a fully operational setup within the time available, this report describes the most important designs, choices, implementations and assemblies which ultimately led to an almost finished setup, that meets the requirements.

First of all, a mechanical frame was designed and constructed, after which various mounting parts were designed and produced. These mounting parts were used to assemble various components like pinches, actuators, encoders and a sheet conduction system to the mechanical frame. After this, appropriate actuators and amplifiers were chosen and implemented. Furthermore, encoders were chosen and implemented and a Paper Input Module was constructed. This report also presents the initial research that was done on a sheet position measurement system. For this case the possibility of using optical mouse sensors was introduced and further recommendations were made. Finally some research on data acquisition boards was done and recommendations regarding these boards were made.

After assembling the setup, some frequency domain identification experiments had been carried out, in order to measure the pinch dynamics. Both motor and load feedback measurements were performed, which were further analyzed and explained in this report.
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Chapter 1

Introduction and project description

The analysis of dynamical systems, together with the control technology of such systems, is a continuously evolving research field. The past few years have brought all kinds of new control theories, like LQG-control, robust control and iterative learning control, each one achieving more performance in one or another way. The development of such control theories is partly driven by more advanced computers and software, which allow us to make more complex simulations of our models and theories. Still, simulations are not enough and every new theory has to be tested in practice, which brings on the need of experimental setups.

The internship project described in this report is an example of the creation of such an experimental setup. It is closely related to other projects which use an Océ copier as a case study to investigate and design advanced real-time controllers. These research projects emphasize on the control of the copier paperpath. Therefore the project described in this report focusses on the creation of an experimental setup of a simplified copier, in such a way that several paperpath controllers can be implemented.

1.1 Project background and context

The project in this report is part of the Boderc project. Boderc is an acronym and stands for "Beyond the Ordinary: Design of Embedded Real-time Control". Started in September 2002, it is the very first project of the Embedded Systems Institute (ESI), located in Eindhoven. Eight parties cooperate with ESI in the Boderc project, including five industrial companies (under which Océ) and three universities (under which Technische Universiteit Eindhoven).

The Boderc project focuses on the design of distributed embedded real-time controllers of complex systems. An Océ copier is taken as a case study and acts as a driver for the Boderc project and several smaller projects derived from it. The target is an integral approach for a systematic architectural design, modeling, analysis, and validation methodology of heterogeneous systems such as the copier.[1]

1.2 Project description

A copier consists of many subsystems, but the project in this report focuses on just one part of the copier, namely the paperpath. Although there are many ways to express the performance of a copier, the designed experimental setup limits itself to paperpath performance, like throughput, paper speed and inter-sheet distance.
The paperpath of a copier is basically formed by multiple pinches, each pinch consisting of a driven (active) and a non-driven (passive) roller. These actuator driven pinches transport the sheets of paper from an input to an exit station. The system consisting of all sheet driving pinches can be modeled as a piecewise linear system, since every driven pinch behaves linearly by approximation. Such a model can be used to design advanced sheet controllers. Although today’s copiers already achieve high performance, this is often done open loop without using sheet controllers, so expensive parts with known dimensions and dynamics have to be used to achieve the desired performance. The usage of advanced and robust sheet controllers should make it possible to use cheaper parts, thereby lowering the costs of a copier, with equal or better performance.

The goal of this project is therefore to construct an experimental setup of the paperpath of a copier, with which sheet control techniques can be tested in reality. This means that the experimental setup only needs to contain those components necessary to transport paper, together which a measurement system and a suitable data acquisition system.

1.2.1 Project elements

The design of the experimental paperpath can be divided into a number of subdesigns, which will extensively be discussed in the following chapters. The pinches, consisting of a driven and a non-driven roller, the transmission and the digital sheet sensors did not need to be designed, since they were supplied by Océ. A short overview of the paperpath elements is given below.

Mechanical frame. Chapter 2 deals with the design and realization of a frame on which all the components of the paperpath have to be mounted. This also includes the design of various parts, like pinch suspension and paper conduction, to connect these components to the frame and the location of these parts in the frame.

Actuators and power supply. The choice of the pinch actuators, given the Océ supplied transmission, is treated in depth in chapter 3. It also discusses the difference between current and voltage based actuation and selects the appropriate amplifiers.

Data acquisition system. Chapter 4 explains the choice of actuator and pinch encoders, and discusses some possible sheet position sensors, like optical mouse sensors. Furthermore it compares some common data acquisition boards from different manufacturers.

Input and exit stations. A paperpath should also have a beginning and end. This means that a device has to be created with which sheets can be ejected, sheet by sheet, into the paperpath. This is called the paper input module (PIM), and is treated in chapter 5. This chapter also shortly discusses a possible exit station.

Unfortunately there was insufficient time to completely finish the project and the experimental setup. The final stage of the setup did not include an implementation of the sheet position sensors and data acquisition system. It was possible though, to estimate the dynamics of a single actuator-pinch set using frequency response function measurements, which are described in chapter 6.

1.2.2 Targets and requirements

In order to design a proper experimental setup, it is necessary to define restrictions, requirements and targets for the setup. The total construction should meet the following targets:
1.2. Project description

- The setup must transport A4-paper in landscape orientation. This means that the minimal width of the paperpath is 297 mm.
- The setup should be able to transport sheets with 60 ppm (pages per minute).
- The setup should be able to achieve sheet speeds of at least 0.5 m/s.
- It is desirable to be able to see the sheets while they are traveling through the paperpath. For this high visibility demand an open setup is required.

Other requirements and targets are categorized by subdesign.

Mechanical frame
The mechanical frame should be large enough to host six pinches, although only five pinches were received from Océ. This is why the initial design was done for six pinches, while the rest of this report focuses on using only five. The distance between two pinches should be flexible, in order to investigate the influence of this distance on the sheet flow. This also implies flexible positioning of other parts, like the actuators. The default distance between pinches is an integer $n$ times the perimeter of a single roller: $n \cdot 2\pi r$. Furthermore, there should be space enough to actuate each pinch individually, or in some cases, to actuate two pinches by one actuator. Finally, it is desirable to be able to vary the pinch pressure (pressure between driven and non-driven roller) to examine its influence. This means that the distance between both roller axes should be adjustable.

Actuators and power supply
The desired power of the actuators (one for each pinch) depends on the inertia and friction of pinches and transmission. The actuators should be able to transport sheets with 0.5 m/s and, according to Océ, provide a sheet acceleration of approximately $5G \approx 49 \, \text{m/s}^2$. Furthermore, in some experiments the coupling of pinches is desired, so each actuator should be able to drive two pinches. The transmission ratio is fixed, and given in section C.4.

Data acquisition system
For all used sensors a resolution of 0.1 mm sheet translation, or equivalent pinch or actuator rotation, is demanded. It must be possible to measure the rotation of each actuator, so encoders will be used for this purpose. Furthermore, to measure frequency response functions, it is recommended to measure the rotation of at least one pinch. Finally, possibilities for making accurate measurements of the sheet position (with 0.1 mm resolution) should be explored. Regarding the data acquisition a comparison should be made between different manufacturers. The data acquisition board should have enough inputs and outputs to fully control the total experimental setup, and should be fast enough to handle this. It is desirable to embed the data acquisition solution with Matlab®.

Input and exit stations
In order to achieve a 60 ppm throughput of the experimental setup, the paper input module should also be able to supply 60 sheets per minute. Of course, this PIM should only feed one sheet at a time, instead of multiple sheets on top of each other. The exit station should only guarantee that sheets actually leave the paperpath.

All these targets and requirements have been taken into account in the design of the experimental setup, described in the following chapters.
Chapter 2

Mechanical frame design

The Technische Universiteit Eindhoven (TU/e) was not the first university to make an experimental setup of a copier machine. The University of Twente (UT), which also participates in the Boderc project, was simultaneously working on a similar setup, for a slightly different research. In designing the mechanical frame for mounting all the other components, which was the first important step in creating the experimental setup, the state of the Twente design in April 2005, when this report’s project started, was taken as a starting point. This starting point is shown in figure 2.1, where the sheets of paper are transported from right to left. Figure A.1 in appendix A identifies some standard parts in this drawing.

This drawing shows that the non-driven rollers (the black rectangles) were connected to the driven rollers via a plexiglass bridge. These driven rollers in turn were connected to the outside of the outer bars of an aluminum frame. The paperpath was therefore situated a few centimeters above the frame. Furthermore, the drawing shows three wires which were supposed to conduct the sheets between the pinches.
2.1 Basic frame design

In order to construct a suitable mechanical frame for the experimental setup of this specific project, the Twente design needed to be adjusted and extended, keeping in mind various requirements. As already mentioned in section 1.2.2, the mechanical frame should host six pinches, be transparent (for visibility) and allow flexible placing of the different parts. Besides these requirements it is also important that the frame is stiff, to prevent unwanted vibrations and movements of the construction during operation.

Considering these requirements, some concepts of the Twente design proved to be quite useful and were therefore copied:

- The base of the Twente design was formed by aluminum profiles supplied by Boikon. A drawing of such a profile is shown in Figure A.1(d) in appendix A. These profiles proved to be light, stiff and easy to work with, enabling an easy assembly. Furthermore, these profiles enabled a flexible positioning of all the parts, since these parts can be mounted anywhere along the profile, using the supplied bolts and nuts.
- Since the same material was used as in Twente, the base design, consisting of two long outside bars (30 mm wide) connected by short bars (20 mm wide) in between, was also copied. The connections between these bars (special triangles) proved to be quite stiff. However, the total rectangular shape has one internal degree of freedom (torsion), which was solved by using four (instead of three) stands. This guaranteed that the frame is static definite.
- The conduction of the sheets from pinch to pinch was done by small wires. With respect to visibility this is of course a much better solution than using plates. The use of wires is also more flexible, since height and tension can easily be adjusted, and friction can be neglected, especially when using steel wire. The influence of electrostatic charge can also be neglected. Furthermore, sheets can not be cut by the wires, since there is no force present to push the sheets towards the wires. However, the physical implementation, treated in section 2.2, differs slightly from the Twente application.

Other concepts of the design were changed though, for various reasons. These adjustments, together with some extensions are discussed below:

- First of all the design was extended to contain six instead of four pinches (as was discussed in section 1.2.2, the final realization contains only five pinches).
- In Twente the driven rollers were mounted on the outside of the outer bars (see figure 2.1), but this way there is too little space to mount a PIM between these bars. Therefore Twente chose to mount the PIM on top of the bars, but here the driven rollers were mounted on the inside of the outer bars, creating more space in between. The width of the paperpath is then approximately 35 cm.
- In order to increase visibility, the driven roller was placed above the non-driven roller, instead of below (as was done in Twente). The non-driven roller is covered on top by a black case, blocking sight to everything below. By placing the non-driven roller below, one can see what happens inside and have a clear sight on the driven roller.
2.2 Part design

- The Twente design connected the non-driven roller via a plexiglass connection to the driven roller, which is not very stiff. By placing the non-driven roller underneath the driven one, it is possible to connect the non-driven roller directly to the frame, thereby increasing stiffness to a large extend.

- By doing so, it is also possible to reduce the height of the driven roller above the frame. This way the paperpath could be shifted right above the frame, instead of a few centimeters above. This again increases the stiffness of the construction.

- This low height of the paperpath forces the actuators to be mounted on the outside of the outer bars, since there is no more room inside (as was done in Twente). The extra advantage of this is the increased accessibility of the actuators.

- Since one of the requirements in section 1.2.2 is to vary the pinch pressure for each pinch individually, a special part was designed to vary the distance between driven and non-driven roller. This is further discussed in section 2.2.

- Finally, in order to measure the rotation of one pinch, some parts were designed to attach an encoder to a driven roller. This is also discussed in section 2.2.

The various parts needed to realize these adjustments are discussed in the following section. The corresponding Microsoft® Visio® drawings can be found in appendix section A.2.
of the driven roller then simply clicked into these holders. This way, with two such plates and corresponding holder, each driven roller was fixed to the frame.

Each driven roller has one conical side with an M4 tap hole. The gearwheels supplied by Océ exactly fitted on these conical shapes, and could be fastened using 15 mm M4 bolts.

The actuators, described in chapter 3, were mounted to the outside of the outer bars with the plates shown in figure A.3. Each actuator has three M2 tap holes in the front, which correspond to the three holes in the upper part of the plate, so each actuator was simply fixed using three M2 bolts. The plate itself was mounted to the frame using the two lower holes. The height of this plate was carefully minimized, such that the actuator axis was positioned just above the frame, since the shorter the plate, the stiffer the construction.

Figures A.4 and A.5 show the parts needed to correctly mount an (incremental) encoder to a driven roller. An encoder basically consists of a case, attached to the fixed world, and a disc attached to a rotating axis within this case. The rotating disc contains a large number of slits through which a ray of light shines on a phototransistor. When the disc moves, the phototransistor counts the number of slits (or increments). In order to connect the encoder disc to the rotating roller, the roller axis needed to be lengthened. Therefore a special bolt was designed (figure A.5), which could both fasten the gearwheel and lengthen the axis, using the M4 tap hole in the roller. The encoder case was connected to the fixed world by using the plate in figure A.4. The lower two holes were used to attach this part to the outside of the outer bars, such that the lengthened roller axis fitted through the upper hole. The three M1.6 holes around this axis hole were used to fix the encoder case to the plate.

The parts in figure A.6 were used to create a sheet conduction system. Between two plates of the kind in figure A.6(a) four rods (like in figure A.6(c)) were fixed, using the upper four holes of the plates. Next, these plates were mounted to the inside of each of the outer frame bars using their lower two holes. On the top and bottom of this construction, the part of figure A.6(b) was fixed. In each of the four tap holes of these strips an M6 bolt with a steel wire attached to it was screwed. By placing such a construction both at the beginning and the end of the paperpath, eight steel wires could be placed along this paperpath (four on the top and four on the bottom), which are guided by the rods and could be tightened by the bolts. This way the distance between two wires, which is the height of the paperpath, is approximately 4 mm. The funnel at the beginning of the conduction system, formed by the placement of the four rods, will make the insertion of sheets in this relatively thin path easier.

All the parts described here were made out of aluminum, since this material is easy to process, relatively light and reasonably stiff. The only exception was the wire-guiding rod. Since this part is exposed to high tensions the stiffness of the material is far more important than for example the mass. Therefore steel was chosen instead of aluminum.

### 2.3 Part assembly

The total design and assembly of the frame is shown in figure 2.2. In this drawing, the paperpath is supposed to go from right to left. Using the formula mentioned in section 1.2.2 and the fact that the radius of a driven roller is 14 mm (so that \( n = \text{floor} \left( \frac{210}{2 \pi 14} \right) = 2 \)), the inter-pinch distance was set to \( n \cdot 2 \pi r = 2 \cdot 2 \pi 14 = 176 \) mm. The upper drawing (side view) does not contains as much detail as the lower drawing (top view), but is merely meant to illustrate the fixation of the non-driven roller and to show some aspects which cannot be seen in the top view. A complete list of used parts during realization of this frame can be found in appendix B.
Figure 2.2: Complete design of the mechanical frame
Chapter 3

Actuators

The term ‘actuator’ is a very general term, since it covers various types of actuators, like combustion engines and electrical motors, linear and rotary motors, etc. For pinch actuation rotary electrical motors are the obvious choice, considering their controllability and small size. Still, various possibilities remain, like the stepper motor and the DC-motor (Direct Current). The latter was selected for this project, because of its approximate linear behavior and ease of use.

After a general introduction on DC-motors, the selection process will be described, after which an appropriate actuator, actuation method (current or voltage based) and amplifier will be chosen.

3.1 Operating principles of the DC motor

3.1.1 Inside the DC-motor

The basic idea behind the DC-motor is illustrated on the basis of figure 3.1[3], where a two-pole DC-motor is schematically represented. A DC-motor contains two magnets: the armature (or rotor) and the field magnet. In most motors, as in figure 3.1, the former is an electromagnet and the latter a permanent magnet.

![Figure 3.1: Representation of DC-motor [3]](image)

The armature consists of a metal core with a thin wire coil. At its center, perpendicular to its core, is the motor axis. When a current is applied to the coil, the armature becomes magnetic, creating a north and a south pole. The north pole of the armature will be repelled by the north pole and attracted by the south pole of the field magnet and vice versa. This way the armature and the motor axis turn, finally coming to rest in the horizontal position, with the armature north pole directed to the south pole of the field magnet. But when the current is reversed at this point (when the armature still has some speed), the poles of the armature will flip, again forcing a half-turn motion of the axis. This current switching is done by the commutator and the brushes.

The brushes are two pieces of metal (or carbon), each one connected to a pole of the power supply. Each brush makes contact with one plate of the commutator, which is basically a pair of
3.1. Operating principles of the DC motor

Curved plates around the motor axis, each plate connected to one side of the armature coil. While the commutator rotates with the axis, each commutator plate alternately contacts one brush or the other. This way the current is constantly reversed and the rotation continues.

Such a two-pole motor has some drawbacks, though. First, at each turning point the commutator short-circuits the power supply, which wastes energy. Second, since the horizontal position of the armature is in fact a equilibrium point, the motor doesn’t have smooth dynamics. In fact, in some cases the motor can get stuck in that position. These problems can be overcome by using three (or more) instead of two poles, which is done in most DC-motors nowadays.

3.1.2 Behavior of the DC-motor

One of the advantages of a DC-motor is its linearity. Theoretically, a DC-motor has linear voltage-speed and current-torque relations. In practice this behavior is not exactly linear, but Maxon motors approximate it fairly well. Therefore this report restricts itself to that manufacturer.

The two linearities are defined by two characteristic values: the speed constant $k_\omega$ and the torque constant $k_T$, according to:

$$\omega = k_\omega \cdot U_{coil}$$
$$T = k_T \cdot I,$$  \hspace{1cm} (3.1a) \hspace{1cm} (3.1b)

where $\omega$ is the rotational speed of the motor axis, $U_{coil}$ is the voltage induced in the rotor coil (which is not equal to the total voltage over the motor, see section 3.4.1), $T$ is the motor torque and $I$ is the current in motor and coil.

The relation between speed and torque is somewhat different, since a motor can provide less torque when it runs faster. This speed-torque line is shown in figure 3.2. The solid line corresponds to the nominal voltage of the motor, which is a normal operating voltage. At this voltage the maximum motor speed is the no-load speed $\omega_0$. Furthermore, a DC-motor provides maximum torque when the speed is zero. This torque is called the stall torque $T_S$. Of course, the motor can also run on other voltages than the nominal voltage, which causes a parallel shift like the dotted line in figure 3.2. The slope of the line, or speed-torque gradient $\frac{\Delta \omega}{\Delta T} = \frac{\omega_0}{T_S}$, is a typical motor characteristic and remains constant. The flatter this gradient, the less the speed changes due to load variations.

Figure 3.3 shows a typical current-torque curve, where (3.1) is depicted as the solid line. The dotted line is equivalent to the speed-torque line. This figure again shows the stall torque $T_S$, corresponding to an input current $I_s$, which is called starting current. $T_F$ denotes the friction torque due to internal friction in the bearings and the commutator of the motor. The current to overcome this torque is called the no-load current $I_0$, where $T_F = k_T \cdot I_0$. The linearity between current and torque is a very convenient property of the DC-motor, since it directly couples input (current) with output (torque). This is further discussed in section 3.4.1.
In principle, a DC-motor can operate at any point in figure 3.2, that is, with any combination of voltage, speed and load torque. A DC-motor however is limited by other factors, most importantly by thermal phenomena. Due to internal friction in the bearings and the commutator, and due to the resistance of the rotor coil, the motor can heat up. This is why operation ranges are defined for each motor, like maximum speed and maximum continuous torque. Together with the nominal speed-torque line, these limits define the recommended operating range (dark area) of figure 3.4. Short term operation is allowed inside the white area, but operating outside this region will possibly harm the motor.

3.2 Choosing an actuator

Choosing a suitable DC-motor for the experimental setup is all about these operating ranges. One has to identify the operation points of the setup, and find a DC-motor for which these points lie inside the recommended operating ranges. Operating points are defined as \([\omega, T]\), which can be drawn in speed-torque diagrams like figures 3.2 and 3.4.

In order to do this, some parameters of the system at hand have to be estimated. It is assumed that one can simplify this system such that only a motor-driven load remains, with some amount of friction to the fixed world. Therefore, parameter estimation means identifying a load inertia \(J\), viscous damping \(b\) and dry Coulomb friction \(T_c\), such that the dynamics are simplified to

\[
T = J\dot{\omega} + b\omega + T_c.
\]

(3.2)

After this, some setup demands in terms of speed and acceleration have to be defined. This means identifying a maximum allowed acceleration \(\dot{\omega}_{max}\) and speed \(\omega_{max}\). These values then determine the maximum torque needed at the load, according to

\[
T_{max} = J\dot{\omega}_{max} + b\omega_{max} + T_c,
\]

(3.3)

which defines the extreme operating point \([\omega_{max}, T_{max}]\). Furthermore, the nominal (or constant) operating point \([\omega_{nom}, T_{nom}]\) has to be defined, where \(\dot{\omega} = 0\) and \(\omega = \omega_{nom}\):

\[
T_{nom} = b\omega_{nom} + T_c.
\]

(3.4)

The torque at the load is in general not equal to the torque supplied by the motor, since there is usually a transmission present between motor and load. Its transmission ratio \(i\) is defined as

\[
i = \frac{\omega_{load}}{\omega_{motor}} = \frac{T_{motor}}{T_{load}},
\]

(3.5)

with \(i\) generally smaller than one, which results in lower torques and higher speeds at the motor. The operating points at the motor are therefore defined as \([\omega_{max} i, i \cdot T_{max}]\) and \([\omega_{nom} i, i \cdot T_{nom}]\). By altering the transmission ratio \(i\), for example by using gearheads, the operating points can be shifted. When choosing an actuator and transmission ratio, the nominal operating point, which will be operated at for longer time periods, should lie inside the continuous operating range of the actuator. Since the extreme operating point will only be operated at for very short time periods, it is sufficient that this point lies inside the short-term operating range. One disadvantage of using
3.3 Actuator choice for the setup

The actuator selection method of section 3.2 was also applied to the experimental setup. The results are presented in this section.

3.3.1 Operating points

The parameters needed for the selection process and the speed and acceleration demands were estimated in appendix C. All these numbers led to an approximation of the required worst case operating points (neglecting the inertia of the motor and its gearwheel):

\[
\begin{align*}
[\omega_{nom}, T_{nom}] &= [700 \text{ rpm}, 2.4 \text{ mNm}] & \text{or} & & [700 \text{ rpm}, 4.8 \text{ mNm}] \\
[\omega_{max}, T_{max}] &= [1400 \text{ rpm}, 19 \text{ mNm}] & \text{or} & & [1400 \text{ rpm}, 38 \text{ mNm}]
\end{align*}
\]

one pinch per actuator two pinches per actuator

3.3.2 Final actuator choice

Using these operating points an actuator was selected, taking into account the requirements at the end of section 3.2. These requirements could be met with a simple 10 Watt motor, the Maxon RE25, in the most common 24V version. This motor and its speed-torque characteristic are shown in figure 3.5, in which the operating points of both the 'one pinch per actuator' and the 'two pinches per actuator' case are depicted.

Figure 3.5: The selected actuator and its speed-torque diagram
3.4 Other considerations

Before a DC-motor can be implemented in an experimental setup, some more decisions have to be made. These include the choice between current and voltage based actuation of the motor, and, resulting from this, the selection of a suitable amplifier.

3.4.1 Current vs voltage based actuation

A motor is fed by an electric current and can therefore be actuated by and controlled with either a voltage or a current input, both with a linear influence on the output (speed and torque respectively). This input has to be provided by an amplifier, which converts (preferably linearly) an analogue control signal, which is typically a $\pm 2.5\text{V}$, $\pm 5\text{V}$ or $\pm 10\text{V}$ voltage, to a suitable current or voltage. Converting voltage-to-current is far more complicated than a simple voltage-amplification though. Since ideal current sources do not exist, a current amplifier must contain a feedback loop in order to guarantee that the output current is equal to the desired current, independent of the load. This is why current actuation is often referred to as current control. However, a voltage source or amplifier with corresponding voltage-speed motor characteristic, does not need this feedback loop in order achieve the desired voltage (within certain limits of course), and is therefore often called voltage steering. The latter option is thus the easiest and cheapest option.

This option has some major drawbacks though, which are illustrated in the following examples. Figure 3.6 shows a DC-motor with its electric circuit, showing the resistance and inductance of the armature coil. The load in figure 3.6(b) behaves like (3.2), with $T_c = 0$. The motor linearity states that $\omega = k_\omega \cdot U_{\text{coil}}$. Note that $U_{\text{coil}}$ is not the same as the applied voltage $U_{\text{pole}}$, since

$$U_{\text{pole}} = U_R + U_L + U_{\text{coil}} = R I + L \frac{dI}{dt} + \frac{1}{k_\omega} \omega,$$

where $I$ is the current through the motor, which is the same at both the coil and the motor poles. This current $I$ is linear with the motor torque $T = k_T \cdot I$, supplied to the load:

$$J \frac{d\omega}{dt} + b \omega = k_T I$$

When $I$ and $\omega$ are considered as states of the combined system (3.7)-(3.8), and $U_{\text{pole}}$ is chosen as

![Figure 3.6: Dynamic representation of a DC-motor](image-url)
the input, the system can be written in a second order state space form:

\[
\begin{align*}
\frac{dI}{dt} &= -\frac{R}{L} I - \frac{1}{k} \omega L + \frac{1}{L} U_{\text{pole}} \\
\frac{d\omega}{dt} &= \frac{k_T}{J} I - \frac{b}{J} \omega
\end{align*}
\]

\[
\Rightarrow \frac{d}{dt}\begin{bmatrix} I \\
\omega
\end{bmatrix} = \begin{bmatrix} -\frac{R}{L} & -\frac{1}{k} \omega L \\
\frac{k_T}{J} & -\frac{b}{J}
\end{bmatrix} \begin{bmatrix} I \\
\omega
\end{bmatrix} + \begin{bmatrix} \frac{1}{L} \\
0
\end{bmatrix} U_{\text{pole}}.
\] (3.9)

In order words, there is a second order dynamic relation between \(U_{\text{pole}}\) and \(\omega\), instead of a linear dependency. If the current \(I\) is taken as the input of the motor, the system in (3.9) reduces to

\[
\frac{d\omega}{dt} = -\frac{b}{J} \omega + \frac{k_T}{J} I,
\] (3.10)

which is simply the dynamics of the load, so no motor dynamics are involved; the linear relation between \(I\) and \(T\) still holds. This linearity and the fact that the motor dynamics are irrelevant, are clearly big advantages of current control over voltage steering.

Furthermore, current control is more gentle for the motor, which is illustrated with a simple open-loop simulation of both cases. Figure 3.7(a) shows a stepwise input for both \(U_{\text{coil}}\) and \(I\). Their amplitudes were chosen such that the steady state velocity changed stepwise from 0 to 10 to 20 to 0 rad/s. The results for both cases are shown in the speed-torque diagram of figure 3.7(b). When voltage is taken as the input, one can see that the motor quickly moves \((U_1\text{ and } U_3)\) to a higher speed-torque line corresponding to a higher voltage, with a resulting high torque. After this, speed increases and torque decreases along the speed-torque line \((U_2\text{ and } U_4)\), until the desired speed is obtained. With current as an input, the torque will first jump to a higher value \((I_1\text{ and } I_3)\), corresponding to \(T = k_T I\), after which the motor will increase its speed with a constant torque (it moves vertically in the speed-torque diagram, so voltage increases) until the desired speed is reached \((I_2\text{ and } I_4)\). It is clear to see in figure 3.7(b) that the former case will always result in more extreme combinations of speed and torque, maybe even outside the recommended operating ranges of the motor, thereby needlessly burdening it. This behavior is inherent to voltage steering, which again shows that current control is preferred.

Because of these advantages, current control was chosen for the experimental setup, despite the fact that the current amplifiers needed to implement this are more expensive.

![Graphs](image-url)
3.4.2 Amplifiers

The most important demand when selecting a suitable current amplifier, is that the amplifier is capable of providing the maximum current (at the right voltage; here 24V), needed to operate the motor at the maximum operating point:

$$I_{\text{max}} = \frac{T_{\text{max}}}{k_T} + I_0.$$  \hfill (3.11)

The values for torque constant $k_T$ and no-load current $I_0$ are motor dependent. For the Maxon RE25 these are $k_T = 43.8$ mNm/A and $I_0 = 14$ mA, resulting in a maximum current of

$$I_{\text{max}} = \frac{38}{43.8} + 0.014 = 0.89 \text{ A}.$$  \hfill (3.12)

Therefore, the standard small current amplifiers in the laboratory of Dynamics and Control Technology at the TU/e could be used, which amplify an input voltage between 0 and 2.5V linearly to a current: $I = C \cdot V_{\text{in}}$. The maximum current could be estimated with a simple experiment, where the load was kept at $\omega = 0$ and the input voltage was set to 2.5V. The resulting current in the motor, which corresponds to the maximum current (and torque), was 1.75A and was therefore large enough.

A similar experiment could be done to re-estimate the viscous damping $b$ in the setup. In a no load situation, using a 2.5V input voltage, motor and pinch turned at some maximum steady state speed and the current became 0.18A. The steady state motor speed was estimated with a tachometer to be 4600 rpm=480 rad/s. The damping $b$ could then be calculated, according to (3.4) and (3.5):

$$b = \frac{k_T l}{\omega} = \frac{43.8 \cdot 0.18}{0.486 - 480} - \frac{1.4}{0.486 - 480} = 0.064 \text{ mNms/ rad}.$$  \hfill (3.13)

This is almost twice as small as the worst case estimate of section C.2, which means that the maximum needed torque and current are also less. Furthermore, since the steady state current of 0.18A is smaller than the maximum continuous current of the motor (0.662A), the nominal operating point indeed lies inside the recommended operating range of the motor. The maximum achieved no load speed (4600 rpm) of the motor could be used to estimate the maximum sheet speed inside the paperpath:

$$v_{\text{max}} = \omega_{\text{max}} R_{\text{pinch}} = 0.486 \cdot 480 \cdot 0.014 = 3.3 \text{ m/s},$$  \hfill (3.14)

which is much higher than the needed 1 m/s maximum sheet speed. This speed and the maximum available torque confirms that the actuators and amplifiers are capable for this setup.

Finally, it should be mentioned that each motor needs its own amplifier. Connecting multiple motors in series to one amplifier results in low speeds (since the voltage over each motor is lower), and connecting them in parallel results in lower maximum torques (since the current through each motor is lower). Both cases also introduce dependencies; applying additional load torques on one pinch, results in different torques and speeds of the other motors. Furthermore, since it is desirable to control each motor separately, it is better to equip each motor with its own amplifier.

All components mentioned in this chapter are included in the inventory list of appendix B.
Chapter 4

Data acquisition system

In order to create a controllable system, data has to be extracted from the system with which control efforts can be calculated, which have to be sent to the system again. In other words, a data acquisition system is needed, which basically consists of measurement tools (sensors) and a data acquisition board which communicates between the controller and the system.

4.1 Sensors

This chapter will start by selecting appropriate sensors. Based on the demands in section 1.2.2, these can be divided into rotational sensors (or encoders) and sheet position sensors.

4.1.1 Encoders

Section 1.2.2 already mentioned that an encoder was needed for each actuator and for at least one pinch. The basic functioning of an incremental encoder was explained in section 2.2, from which it could be concluded that there are basically two parameters in choosing an encoder: axis diameter and number of slits on the encoder disc.

The encoder on the pinch side had to be mounted on the 8 mm axis extension bolt shown in figure A.5. The required number of slits of the encoder was calculated using the desired sheet position resolution $R_{\text{sheet}}$ of 0.1 mm. Using the roller wheel radius of 14 mm, this resolution corresponds to a rotational pinch resolution $R_{\text{pinch}}$ of

$$R_{\text{pinch}} = \frac{R_{\text{sheet}}}{2\pi r} = \frac{0.1}{2\pi 14} = 0.0011 \text{ rev}, \quad (4.1)$$

which corresponds to $\frac{0.0011}{0.0011} = 880$ counts per revolution. The standard Maxon incremental quadrature encoders have 500 CPT (counts per turn), but since the encoder output is a quadrature signal, the encoder actually returns $500 \cdot 4 = 2000$ counts per revolution. In other words, the standard encoder (HEDL-5540) is sufficient for this purpose, of course with an 8 mm axis.

The actuators, described in chapter 3, each have a 3 mm axis. With $R_{\text{sheet}} = 0.1$ mm, and transmission ratio $i = 0.486$, the rotational motor resolution $R_{\text{motor}}$ should then be

$$R_{\text{motor}} = \frac{R_{\text{pinch}}}{i} = \frac{R_{\text{sheet}}}{2\pi ri} = \frac{0.1}{2\pi 14 \cdot 0.486} = 0.0023 \text{ rev}, \quad (4.2)$$

which means $\frac{1}{0.0023} = 428$ counts per revolution. The standard Maxon incremental quadrature encoders (HEDL-5540) are again suited for the job. When actuators and encoders are ordered...
4.1. Sensors

Together, Maxon mounts the encoders to the actuators, so each of the five actuators were already equipped with the appropriate 3 mm encoder on delivery.

4.1.2 Sheet position sensors

To measure sheet position a device was needed with a preferred resolution of 0.1 mm. There are some devices available for this purpose, but most of these solutions are relatively expensive and not able to meet the desired resolution. As an example, measuring light screens and photo-electric sensors are mentioned, which use multiple small beams of light that are interrupted when a sheet passes by. They have optimal resolutions of 0.3 and 2.5 mm respectively. Although the former is quite accurate, it is only available in 1 cm devices.

An alternative is using a high speed camera. Its accuracy is determined by the frame rate (in Hz) and the frame resolution of the camera. To measure with 0.1 mm resolution at 1 m/s, high frame resolutions are needed, but in general frame rates at these resolutions are below 300 Hz. However, chapter 6 will show that the first resonance of the system is around 100 Hz, so it is advisable that the sample frequency is at least ten times 100 Hz (rule of thumb). In case of a high speed camera this means that an observer has to be used, which interpolates the 300 Hz data into 1000 Hz data, but this presumably influences the desired 0.1 mm accuracy. Therefore, such camera’s are not suited for these measurements, so higher resolution devices are desirable.

4.1.2.1 Introduction on optical mouse sensors

Optical computer mice, which are in fact very small high speed camera’s, form another option. Their working principle is illustrated in figure 4.1(a), [8] [9]. Light from a (usually red) LED is transported through a specially designed lens to illuminate the surface the mouse is working on. The LED light bounces off that surface and is projected through another lens on the mouse sensor. This mouse sensor, shown in figure 4.1(b), consists of three parts: the Image Acquisition System (IAS), the Digital Signal Processor (DSP) and the Serial Peripheral Interface (SPI). The LED reflection of the surface falls on the IAS, which makes images of this surface. The DSP then identifies patterns (light-dark-differences) in these images and calculates how much the mouse has moved based on the change in patterns, see figure 4.2. This $\Delta x$ and $\Delta y$ information is then sent to the SPI which sends the data to a mouse processor, from which the communication with the computer takes place.

Since mouse sensors can sample internally with very high frame rates (between 1500 and 6400 Hz) and have resolutions between 400 and 1600 cpi (counts per inch), these devices seem

![Cross section of an optical mouse](image1.png)

(a) Cross section of an optical mouse

![Optical mouse sensor](image2.png)

(b) Optical mouse sensor

Figure 4.1: An optical mouse and its sensor [7]
very useful for the setup. The idea is to mount an optical mouse or a sensor made with an optical mouse sensor shortly above the paperpath. The movement of the sheets underneath the mouse will be similar to moving the mouse itself. The surface of a blank sheet of paper is rough enough for the mouse to accurately detect its position and velocity.

4.1.2.2 Comparison between optical mouse sensors

Agilent, the supplier of Logitech, is the main manufacturer of optical mouse sensors. Agilent produces various types of optical mouse sensors, some of which are compared in table 4.1.

Table 4.1: Comparison between various optical mouse sensors

<table>
<thead>
<tr>
<th>Type</th>
<th>HDNS-2000</th>
<th>ADNS-2051</th>
<th>ADNS-3060</th>
<th>ADNS-3080</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling frequency</td>
<td>1500 Hz</td>
<td>2300 Hz</td>
<td>6400 Hz</td>
<td>6400 Hz</td>
</tr>
<tr>
<td>Max. resolution</td>
<td>400 cpi</td>
<td>800 cpi</td>
<td>800 cpi</td>
<td>800 cpi</td>
</tr>
<tr>
<td></td>
<td>63.5 µm</td>
<td>31.2 µm</td>
<td>31.2 µm</td>
<td>31.2 µm</td>
</tr>
<tr>
<td>Max. speed</td>
<td>12 inch/s = 0.30 m/s</td>
<td>14 inch/s = 0.36 m/s</td>
<td>40 inch/s = 1.02 m/s</td>
<td>40 inch/s = 1.02 m/s</td>
</tr>
<tr>
<td>Max. acceleration</td>
<td>0.15 g = 1.47 m/s²</td>
<td>0.15 g = 1.47 m/s²</td>
<td>15 g = 147 m/s²</td>
<td>15 g = 147 m/s²</td>
</tr>
<tr>
<td>Output</td>
<td>2 quadrature</td>
<td>2 quadrature</td>
<td>USB signal</td>
<td>USB signal</td>
</tr>
</tbody>
</table>

The HDNS-2000 and ADNS-2051 sensors are not able to meet the velocity and acceleration targets of section C.3 (1 m/s and 49 m/s²). The only difference between the ADNS-3060 and ADNS-3080 sensors is the resolution. Therefore, the ADNS-3080 was chosen for the setup.

Nowadays, there are also laser mice available, which operate in a similar way as LED mice. Their sensors show similar performance as the ADNS-3080, but are able to operate at smoother surfaces. This advantage is irrelevant here, so sheet measurements can be done with the ADNS-3080 or with a mouse containing this sensor. Although the implementation of these sensors is outside the scope of this report, some recommendations are given in the next section.
4.1.2.3 Implementation recommendations

The implementation of the optical mouse as a sensor could be quite difficult. The HDNS-2000 and ADNS-2051 both have quadrature outputs, which can be connected to and read with a simple incremental encoder counter. Four of the sixteen pins of these sensors are position signals $x_a$, $x_b$, $y_a$ and $y_b$, from which the displacements $\Delta x$ and $\Delta y$ can be derived directly [11]. The ADNS-3080 however, needs to communicate with a mouse processor in order to function properly. This processor requests data from the sensor, which is then sent over a four-wire serial port back to the mouse processor, which transforms this into USB-signals for the computer. Reading this signal as sensor data could be a problem. First of all, could such an USB signal be connected to a data acquisition system and how could the relevant serial data be filtered from it? Or could it be converted to an encoder signal? Second, is the communication fast enough? Normal USB-devices function at 125 Hz in Windows, limiting the sampling frequency which should be at least 1 kHz. In other words, the possibilities of this sensor and its interface should further be explored.

Furthermore, according to the data sheet of the ADNS-3080, the distance to the surface is quite critical. The sensor functions optimally when the bottom of the lens is mounted between 2.3 and 2.5 mm above the paperpath. Between 2.0 and 3.2 mm the sensor performance is still acceptable with resolutions higher than 1400 cpi and maximal errors of 3 per 6400 counts at 0.15 m/s, which corresponds to 0.048 mm per 101.6 mm displacement (0.047%). Outside these limits the error will rise and the resolution will drop dramatically, see [7].

Finally, in order to measure the sheet position at any place along the paperpath, six sensors (or mice) are needed, i.e. one shortly after each pinch and one for the PIM.

4.1.3 Digital sensors

In industrial copiers the sheet position is usually not measured continuously. Instead, ordinary reflective optical sensors are used. These digital sensors are placed at several places along the paperpath and fed by a 5V source. As soon as a sheet passes by the sensor the output voltage changes from high to low, and back to high when the sheet has passed. Since these sensors are only triggered when a sheet is passing by, no position measurements can be done but instead they can be used to verify the mouse sensor measurements.

Océ supplied sixteen of these digital sensors, namely the Sharp GP2A25, which is a reflection type photointerrupter. These photointerrupters should be mounted between 3 and 7 mm above the paperpath for optimum detecting and be fed by a 5V source. Connection to a data acquisition board could be done using digital I/O ports, since the sensor only returns high or low voltages. More information can be found in [12].

These digital sensors were mounted to the frame with aluminum bridges with low stiffness. This way the height of the sensors above the paperpath could easily be altered. The sensor interface was not yet implemented at the time this report was written.

4.2 Data acquisition boards

The final part of the data acquisition system is the data acquisition board itself, which is basically a device which reads the sensor signals, communicates these values with the controller,
and then sends control inputs to the actuators based on calculations done by the controller. Each data acquisition solution consists of a terminal (or I/O) board and a control board. The first one contains all connection plugs with which the sensors and actuators can be connected. This board is connected to the control board which is mounted inside a computer (e.g. via a PCI-bus) and performs all necessary operations.

4.2.1 Comparison

There are multiple manufacturers of data acquisition boards, each with their own (dis)advantages. In order to select an appropriate board, some requirements have to be formulated. The most important demand is the needed number of in- and outputs of each interface. For the specific setup of this report these are listed in table 4.2.

Other demands include channel resolutions and the conversion times of each channel. The latter defines the delay time of the system and can limit the sample frequency (which should be at least 1 kHz, see section 4.1.2). Appendix D makes a comparison between data acquisition boards of Quanser, Humusoft and dSpace regarding these demands. A final choice was not made yet, since the physical implementation of the sheet position (mouse) sensors and consequently their connection to an acquisition board, was still unknown at the time of this writing. When selecting an acquisition board though, one should consider that it is possible to use multiple boards of the same type, for example when a single one has too little in- or outputs.

Depending on the final mouse sensor implementation, the TUeDACS system might also be an alternative. A single TUeDACS has two encoder inputs, two 16-bit ADC and two 16-bit DAC channels and an 8-bit I/O port. If the mouse sensor signal can be converted to an I/O, encoder or ADC signal, three TUeDACS will be sufficient to actuate all DC-motors and measure the sheet position. The TUeDACS solution can also be implemented with a USB interface, which might be advisable when the mouse sensors are read with a USB interface. In that case a USB hub can be used to connect all six sensors and three TUeDACS to one computer.

However, some care must be taken when combining multiple data acquisition boards or TUeDACS. Connecting them in series will limit the sample frequency and introduce delays to a large extent, since more data has to flow through the same bus. Connecting the devices in parallel will only influence the calculation time, which is normally relatively small and will thus hardly influence the delay. Investigating this phenomenon for this setup is advisable though.

<table>
<thead>
<tr>
<th>Part</th>
<th>Interface</th>
<th>Qnty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pinch actuator</td>
<td>DAC</td>
<td>5</td>
</tr>
<tr>
<td>Actuator encoder</td>
<td>Encoder</td>
<td>5</td>
</tr>
<tr>
<td>Pinch encoder</td>
<td>Encoder</td>
<td>1</td>
</tr>
<tr>
<td>Digital sensors</td>
<td>Digital I/O</td>
<td>16 (or less)</td>
</tr>
<tr>
<td>PIM actuator</td>
<td>DAC</td>
<td>1</td>
</tr>
<tr>
<td>PIM encoder</td>
<td>Digital I/O</td>
<td>1</td>
</tr>
<tr>
<td>Optical mouse sensors</td>
<td>Unknown</td>
<td>6</td>
</tr>
</tbody>
</table>
Chapter 5

Input and exit module

The final step in the creation of the setup is designing a paper feed mechanism to insert paper into the paperpath, and a finisher which collects all the sheets at the end of the paperpath.

5.1 Paper Input Module

The paper feed mechanism is usually called the Paper Input Module (PIM). There are various new designs thinkable to feed paper into the funnel of the wire conduction system, but instead it was chosen to use a proven concept, namely the PIM of a laser printer.

Therefore an old HP Laserjet 4M was taken apart and its PIM and paper tray (with a capacity of 250 sheets) were taken out. The original stepper motor of the PIM was replaced by a DC-motor. The paper tray had a portrait orientation, which needed to be converted into landscape. Therefore the paper tray was cut in half and aluminum plates were fixed in between in order to make the tray wider. Next a partition was mounted to shorten the tray to prevent the sheets from slipping backwards. Furthermore, in order to be able to mount them as close to the paperpath as possible, as much material as possible was removed from both PIM and tray, without losing too much stiffness. The result is shown in the pictures of figure 5.1.

These pictures help to explain the PIM mechanism. The paper in the paper tray is, by means of a spring, pushed upwards against a small triangle on the side of the paperpath. Situated slightly above this triangle is the feeder mechanism, which consists of a rubber half wheel and, on the same axle, a special gearwheel. This gearwheel, tensed by a spring, misses a number of teeth, so that in rest it does not make contact with the transmission and can be stopped by a lever. An electromagnet can withdraw this lever, so that the spring slightly rotates the gearwheel until its teeth grip into the constantly spinning transmission, causing the gearwheel to be driven by the motor. The rubber half wheel makes one complete revolution, gripping the top sheet and moving it a couple of centimeters, into the paperpath, after which it is stopped by the lever again. The small triangle and, more important, the high friction between sheet and rubber wheel guarantee that only one sheet at a time enters the paperpath.

This mechanism thus enables a constant speed of the DC-motor and PIM, since a sheet is only inserted when a current is sent through the electromagnet. Tests have shown that this magnet triggers at approximately 14V, so voltages sources of for example 15, 24 or 30V could be used. Furthermore, because of the large load torque variations during one revolution, it is better to supply the motor with a constant voltage instead of a constant current, to prevent large voltage and thus speed drops. Alternatively, the PIM speed could be fed back and controlled.
5.2 Exit station

A special point of interest is the maximum speed of the PIM, since its transmission ratio $i_{pim}$ is rather small. While the sheet speed in the paperpath is 3.3 m/s at the most, the theoretical maximum sheet speed of the PIM is, assuming the same maximum motor speed as in section 3.4.2,

$$v_{pim, max} = i_{pim} \omega_{max} R_{pim} = \frac{4}{75} \cdot 480 \cdot 0.03 = 0.77 \text{ m/s},$$ (5.1)

which is theoretically about $\frac{0.77}{0.21} = 219$ ppm. Although the PIM is fast enough to operate at the nominal speed of 0.5 m/s, some care must be taken when the setup runs at high speeds. When speeds higher than 0.7 m/s are considered, a new PIM design might be necessary.

5.2 Exit station

The exit station, or finisher, is not an essential part of the setup. The paperpath setup would still function without a finisher, and therefore no effort was put in designing a special finisher during this project. The frame of the setup does have some space left for e.g. an ordinary tray, which would be sufficient for an exit station.
Chapter 6

Frequency response

The choices, designs and assemblies of the previous chapters finally resulted in the setup shown in the pictures of appendix E. These pictures show the state of the setup at the end of the project, including pinches, actuators, amplifiers, encoders and the PIM. Unfortunately the setup still lacked sheet measurement sensors (optical mouse sensors) and a data acquisition system.

Despite this, an initial system identification was feasible. The dynamics of a single actuator-transmission-pinch system is approximately linear and could therefore be measured using a frequency response function measurement. Since this involved only a single input (actuator current) and a single output (encoder position), a TUeDACS device could be used.

6.1 Theory of frequency response measurements

Linear systems exhibit a number of favorable properties, like their linear frequency behavior. If an input signal with a certain frequency \( f \) is applied to a linear system, the output signal has the same frequency \( f \). The output is simply an amplification of the input, plus a certain phase difference. The size of this amplitude and phase only depends on the frequency \( f \). If this response (amplitude and phase) is plotted for all \( f \), the specific frequency response function (FRF) of the system is obtained, often represented by \( H(f) \).

Measuring such FRFs comes down to applying an input \( u(t) \) which contains all frequencies, and measuring the output \( y(t) \), see figure 6.1. A Fourier transform returns the frequency contents \( U(f) \) and \( Y(f) \) of these signals, and the auto and cross power (or ‘energy’ content) are given by

\[
S_{uu}(f) = \lim_{T \to \infty} \frac{1}{T} U(f)U^*(f) \quad \text{and} \quad S_{uy}(f) = \lim_{T \to \infty} \frac{1}{T} U(f)Y^*(f).
\]  

(6.1)

The output \( Y(f) \) in figure 6.1 is, in the presence of measurement noise \( v(t) \), given by

\[
Y(f) = V(f) + H(f)U(f).
\]

(6.2)

Multiplying this with the complex conjugated of \( U(f) \) and then averaging over time returns

\[
Y(f)U^*(f) = V(f)U^*(f) + H(f)U(f)U^*(f) \\
S_{yu}(f) = S_{uu}(f) + H(f)S_{uu}(f)
\]

(6.3)
When there is no correlation between \( u(t) \) and \( v(t) \) the term \( S_{vu} = 0 \), so (6.3) simplifies to

\[
H(f) = \frac{S_{yu}(f)}{S_{uu}(f)}.
\]  
(6.4)

This is true when \( u(t) \) is chosen as white noise; then it contains all frequencies and is uncorrelated with the unknown disturbance \( v(t) \). The accuracy of this result is indicated by the coherence:

\[
Coh(f) = \left| \frac{H(f)S_{uy}(f)}{S_{yy}(f)} \right| = \left| \frac{S_{yu}(f)S_{uy}(f)}{S_{uu}(f)S_{yy}(f)} \right|.
\]  
(6.5)

In general, the obtained result for \( H(f) \) is quite reliable for those frequencies where the coherence is close to 1 (i.e. \( Coh(f) > 0.95 \)). Noise effects and non-linearities of the plant can cause low \( Coh(f) \), indicating less reliable result at the corresponding frequencies.

FRF measurements are normally done in closed loop systems as in figure 6.2 however, since open loop measurements are often too risky (they could harm the system). More importantly, closed loop measurements are more accurate since they usually result in better coherences. For closed loop measurements a zero mean white noise \( w(t) \) is inserted after the controller, and the resulting control input \( u(t) \) is measured as the output. The transfer between these signals is the Sensitivity \( S(f) \):

\[
S(f) = \frac{U(f)}{W(f)} = \frac{1}{1 + H(f)C(f)}.
\]  
(6.6)

Most Sensitivity functions are characterized by \( S(f) \to 0 \) for \( f \ll f_{bw} \), and \( S(f) \to 1 \) for \( f \gg f_{bw} \), where \( f_{bw} \) stands for the bandwidth of the closed loop system. This means that high frequent noise \( w(t) \) is well detected by the closed loop, but low frequent noise \( w(t) \) will hardly be amplified in \( u(t) \). In general this will cause bad coherence for \( f < f_{bw} \), but good coherence for \( f > f_{bw} \). Therefore, a controller resulting in a stable system with a very low bandwidth should be used when measuring FRFs. The system dynamics can then be derived from the Sensitivity using

\[
H(f) = (S(f)^{-1} - 1)C(f)^{-1}.
\]  
(6.7)

The controller \( C(f) \) needed for this calculation can be determined by a separate offline measurement of the open loop FRF of the controller, using the same parameters and settings as for the closed loop measurement. This returns \( Coh(f) = 1 \) for all frequencies and is therefore very reliable. Examples of Simulink® files and Matlab® code can be found in appendix F. More information on FRF measurements can be found in [13].

6.2 FRF results

The closed loop method described above was also used for the FRF measurements of a single pinch dynamics. Since both motor and driven roller were equipped with an encoder, both a motor and a load feedback measurement could be performed. Furthermore, since some parameters of the setup were adjustable, the construction has some degrees of freedom, whose influences on the dynamics were monitored using the same FRF measurements. In order to compare these parameters a standard (normal) configuration was defined:
6.2. FRF results

- low pinch pressure (non-driven roller mounted in lowest position),
- relatively high tension of the transmission belt,
- no sheets of paper between the pinches.

Four different parameter settings were investigated using four different configurations:
- **Case 1** No non-driven roller present.
- **Case 2** Maximal pinch pressure (i.e. non-driven roller mounted in highest position).
- **Case 3** Low tension of the transmission belt.
- **Case 4** Same as standard, only with a sheet of paper between the pinches.

The following sections will show and discuss the results for each configuration, for both the motor and the load feedback case.

6.2.1 Motor feedback

6.2.1.1 Influence of the speed

In chapter 3 it was assumed that the single pinch dynamics were linear. This is of course not exactly true, partly because of the friction present, which indicates that the dynamics might be dependent on the speed of the motor. This dependency was examined with a closed loop FRF measurement for the motor feedback case with the standard configuration. Three different jogmode (constant speed) references for the motor were applied, namely 10, 20 and 40 rad/s, and the FRF was measured using the method described in section 6.1, with a sample frequency $f_s$ of 8 kHz and a sample time $T_s$ of 600 s. The used controller consisted of a PD ($k_p = 10$, $k_v = 1$ and gain = 0.001) and a second order low-pass filter (cut-off frequency $f_c = 200$ Hz and damping $\beta = 0.2$), which resulted in a bandwidth below 1 Hz. For the 10 rad/s case the coherence and the Bode diagrams of both the controller and the Sensitivity are shown in figure 6.3, which indeed show that the low bandwidth controller resulted in high coherences for almost all frequencies.

The resulting plant dynamics for all three jogmodes are shown in figure 6.4. All jogmodes show approximately the same behavior:

![Controller and Sensitivity Diagrams](image)

*Figure 6.3: Coherence and Bode diagrams for motor feedback*
Section 6.2.1.2 attributes phenomena 2 and 3 to their mechanical equivalents using the results of appendix F. Nevertheless, there are two very small differences between the three jogmodes. First, the location of the large (anti-)resonance is slightly shifted to the left for higher speeds; speeds of 10 and 40 rad/s differ approximately 20 Hz. Second, low speeds seem to introduce a friction to the fixed world (due to the bearings), ultimately resulting in a $-\frac{1}{2}$ slope and a phase of $-90^\circ$ for frequencies far below the point of interest of this report (i.e. below 1 Hz).

So although the dynamics are not exactly linear, the non-linearities are fairly small and were therefore neglected. For the remainder of this section a jogmode of 10 rad/s was chosen.

### 6.2.1.2 Influence of various configurations

Appendix section F.1 shows the results for the various configurations (or cases) discussed in section 6.2. This section will analyse these results. For all cases the same $f_s$, $T_s$ and low bandwidth controller as in section 6.2.1.1 were used.

Figure F.4 compares the standard situation with case 1, where no non-driven roller is connected. It shows that the small resonance at 120 Hz is no longer present in this configuration, which can thus be attributed to the decoupling of the non-driven roller. Since the total mass is less due to the absence of the non-driven roller, the mass line is slightly shifted upwards for low frequencies, and the location of the transmission (anti-)resonance is shifted to the right. The damping of the (anti-)resonance is not altered.

The influence of the pinch pressure in shown in figure F.5. This figure shows that the influence of pinch pressure is hardly noticeable for any frequency, and is therefore neglected.
6.2. FRF results

Case 3, where the tension in the transmission belt is very low, is depicted in figure F.6. It corresponds to a low stiffness of the transmission, which results in a shift to the left of the large (anti-)resonance, while the other dynamics are not influenced. This illustrates that the large (anti-)resonance is caused by the decoupling of the driven roller from the motor axis due to the flexibility of the transmission belt. Furthermore, since the mass is not altered, the low frequent mass line remains the same.

Finally, figure F.7 shows the case when there is a sheet inside the pinch. The FRF seems identical to the standard configuration, but there are some important differences. Since paper has a limited length, the jogmode had to be set to 0 rad/s which introduces friction to the measurement and causes a small shift of the (anti-)resonance to the right, as was already shown in section 6.2.1.1. However, the presence of a sheet also introduces a very small stiffness to the fixed world, resulting in a slope of 0 and a phase of 0° for very low frequencies. When controlling this system an integrator might therefore be necessary in order to avoid steady state errors.

6.2.2 Load feedback

Appendix section F.2 shows the results for the load feedback FRF measurements, which will be discussed in this section. The controller and the parameters \( f_s \) and \( T_s \) were the same as before. The jogmode of the pinch was chosen to be 40 rad/s.

Figure F.8 shows the FRFs of both the standard configuration and case 1 (no non-driven roller). The standard configuration again shows the small (anti-)resonance of the non-driven roller, which is absent for case 1. For load feedback the anti-resonance of the transmission is absent, causing the slope of the magnitude to go from -2 to -4 after the resonance at 440 Hz (which is again shifted to a higher frequency for case 1). Furthermore, due to this -4 slope and the absence of the anti-resonance, the magnitude becomes very small at high frequencies, smaller than the resolution of the encoder can handle. Magnitudes of approximately -70 dB or smaller are not correctly measured by the encoder (encoder noise level), so figure F.8 shows a zero slope for frequencies higher than 1000 Hz. Furthermore, the FRF indicates the presence of high frequent dynamics between 600 and 1000 Hz, which is explained hereafter. Finally, note the absence of the encoder resonance. The load encoder is mounted on a ticker axle than the motor encoder (8 mm instead of 3 mm), resulting in an eigenfrequency larger than 4 kHz.

The assumption that the pinch pressure hardly influences the dynamics is confirmed by figure F.9, which shows hardly any difference between the standard configuration and case 2. Figure F.11, showing case 4, again shows the small shift in the resonance frequency (since the speed is 0 rad/s) and confirms the introduction of stiffness due to the presence of sheets.

Finally, figure F.10 compares case 3 with the standard case. As expected, the transmission resonance is shifted to approximately 50 Hz, while the other dynamics remain unchanged. However, the FRF indicates the presence of a new anti-resonance at 80 Hz and a resonance at 125 Hz. Although this phenomenon seems to occur solely in this figure, it is in fact present in all load feedback FRFs and equal to the high frequent behavior; both show the same phase lead and have the same location relative to the transmission resonance. This dynamic behavior is linked to the transmission, since it moves with the resonance of the transmission belt. In fact, when looking in detail at the motor feedback FRFs, one could recognize the same behavior: there is a very tiny wave in the amplitude and a very small phase lead around 800 Hz. Indeed, for case 3 of the motor feedback it is shifted to the left, creating a small phase lead on top of the one due to the decoupling of the non-driven roller. Although the actual mechanical counterpart of this behavior is unknown, it might be attributed to the decoupling of e.g. a gearwheel.
Chapter 7

Conclusions and recommendations

7.1 Conclusions

At the end of the project described in this report the experimental setup of the copier was in an advanced phase. Various designs, choices, realizations and implementations resulted in the final state shown in appendix E. This final state includes a mechanical frame (with pinches), various mounting parts, a sheet conduction system, pinch actuators with transmission and amplifiers, encoders, digital sensors and a Paper Input Module. In order to finish the setup a sheet position measurement system and a data acquisition system still have to be implemented.

The end state of the experimental setup met the requirements defined for it. First of all, it is flexible, adjustable and extendable; the frame and the mounting parts are constructed such that various parts can be (re)moved and other parts can easily be added. Second, the setup is very open, creating high visibility; for instance the wire sheet conduction system gives an open view of the sheets while traveling through the paperpath. Moreover, the chosen actuators and amplifiers are capable of meeting the required speed and acceleration and can be actuated separately.

Finally, this report concluded with some FRF measurements for the motor and the load feedback case, to visualize the pinch dynamics. The dynamic behavior in these FRFs was analyzed and explained for various configurations, which means that various (anti-)resonances were attributed to mechanical components, like the non-driven roller, the transmission and the encoder.

7.2 Recommendations

The obvious recommendation is of course to finish the setup. This report already introduced the optical mouse sensors as possible sheet position measurement sensors, and this possibility should be further examined. Furthermore a data acquisition board or system has to be chosen and embedded in the setup, for which this report already gave the introduction and defined some requirements. In case the PIM turns out to be not fast enough, which seems unlikely based on the requirements, a new PIM has to be designed. Finally, if desirable, a finisher could be constructed.

When sheet position sensors can be implemented the FRF between motor input and sheet position should be measured and analyzed. This way three different FRFs are obtained: motor, pinch load and sheet feedback. For each case feedback controllers can be designed, implemented and compared. Ultimately, this could lead to an advanced controller design for the complete paperpath.
References


Appendix A

Frame part drawings

This appendix shows drawings of the various parts used in the mechanical frame. All drawings were made with Microsoft® Visio® 2003.

A.1 Standard parts

The following figure contains simplified drawings of various standard components, components which did not have to be designed. These components include the actuators, the pinches (driven and non-driven rollers) and the aluminum profile.

![Standard parts drawings](image)

(a) Driven roller  
(b) Maxon actuator  
(c) Non-driven roller  
(d) Aluminum profile  
(e) Pinch holder

Figure A.1: Drawings of several standard components

A.2 Designed parts

This section shows the Visio® drawings of the various designed parts. These parts were specially designed during the project to mount various components unto the mechanical aluminum frame. Descriptions of the parts are given in section 2.2, and the total assembly of these parts is shown in figure 2.2 of section 2.3.
Appendix A.

(a) For the non-driven rollers

(b) For the driven rollers

Figure A.2: Mounting devices for the pinch rollers

Figure A.3: Mounting plate for the actuators

Figure A.4: Mounting plate for pinch encoder
Appendix A.

Figure A.5: Bolt to tighten gearwheel and connect encoder to

(a) Aluminum side plate

(b) Aluminum top construction

(c) Steel rod for guiding the wires

Figure A.6: Parts for mounting the conduction wires
This appendix shows a complete list of parts used in the final realization of the experimental setup. Ordered parts are identified by their part number (if present) and supplier, designed parts are identified with their corresponding figure number elsewhere in this report.

<table>
<thead>
<tr>
<th>Part no.</th>
<th>Supplier</th>
<th>Part description</th>
<th>Length</th>
<th>Qnty</th>
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<tr>
<td>13000</td>
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</tr>
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<td>Boikon</td>
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<tr>
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</tr>
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<td>Boikon</td>
<td>Damping ring for 45000</td>
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<td>4</td>
</tr>
<tr>
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<td>Boikon</td>
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<td>-</td>
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<tr>
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<td>Driven pinch roller</td>
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<td>Fig A.1(c)</td>
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<td>Non-driven pinch roller</td>
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<tr>
<td>Fig A.1(e)</td>
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<td>Océ</td>
<td>Pinch gearwheel (37 teeth)</td>
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<td>Océ</td>
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<td>Océ / Gates</td>
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*continued on next page*
### Appendix B.

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<td>Sharp</td>
<td>Digital sensor; photointerrupter</td>
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<td>GTD</td>
<td>Mirror of figure A.2(a)</td>
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<td>Fig A.2(b)</td>
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<td></td>
<td>GTD</td>
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<td>Fig A.3</td>
<td>GTD</td>
<td>Mounting device for actuators</td>
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<tr>
<td>Fig A.4</td>
<td>GTD</td>
<td>Mounting device for pinch encoder</td>
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<td>Fig A.5</td>
<td>GTD</td>
<td>Bolt for lengthening roller axis</td>
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<td>Fig A.6(a)</td>
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<td>Side plates for sheet conduction</td>
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<td>Fig A.6(b)</td>
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<td>Steel wires</td>
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<td>Paper Input Module</td>
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Appendix C

Parameter estimation for actuator choice

This appendix estimates some system parameters, in order to choose an appropriate actuator for each pinch. It was assumed that the dynamics of a single pinch could be simplified to:

\[ T = J \dot{\omega} + b \omega + T_c. \]  

(C.1)

C.1 Inertia

The total load inertia is formed by the pinch gearwheel and both pinch rollers. The mass of the driven roller is about 190 g, which can be attributed to its components (axis, wheels and bearings) using the specific masses of steel (7850 kg/m³) and rubber (1100 kg/m³).

The axis of the roller is made of steel, is 380 mm long \((L)\) and has a diameter \((d)\) of 8 mm. Each rubber wheel on the axis is 10 mm wide \((l)\) and has an outer diameter \((d_w)\) of 28 mm. The bearings each have an outer diameter \((d_b)\) of 22 mm and are 7 mm high \((h)\). Since the bearings contain oil, their density is lower than the density of steel and is assumed to be 6000 kg/m³.

Using these dimensions the masses were estimated to be

\[
\begin{align*}
m_{\text{axis}} &= \rho_{\text{steel}} V = \rho_{\text{steel}} L \frac{1}{4} \pi d^2 = 0.150 \text{ kg} \\
m_{\text{wheel}} &= \rho_{\text{rubber}} V = \rho_{\text{rubber}} \frac{1}{4} \pi \left( d_w^2 - d^2 \right) = 0.006 \text{ kg} \\
m_{\text{bear}} &= \rho_{\text{bear}} V = \rho_{\text{bear}} h \frac{1}{4} \pi \left( d_b^2 - d^2 \right) = 0.014 \text{ kg}
\end{align*}
\]

making the total mass of the roller \(m_{\text{driven}} = 150 + 2 \cdot 0.6 + 2 \cdot 0.14 = 190 \text{ g}\). Its inertia is approximately

\[ J_{\text{driven}} = \frac{1}{8} \left( m_{\text{axis}} d^2 + 2 m_{\text{wheel}} (d_w^2 + d^2) + 2 m_{\text{bear}} (d_b^2 + d^2) \right) = 4.4 \cdot 10^{-6} \text{ kgm}^2 \]

The driven roller is equipped with a 31 g gearwheel with an outer diameter \((d_g)\) of 23 mm. The inertia of this gearwheel is approximately equal to

\[ J_{\text{gear}} = \frac{1}{8} m_{\text{gear}} d_g^2 = 2.0 \cdot 10^{-6} \text{ kgm}^2 \]

The inertia of the non-driven roller is equal to the inertia of its bearings. These were assumed to be the same as for the driven roller:

\[ J_{\text{non-driven}} = \frac{2}{8} m_{\text{bear}} (d_b^2 + d^2) = 1.9 \cdot 10^{-6} \text{ kgm}^2 \]
Since the diameters of both rollers slightly differ, there is a transmission ratio between those two:

\[ i_{\text{pinch}} = \frac{d_{\text{driven}}}{d_{\text{non-driven}}} = \frac{28}{25} = 1.12. \]

The total load inertia then becomes

\[ J_{\text{load}} = J_{\text{driven}} + J_{\text{gear}} + i_{\text{pinch}}^2 J_{\text{non-driven}} = 8.8 \cdot 10^{-6} \text{kgm}^2. \]

### C.2 Friction

The Coulomb friction was obtained using a simple experiment. Applying a 4 g mass at 35 mm of the center of the driven roller brought the pinch into motion. This means that the Coulomb friction is approximately

\[ T_c = m g r = 0.004 \cdot 9.81 \cdot 0.035 = 1.4 \cdot 10^{-3} \text{Nm}. \]

The viscous friction is very small and was estimated using data supplied by Océ. These data indicated that in a worst case this friction is \( b = 10^{-4} \text{Nms/rd} \), which seemed like a reasonable value and was therefore adopted.

### C.3 Speed and acceleration demands

The nominal sheet speed was already mentioned in section 1.2.2, and was assumed to be 0.50 m/s. Since the roller radius is 0.014 m, this corresponds to a rotational speed of \( \frac{0.50}{0.014} = 35.7 \text{ rad/s} \), which is the same as 341 rpm. The torque needed to operate at this point is

\[ T_{\text{nom}} = b \omega + T_c = 35.7 \cdot 10^{-4} + 1.4 \cdot 10^{-3} = 4.9 \cdot 10^{-3} \text{Nm}. \]

In case two pinches are connected to one motor, the torque is doubled to \( 9.9 \cdot 10^{-3} \text{Nm} \).

The same section also mentioned the extreme acceleration of 49 m/s\(^2\), which is equal to \( \frac{49}{9.81} = 3500 \text{ rad/s}^2 \). The extreme sheet speed was set to 1 m/s, which is 71.4 rad/s (or 682 rpm) at the pinch. This maximum operating point then requires a torque of

\[ T_{\text{max}} = J_{\text{load}} \dot{\omega}_{\text{max}} + b \omega_{\text{max}} + T_c = 8.8 \cdot 10^{-6} \cdot 3500 + 71.4 \cdot 10^{-4} + 1.4 \cdot 10^{-3} = 0.040 \text{ Nm}, \]

which is doubled to 0.079 Nm when two pinches are connected to one actuator.

### C.4 Transmission

As mentioned before, the transmission between motor and load was supplied by Océ. This transmission consisted of two gearwheels and an enforced rubber belt. This toothed belt was 9 mm wide, 160 mm long and had a pitch of 2 mm. The transmission ratio was determined by the ratio between the number of teeth of the gearwheels. The gearwheel at the motor had 18 teeth, and at the pinch 37 teeth, making the transmission ratio, defined in (3.5), equal to \( i = \frac{18}{37} = 0.486 \).

All these parameters and demands finally lead to the nominal and the maximum operating point at the motor:

\[
\begin{align*}
[\omega_{\text{nom}}, T_{\text{nom}}] &= [\frac{\omega_{\text{nom, load}}}{i}, iT_{\text{nom, load}}] = [\frac{341}{0.486}, 0.486 \cdot 9.9] = [700 \text{ rpm}, 4.8 \text{ mNm}] \\
[\omega_{\text{max}}, T_{\text{max}}] &= [\frac{\omega_{\text{max, load}}}{i}, iT_{\text{max, load}}] = [\frac{682}{0.486}, 0.486 \cdot 79] = [1400 \text{ rpm}, 38 \text{ mNm}] 
\end{align*}
\]
Appendix D

Data acquisition board comparison

This appendix gives a short comparison between the best data acquisition boards of different manufacturers: Quanser, Humusoft and dSpace. The latter offers by far the most possibilities, since dSpace offers custom solutions where the type and number of in- and outputs are free to choose. Therefore the dSpace solutions are treated separately. It should be noted that all three options can be used in combination with Matlab® /Simulink®.

Quanser and Humusoft offer by far the least expensive solutions. Both manufacturers offer only a few different types of acquisition boards, of which the best options are shown in table D.1. This table compares the two products on number of in- and outputs and their resolution and conversion times.

Table D.1: Comparison between Quanser and Humusoft

<table>
<thead>
<tr>
<th></th>
<th>Quanser Q8</th>
<th>Humusoft MF614</th>
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<tbody>
<tr>
<td># DAC</td>
<td>8</td>
<td>4</td>
</tr>
<tr>
<td>DAC resolution</td>
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<td>12-bit</td>
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<tr>
<td># ADC</td>
<td>8 single ended</td>
<td>8 single ended</td>
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<tr>
<td>ADC resolution</td>
<td>14-bit</td>
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<td>ADC conversion time</td>
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<td># encoders</td>
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<td>4 single ended or differential</td>
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<td>Encoder resolution</td>
<td>24-bit</td>
<td>24-bit</td>
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<tr>
<td># digital I/O</td>
<td>32 programmable</td>
<td>8 in, 8 out</td>
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<tr>
<td># timers / counters</td>
<td>2</td>
<td>4 + 1 internal</td>
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<tr>
<td>Timer resolution</td>
<td>32-bit, 30 ns</td>
<td>16-bits, 50 ns</td>
</tr>
<tr>
<td>Max sample frequency</td>
<td>100 kHz</td>
<td>100 kHz</td>
</tr>
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</table>

Purely based on the data in table 4.2 and ignoring the mouse sensors, one single Quanser Q8 board or two Humusoft MF614 boards seem sufficient. The final choice should of course depend on the other requirements mentioned in section 4.2.1 as well as on the price (Quanser is more expensive than Humusoft). Furthermore, at this stage the data acquisition for the mouse sensors
As was mentioned before, an alternative is dSpace, which offers custom solution for much higher prices. The basic idea is that dSpace sells connector panels in which different types of terminal boards can be inserted. The assembled panel is then connected to an appropriate control board. A typical connector panel has 14 MU (measurement units) of available space, which can be filled by the terminal boards mentioned in table D.2.

dSpace offers two different control boards to go with these terminal boards, namely the DS1005 and DS1006. Furthermore dSpace also offers standard solutions like Quanser and Humusoft, by means of the DS1103 and DS1104. All of these four outperform the Quanser and Humusoft boards, however the same question remains whether such high performance (but expensive) boards are necessary for this specific experimental setup.

---

**Table D.2: Possibilities with dSpace**

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<th>Digital I/O</th>
<th>Timing I/O</th>
<th>Encoder</th>
<th>Space</th>
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<tr>
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<td>3 MU</td>
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<td>4 MU</td>
<td>4 MU</td>
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Appendix E

Setup pictures

This appendix shows some pictures of the experimental setup, in order to visualize parts and assemblies and to show the end state of the setup. Explanations are given in the figure captions.

Figure E.1: Pinches; notice the non-driven roller suspension and the wires for sheet conduction

Figure E.2: Suspension of a driven roller

Figure E.3: Transmission showing pinch gearwheel and transmission belt

Figure E.4: Transmission showing the motor and its gearwheel
Figure E.5: Motor, amplifier and encoders

Figure E.6: Sheet conduction funnel

Figure E.7: Total setup; view from the end of the paperpath

Figure E.8: Total setup; top view
Appendix F

Frequency Response Measurements

This appendix shows the results of the frequency response measurements described in chapter 6. These results are presented in a number of figures, subdivided in two cases: motor feedback and load feedback. Section 6.2 defined a standard configuration for each case, which is compared to four other configurations, defined by cases 1 through 4:

Case 1 No passive pinch present.
Case 2 Maximal pinch pressure (i.e. passive pinch mounted in highest position).
Case 3 Low tension of the transmission belt.
Case 4 Same as standard, only with a sheet of paper between the pinches.

The Simulink® files used for the FRF measurements are depicted in figures F.1 (closed loop measurement) and F.2 (controller measurement), by way of illustration. Their parameters and settings are discussed in section 6.2.1.1. Figure F.3 shows some example Matlab® code to estimate the process \( H \) using the Simulink® data.

![Figure F.1: Simulink® file used for closed loop FRF measurements](image1)

![Figure F.2: Controller measurement](image2)

![Figure F.3: Estimation of process \( H \) in Matlab®](image3)
F.1 Motor feedback

Figures F.4 through F.7 show the results of each of the four cases compared to the standard configuration for the motor feedback (MF) case. The black lines in each figure correspond to both magnitude and phase of the specific case, the gray lines denote the standard configuration.

Figure F.4: MF case 1: no passive pinch

Figure F.5: MF case 2: high pinch pressure

Figure F.6: MF case 3; low belt tension

Figure F.7: MF case 4; paper inside pinch

For cases 1 through 3 the motor was given a jogmode (speed) of 10 rad/s, in order to overcome the dry friction. Because of the limited length of a sheet, the jogmode for case 4 was set to 0 rad/s.

F.2 Load feedback

Figures F.8 through F.11 show the results of each of the four cases compared to the standard configuration for the load feedback case (denoted by LF). Again, in all figures the black lines show the magnitude and phase of each specific case, the gray lines denote the standard configuration.
The measurements of case 1 through 3 were done with a pinch jogmode of 40 rad/s, the final case was again measured with 0 rad/s.

Figure F.8: LF case 1; no passive pinch

Figure F.9: LF case 2; high pinch pressure

Figure F.10: LF case 3; low belt tension

Figure F.11: LF case 4; paper inside pinch

All results shown in this appendix were obtained with sample frequency \( f_s = 8000 \) Hz and sample time \( T_s = 600 \) s. The controller used for this measurement is described in section 6.2.1.1. Further analysis of the results is done in sections 6.2.1 and 6.2.2.