

Development of a vane motor servo system for precision
positioning in an MR- and CT-environment

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MSc Report

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September 2017

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Acknowledgements

Doing research and writing this thesis has been a long and interesting experience, at times fun, at times tough. I wouldn't have been able to reach this point without the many people around to support me.

First of all, I want to thank my parents, Anja and Mieke, for being extremely supportive of me all throughout my education these past 20 years. Without their unending support I would definitely not be here.

All of my friends deserve a thanks as well. During a time where all day, every day, is dominated by this thesis, they have been indispensable in showing me that there is, in fact, life besides graduating. The great times we had definitely helped reduce stress.

I would also like to thank my daily supervisor at Demcon, Maarten Arnolli. When he was not busy crossing 250 km of ice days before receiving his PhD, he would always be there to give constructive, useful, and sober advice for the project during our weekly meetings. I feel that our sparring has really improved my understanding of mechanical and control concepts.

Thanks also goes to my supervisors from the University of Twente, Vincent Groenhuis and Françoise Siepel. They were always just a small bike ride away and ready to give valuable input for my research and report.

The people I've met and worked with at Demcon also deserve a thank-you. My fellow interns/graduates Michael ten Den, Linda Lieferink, Just Herder, and Hidde Wieringa have provided many interesting conversations and afternoon walks. The regular employees such as Nicky Tahapary, Evyatar Bukai, Jos Vos, Erik Hoogma, and Sjoerd Ligthart also contributed to this with interesting talks during lunch breaks and company drinks. Finally, some of the employees, including Wouter Hakvoort, Martijn Buijze, and Peter Rutgers have provided some valuable input to the project and deserve a thank-you as well.

Summary

Electromechanical actuators are widely used in a range of applications because of their good speed, torque, and bandwidth characteristics. In a CT/MRI-environment, their application is limited as the materials used can cause damage to the system and patient, or degrade image quality. There is a need for precise actuators that can function in a CT/MR-environment without the aforementioned complications.

Various systems have been developed for use in a CT- or MRI-scanner [1, 2, 3]. However, these systems generally suffer from not being completely MRI-compatible or from having poor precision or speed characteristics.

The main objective of this thesis is to explore the possibilities of, and to develop, an actuator system that can be used to precisely position a needle guide in a CT- and MRI-environment.

Due to its high power-to-weight ratio, inherent cleanliness, simple design, and promising positioning precision [4, 5], a pneumatic vane motor forms the basis of a novel actuator system designed in this thesis.

Other works of literature cover the subjects of CT/MR-compatible actuation [1, 2, 3, 6] and positioning of vane motors [4, 5] separately. This thesis uses the knowledge on CT/MR-compatible actuation to design a pneumatic system that can be used in a CT- or MRI-scanner. An important distinction with normal positioning of vane motors in e.g. [4, 5] is that control valves, which typically use solenoid actuation, can not be present near the vane motor. Valves and actuator are separated by 5 meters of tube, introducing a delay into the system.

Dynamical behaviour of the vane motor is described using models from literature. The filling and emptying of pneumatic tubes is modelled as well. Insight gained by this modelling serves as the basis for a novel system design which includes valve and tube topology. A non-linear simulation model and a proof-of-principle setup are created from this system design.

Identification using the lab set-up and the non-linear model is done to clarify various system characteristics, such as the relation between torque and angle, which is non-linear. Several Linear Time-Invariant (LTI) input/output models are fitted to experimental data.

Three controllers are designed to achieve precise positioning results: a PID controller, a state feedback controller, and a controller utilising a brake attached to the vane motor's shaft.

PID control is implemented on the lab set-up to analyse the vane motor servo system's performance for various scenarios. In the majority of scenarios where control valves are placed close to the vane motor, a precision of 0.5° is achieved in 100% of the cases; other experiments attained a precision of 1° , in line with design goals. When tubes of 5 meters are placed between actuator and control valves, to simulate control valves being placed outside of the MRI bore or CT zone, 1° precision is achieved within 10 seconds in 90% of the cases.

Contents

1	Introduction	5
1.1	Research context	6
1.2	Objective	7
1.3	Outline	7
2	Background	8
2.1	Objective	8
2.2	State of the art	8
2.2.1	Computed Tomography	8
2.2.2	Magnetic Resonance Imaging	9
2.2.3	CT- and MR-compatible actuation principles	10
2.2.4	Choice of actuator	14
3	System design and modelling	17
3.1	Objective	17
3.1.1	Requirements	17
3.2	Materials & methods	19
3.2.1	Vane motor operating principle	19
3.2.2	Dynamic model	20
3.2.3	Tubes and valves model	25
3.2.4	System design	26
3.3	Results	28
3.3.1	System design	28
3.4	Discussion	31
4	System identification	32
4.1	Objective	32
4.2	Materials & methods	32
4.2.1	Torque-angle relation	32
4.2.2	Motor speed and torque	32
4.2.3	Pressure dynamics	33
4.2.4	Dynamic response	33
4.3	Results	37
4.3.1	Torque-angle relation	37
4.3.2	Motor speed and torque	39
4.3.3	Pressure in tubes	39
4.3.4	Dynamic response	41
4.4	Discussion	45

5	Controller design	47
5.1	Objective	47
5.2	Materials & methods	47
5.2.1	PID control	47
5.2.2	State feedback controller	50
5.2.3	Brake control	53
5.3	Results	53
5.4	Discussion	54
6	Performance analysis	56
6.1	Objective	56
6.2	Materials & methods	56
6.2.1	Experiments	56
6.3	Results	57
6.4	Discussion	58
7	Conclusions & outlook	61
7.1	Conclusions	61
7.2	Discussion	62
7.3	Outlook	64
	References	66

Chapter 1

Introduction

Over the past decade, the number of yearly Computed Tomography (CT) and Magnetic Resonance Imaging (MRI) scans in the Netherlands has tripled [7, 8]. Simultaneously, technological advances allowed surgery to transition from open procedures, where incisions are made in the patient's body to access various sites, to minimally invasive procedures. Laparoscopy, angioscopy, and endoscopy are some minimally invasive surgical procedures. Percutaneous surgery, where access to a patient's internal sites is provided via a needle-puncture of the skin, is minimally invasive as well. Examples of percutaneous procedures include biopsy and tissue ablation using radio-frequency (RF) or microwave (MW) radiation.

Working in an MRI- or CT-environment imposes constraints on the materials which can be used, and therefore on the type of actuation. It is desirable to make a system that is fully CT- and MR-compatible. Different terminologies exist for CT- and MRI-compatibility. CT-compatibility is defined as the system being sufficiently X-ray transparent to avoid unacceptable image distortion. For MRI, the terminologies adopted by the US FDA are as follows. A system or object is MR-safe if it poses no known hazards to the patient or environment in the form of magnetic or induced forces, torques, RF heating, or induction of voltages. The rationale behind MR-safeness of a material should be grounded in scientific reasoning instead of tests [9]. If a system is safe to use in MRI-scanners up to a certain field strength, the system is dubbed MR-conditional.

MR-safeness considers only hazards in any form to patient, radiologist, or environment. MRI-compatibility of a system considers the effect of an MR-safe device on the diagnostic image data as well. Akin to CT-compatibility, MRI-compatible systems avoid unacceptable image distortion.

Various robotic systems have been developed to improve minimally invasive surgical procedures with varying degrees of automation [1, 3]. Some of these systems use piezoelectric actuation, which makes them inherently not MRI-compatible. Pneumatic actuators in a CT or MRI context are often in the form of stepper motors. Some of these systems, such as the PneuStep [10], are complex in design. Other pneumatic stepper motors generally suffer either from low bandwidth or low resolution.

Developing a completely CT- and MRI-compatible system that can precisely and quickly position an end-effector can contribute greatly to the field of

CT/MRI-compatible robotics. This development includes: a choice of actuator type, design of a system that interfaces the actuator with its environment, and a mechanical design of the actuator with a focus on CT/MRI-compatibility.

This research focuses on picking a suitable actuator, identifying its underlying mechanics and challenges, and designing a controller with which to accurately position the actuator.

1.1 Research context

A novel system for use in a CT-scanner is the Needle Placement System (NPS) developed by DEMCON, see Figure 1.1. The NPS provides physical guidance of a needle for manual insertion in the liver and lungs. When a patient develops a liver tumour, the usual procedure is to remove it surgically. This is often done by ablating the tumour's cancerous tissue with microwave radiation. Ablation is a complex process and as such, surrounding healthy tissue is often removed as collateral damage. To minimise the damage to a patient's healthy tissue, it is imperative that the needle be placed accurately. In the traditional work flow, the physician places the patient inside a CT- or MRI-scanner to assess the tumour's location, orientation, and size. With the help of several markers which show the angle with respect to the patient's body, the physician aims and inserts the needle manually. As tumours can be quite small at around 2 cm, and situated deep inside the patient, the physician often does not place the needle correctly in one try. This leads to patient inconvenience and higher costs.

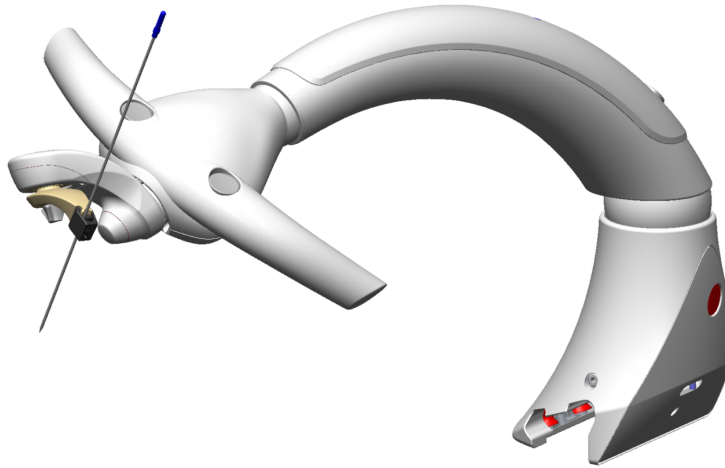


Figure 1.1: Impression of the DEMCON NPS.

The NPS, pictured in Figure 1.1, automates part of this process. It receives and interprets CT-scans automatically, plans a path for the needle to the tumour, and aligns two segments (seen in the left of the picture) with a precision of 0.01° . Positioning is achieved by two DC motors and a 1:100 transmission. The physician then places a needle inside the needle guide and manually inserts it to the correct depth.

Currently, the NPS uses electromechanical actuation at the base of the arm, which is transmitted via cables to the end-effector. Since the motors themselves are outside of the scanning zone, they produce no problems with the CT-scanning process. In an MRI-scanner, DC motors pose risks for patient and environment. The NPS is thus not MR-safe. If the NPS is to be made MRI-compatible as well, an alternative method of actuation will have to be developed that is inherently CT- and MRI-compatible.

1.2 Objective

The main objective of this research is:

To develop a servo system for precision positioning of a needle guide in a CT- or MR-environment

This objective comprises making a design of the hardware on a system level, creating a model of the system, realisation of the system in a proof-of-principle set-up, system identification, controller design, and performance assessment. A redesign of the actuator to a version made out of MR/CT-compatible materials is not made in this thesis. The first step when developing a new actuator for the NPS is to evaluate its performance. Changing the design of an actuator slightly to accommodate CT/MRI-compatible materials is not seen as an important goal of this research. A few notes on the actuator's redesign are given in Chapter 7.

1.3 Outline

This thesis is structured as follows. Chapter 2 provides background on the processes of CT- and MRI-scanning. Several methods of actuation used in a CT- and MR-environment are discussed. The chapter concludes by choosing an actuator type and reformulating the objective of this research. In Chapter 3, system design and model development are addressed. Chapter 4 deals with identification of the system designed in Chapter 3. Controller design is tackled in Chapter 5. In Chapter 6, the system's performance is evaluated and the results are fed back to the objective stated in Chapter 2. Finally, Chapter 7 evaluates the thesis' objective, and provides a discussion on the results from this thesis. The thesis concludes with an outlook on future work.

Chapter 2

Background

2.1 Objective

The objective of this chapter is to provide a background in CT- and MR-compatible actuation, and to choose an actuator for the rest of this thesis.

2.2 State of the art

In order to make an informed decision about the type of actuator to use in this thesis, it is necessary to explain more about image modalities such as CT and MRI. These processes are discussed in this section, as are materials which can be used in both methods.

2.2.1 Computed Tomography

Computed tomography (CT) provides 2D and 3D cross-sectional images reconstructed from multiple projectional images. A picture demonstrating this can be found in Figure 2.1. On one side of the patient, an X-ray source is located, which emits X-rays through the patient and into the X-ray sensors on the other side. The amount of absorption then provides information about the material in between. By spiralling the source and sensors around the patient, a full 3D image can be reconstructed. Contrast material can be injected into the patient to highlight e.g. the blood vessels.

X-ray transparency of materials Since the scan output depends on the materials that the X-rays pass through, care should be taken about which materials are present in the CT-scanner. Materials which absorb or scatter radiation should be avoided. X-ray intensity I_t after passing through a material of thickness t , with initial X-ray intensity I_0 is defined according to Beer's law:

$$I_t(t) = I_0 e^{-\mu(E)t}, \quad (2.1)$$

where $\mu(E)$, is the material-specific X-ray absorption coefficient, dependent on energy E . Beringer [11, Figure 31.16] shows that the absorption length of radio waves becomes smaller as the atomic number of the material becomes

higher. This means that materials with high atomic numbers, such as iron and copper, are undesirable in CT-scanners. Since plastics and ceramics are mostly comprised of smaller atoms such as carbon, silicon, hydrogen, and oxygen, these materials will not distort the quality of a CT-scan as much.

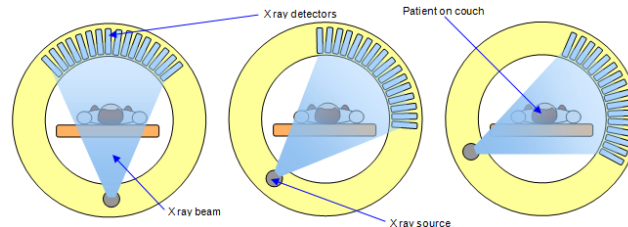


Figure 2.1: Process of a computed tomography scan [12].

2.2.2 Magnetic Resonance Imaging

Magnetic Resonance Imaging (MRI) relies on *nuclear magnetic resonance* (NMR) to create an image of a patient. NMR is the process where atomic nuclei absorb and re-emit electromagnetic radiation when exposed to a magnetic field. The resonance frequency at which this happens depends on the magnetic field properties and on the properties of the isotope that is being excited. Specifically, isotopes with an odd number of protons and neutrons have an intrinsic magnetic moment (spin), and produce NMR spectra. When an isotope with a magnetic moment, such as hydrogen or carbon, is placed in a magnetic field, its magnetic moment aligns with the field direction.

A change in field strength then causes the protons to change their alignment. When changing back to their original orientation, a magnetic flux is produced, which changes the voltage in receiving coils that process the signal.

Since the resonance frequencies vary linearly with field strength, a spatial gradient can be applied to the magnetic field strength. Based on the detected resonance frequencies, a 3D-image can be obtained showing the location and amount of e.g. water in a patient's body. This can then be used to deduce the composition of tissue in various locations.

MR-safe, -conditional, and -compatible materials Because a magnetic field is present in the MR bore, materials that are magnetic will experience a force and/or torque. Depending on the nature of this magnetic field, conductors can be displaced, risking damage to both the patient and machine. If a DC magnetic field is applied, conductors which are moving experience forces, torques and currents. If the magnetic field is AC, conductors which are at rest are affected as well.

When RF radiation is applied, conductors serve as antennae which could cause sparking. This immediately rules out all metals, and thus all forms of actuation involving magnetic or conductive materials, such as electric motors.

2.2.3 CT- and MR-compatible actuation principles

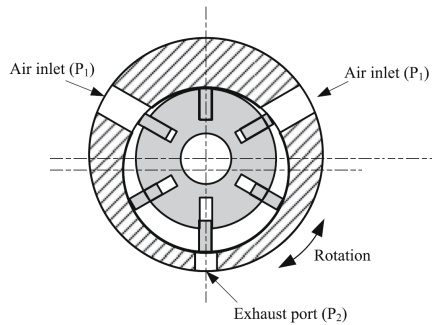
In this section, several methods of actuation that are used in a CT or MR environment are discussed. For each method, advantages and disadvantages will be given, along with approximate values for resolution, speed and power. All statements made are backed up with references either to actuators that are on the market at the time of this writing, or to a paper that shows these performances. A summary of each actuation type's strengths and potential design challenges can be found in Table 2.2. An overview of several actuators for each type, along with their performance, is found in Table 2.1.

Pneumatic actuation

Air motors have a high power-to-weight ratio, are simple in design, are self-cooling, and can be used in volatile environments where sparking could result in an explosion [4]. Air leakage poses low risks in most environments, and exhaust air can be returned to the atmosphere. Furthermore, its design allows for an MRI-compatible version by adequate material selection. Pneumatic motors have been deployed in e.g. functional MRI (fMRI) actuation [13, 14]. Pneumatic actuators can be divided in continuous and stepper motors.

Continuous motors The first type of pneumatic actuators operate as long as power is supplied. Once power supply is stopped, the motor stops moving. Theoretically, a continuous motor is able to be positioned in any configuration.

Several continuous pneumatic actuators exist, such as the vane motor (see Figure 2.2), the harmonic drive, and the piston.



(a) Working principle of a vane motor, which exist in both pneumatic and hydraulic versions [15].



(b) A pneumatic vane motor as developed by NSK.

Figure 2.2: Pneumatic motors, operation principle (left) and example motor (right).

Continuous pneumatic actuators are being used in MRI [16], though literature on the subject is scarce.

Pneumatic actuation, in the form of both continuous and stepper motors, suffers from the compressibility of air. In [17], a linear force controller was

made for a pneumatic actuator with low friction. The authors achieved a 16 Hz bandwidth for ± 2 N force changes. By using a complex non-linear model and sliding mode controller, Richer et al. increased the bandwidth to 58 Hz [18]. There is also a lot of friction present in pneumatic systems, which make them hard to control. Still, accurate positioning of a ball screw table actuated by a pneumatic vane motor was achieved in [4]. The authors report a positioning error of 6 mrad by using sliding mode control.

Stepper motors The other type of actuator is powered by alternately filling and evacuating a chamber with air, which causes a mechanism to engage and disengage accordingly. Stepper motors are capable of moving only in certain increments defined by the system's geometry. This poses a hard limit on the achievable precision. Figure 2.3 depicts a linear pneumatic stepper motor.

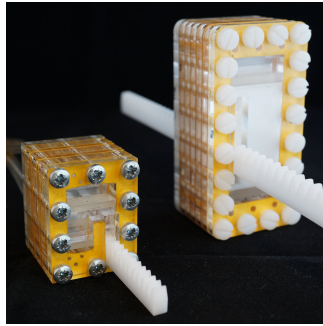


Figure 2.3: Example of an MRI-compatible pneumatic stepper motor [19].

Bandwidth is also a major challenge for stepper air motors. In [20], a bandwidth of 10 Hz was achieved for tubes of 5 meters.

Compared to continuous actuators, MRI-compatible pneumatic stepper motors are more numerous in literature. The PneuStep, developed in [10], has a resolution of 3.3° and a maximum speed of 180 rotations per minute. A more recent example are the T-63, R-80, and differently sized counterparts developed in [20]. These stepper motors are completely metal-free. The rotary motors have a resolution of 10° and a speed of 133 rpm, while the linear motors have a resolution of 1 mm and a speed of 200 mm/s.

Hydraulic actuation

Like pneumatic motors, hydraulic actuators have a high power-to-weight ratio and can be completely MR compatible since the compressor can be located outside of the MR room. This makes hydraulic actuation also a possibility for use in fMRI [2]. As with pneumatic systems, an example of a hydraulic actuator is the vane motor shown in Figure 2.2. The difference between hydraulic and pneumatic actuation is the working fluid; hydraulic motors use a liquid. Unlike air, liquids are incompressible, allowing for higher pressures, which are able to produce larger forces and torques. The bandwidth also increases compared to pneumatic motors. In [21], a hydrostatic transmission of 10 m provided a bandwidth of 20 Hz for the system. Hydraulic motors could spill fluid,

which should be assessed as a potential cause of a hazardous situation in risk management.

Hydraulic actuation was compared with pneumatic actuation in a MR-environment by Yu et al. [22]. A device was developed with one translational DOF to be used by a patient in fMRI tasks. The actuator was a piston cylinder, either hydraulic or pneumatic. The authors reported that the hydraulically actuated system achieved a higher positioning accuracy, smoother movements and a greater robustness against force disturbances than the pneumatic system. On the other hand, the pneumatically actuated piston is backdrivable and had better force control.

Electrical actuation

Two types of electrical actuators are discussed: piezoelectric motors and electroactive polymer (EAP) actuation. Their reliance on electrical and magnetic fields make them MR-conditional at best.

Piezoelectric actuation The electric dipole moment of atoms in a crystal structure varies between regions, the so-called Weiss domains. In some solids, applying a mechanical stress aligns the dipoles of these domains. This creates a change in polarization, which produces a current. This is called the *piezoelectric effect*. It is seen to occur in some synthetic ceramics and plastics, but also in natural materials such as wood and DNA.

In piezoelectric materials, the converse effect also occurs, where a deformation is created by applying an electrical field to the crystal. This is the basis of piezoelectric actuation. An example of a piezoelectric actuator is shown in Figure 2.4. The piezomotor slowly extends, then rapidly contracts, producing a force which moves the base. By accurately controlling the electrical field, very high resolutions can be achieved, even under 100 nm [23]. The large bandwidth of electrical systems is also an advantage here, meaning high speeds can be achieved. These properties make piezoelectric actuation popular among MR-conditional systems: in [24], a 9-DOF robot was designed for minimally invasive interventions in the prostate, which makes use of several ultrasonic motors to position, insert, and steer a needle. A compliant force-feedback ultrasonic actuator for wrist protocols during functional MRI was designed, built and tested in a 3 T MRI scanner [25].

Piezoelectric actuators experience a lot of friction by design, quickly leading to wear. The actuators are also non-backdrivable, so a more complex design is needed to make a system that is backdrivable. Although they are not magnetic, piezoelectric actuators produce artefacts in MRI scans, especially when turned on [27, 28]. When the actuators are covered in RF-shielding material such as aluminium, some authors report an improvement in the signal-to-noise ratio (SNR) [29]. Still, most studies are done in 1.5 - 4.0 T scanners, while magnets with field strengths up to 11 T are already developed for use in high field neuroimaging [30], making the MR-compatibility of piezomotors questionable.

Electroactive polymer actuation Electroactive polymers (EAPs) resemble piezoelectric materials in that they also deform when an electric field is applied.

EAPs however exhibit much larger strains of up to 200% and are popularly referred to as artificial muscles.

EAPs come in two types: dielectric and ionic EAPs. The first type relies on electrostatic forces which shrink the polymer in thickness and expand it in area. These EAPs have a fast response time and can provide large actuation forces. A strain can be maintained by a DC voltage. The electric field needed to deform a dielectric EAP (DEAP) is high, however, with values of 100 MV/m cited [31]. Ionic EAPs (IEAPs) provide actuation by displacing ions inside a polymer (Figure 2.5). They require an electrolyte to provide actuation. Their main advantage is that they require a relatively low operating voltage (1-3 V) and are generally bi-directional. However, the electrolyte can evaporate over time, making the material dry and decreasing performance. IEAP response times are much higher than DEAPs (100 ms vs 1 ms). For most IEAPs, it is difficult to maintain a displacement by applying a DC voltage [31].

The MR-compatibility of a DEAP was tested in [32]. It was found that an applied electrical field of 8 MV/m on the DEAP produced almost no difference in SNR in a 3 T MRI scanner. In [33], the authors produced a linear DEAP actuator which is capable of lifting a 7.5 kg mass 2.2 mm, and deflecting a spring by 2.2 mm, producing a force of 66 N. A commercial EAP-based motor has yet to be developed.

Remote actuation

It is possible to use electromagnetic actuation, as long as the actuators themselves are either not in the way of X-ray beams (in CT) or are outside the MRI bore. This means that transmissions, such as a cable/belt or rod, are needed to control the end-effector.

Electromagnetic DC motors are the actuation method of choice for most robotics applications. They have a high bandwidth, are usually backdrivable and allow for simple control of position and force. The working principle of a DC motor is shown in Figure 2.6. By alternating the current's direction inside a coil, the armature is alternately attracted and repelled by two permanent static

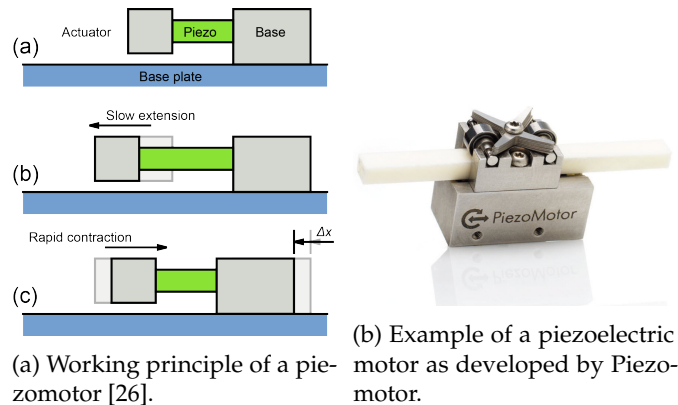


Figure 2.4: Piezoelectric motors, operation principle (left) and example motor (right).

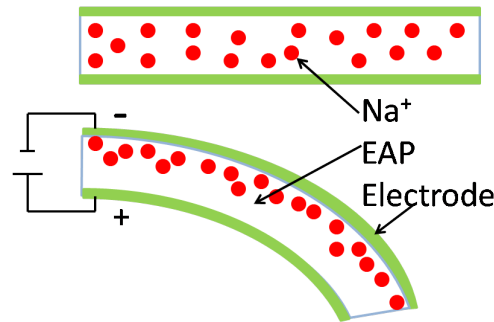
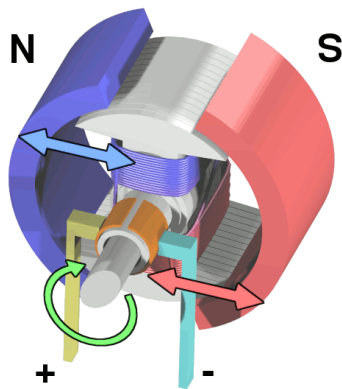


Figure 2.5: Working principle of an ionic EAP motor [34].

magnets around the armature. In a MR-environment, these motors have to be placed at a safe distance from the magnet's isocenter. Even then, shielding may be required. In [2], a DC motor was placed at a distance of 2.5m from the isocenter of a 1.5 T MR scanner. The motor was then repeatedly switched on and off, keeping track of an object's SNR in the scanner bore. This was done with and without shielding by a mu-metal. When the motor was not shielded, SNR changed heavily depending on if the motor was running or not. After applying shielding, SNR remained relatively constant throughout the experiment. Keeping in mind the increase in scanner strength, the actuators would have to be placed further away from the bore, making it difficult to design a compact and efficient electromagnetically actuated MR-safe robot.



(a) Working principle of a DC motor [35].



(b) Example of a DC motor as developed by Maxon.

Figure 2.6: DC motors, operation principle (left) and example motor (right).

2.2.4 Choice of actuator

Various actuation principles have been considered in search of a CT and/or MRI-compatible motor, and a wealth of designs has been developed. However,

Table 2.1: Overview of a selection of actuators, with specifications. Actuator specifications were found in ranges corresponding to the application in this thesis.

Actuator	Resolution	Speed	Force/Torque	Power	Volume (cm ³)
Pneumatic: continuous					
GAST 1AM-NRV-39A [36]	6 mrad [4]	10000 rpm	0.75 Nm	300 W	326
NSK EX-203c [37]	-	25000 rpm	-	-	18.8
MORITA AIR TORX [38]	-	20000 rpm	0.03 Nm	10 W	22.5
Deprag Adv. Line 67-010 [39]	-	3000 rpm	0.06 Nm	20 W	48
Deprag Adv. Line 67-030 [39]	-	1300 rpm	0.29 Nm	20 W	50.6
Modac Air motor MT05 ¹ [40]	-	2772 rpm	2 Nm	92 W	253.7
Desoutter MR16 KL-ATEX [41]	-	13000 rpm	0.15 Nm	110 W	75
Desoutter MR25 KL-ATEX [41]	-	13100 rpm	0.24 Nm	160 W	137.4
Pneumatic: stepper					
Groenhuys R-44 (stepper) [20]	10°	333 rpm	0.45 Nm	3.7 W	60
Groenhuys R-25 (stepper) [20]	6.9°	231 rpm	0.1 Nm	1.1 W	12.5
PneuStep (stepper) [10]	3.3°	180 rpm	0.6 Nm	3 W	198.6
Hydraulic					
HydroLeduc PB32.5 [42]	-	5000 rpm	0.27 Nm	22 W	122.4
HydroLeduc PB36.5 ² [42]	-	5000 rpm	0.69 Nm	57 W	268.6
HydroLeduc PB1.3 [42]	-	4000 rpm	7.6 Nm	600 W	726.4
MOOG A085 [43]	-	-	7.4 Nm	-	6135.8
Bansbach HZ16 ³ [44]	-	-	4000 N	-	28.4
Flodraulic LP-12 [45]	-	-	4.75 Nm	-	219
Flodraulic MPJ-11 [45]	-	69 rpm	36.4 Nm	42 W	210.7
Piezoelectric⁴					
PI N-412.50 [46]	300 nm	20 kHz	7 N	0.042 W	10.2
PI M-664.164 [47]	200 nm	400 mm/s	4 N	1.6 W	81
DSM I-90 [48]	< 1 μ m	30 mm/s	90 N	2.7 W	64.6
DSM I-30 [48]	< 1 μ m	30 mm/s	30 N	0.9 W	16.1
NanoMotion HR1 [23]	< 100 nm	250 mm/s	3.5 N	0.875 W	8.3
NanoMotion HR2 [23]	< 100 nm	250 mm/s	7 N	1.75 W	13.5
Electromagnetic (for comparison)					
Maxon RE-max 24 222049 [49]	-	7470 rpm	0.0525 Nm	9 W	14.5
Maxon RE-max 24 222053 [49]	-	9270 rpm	0.0519 Nm	9 W	14.5
Maxon RE-max 29 226802 [49]	-	9350 rpm	0.299 Nm	9 W	29.7
Maxon RE-max 29 226806 [49]	-	8860 rpm	0.299 Nm	9 W	29.7
Maxon DCX 22 12V [50]	-	6240 rpm	0.015 Nm	8 W	12.9
Maxon DCX 22 24V [50]	-	6384 rpm	0.014 Nm	8 W	12.9

a compact and high performance solution that lives up to the way electrical servomotors dominate the field of regular mechatronics is yet to be found.

Pneumatic continuous actuation is often dismissed due to non-linear behaviour and a high amount of friction [6]. Precise positioning using a pneumatic vane motor has however been shown feasible [5, 4]. The vane motor furthermore offers a high power-to-weight ratio, cleanliness and simple design of which an MR-compatible version has readily been developed [16]. This makes the vane motor an excellent candidate for a CT- and/or MRI-compatible actuation system.

¹ Not reversible. ² Non-backdrivable. ³ Maximum load time 10 s, 15% of duty cycle. ⁴ Powers are computed, not taken from listing.

Table 2.2: Summary of several actuation types used in an MRI-environment, with their advantages and challenges listed.

Actuation type	Advantages	Challenges
Pneumatic	◦ High power-to-weight ratio ◦ Safe in volatile environments ◦ Leakage not a safety issue ◦ Completely MR-compatible	◦ Low bandwidth ◦ Losses due to leakage ◦ Low efficiency ◦ Large amounts of friction
Continuous	◦ High speeds attainable	◦ Position control difficult
Stepper	◦ Position control simple	◦ Resolution and speed limited by geometry ◦ Higher friction than continuous motors
Hydraulic	◦ High power-to-weight ratio ◦ High stiffness ◦ Actuation principle is MR-safe ◦ High pressure allows for high torques	◦ Leakage can be an issue ◦ High inertia due to fluid ◦ High friction
Piezoelectric	◦ Extremely high resolution ◦ High bandwidth	◦ Not completely MR-safe ◦ Not backdrivable ◦ High amount of wear ◦ Low power ◦ Not X-ray transparent
Electroactive polymer: DEAP	◦ High forces possible ◦ High energy density	◦ Need a strong electric field ◦ Actuation mostly unidirectional
IEAP	◦ Fast response time (10^{-3} s) ◦ IEAP: Bi-directional ◦ IEAP: Low voltage required	◦ Not available as motor yet ◦ IEAP: Performance degrades material dries ◦ IEAP: Slow response time (10^{-1} s) ◦ Not available as motor yet ◦ IEAP: Requires electrolyte
Remote actuation	◦ Standard servomotor possible ¹⁶	◦ Reducing friction ◦ Achieving sufficient stiffness

Chapter 3

System design and modelling

3.1 Objective

The objective of this chapter is to develop a vane motor servo system with control valves and tubes in favour of precision position control. A non-linear model is constructed of the system, which is then built in the form of an experimental set-up to test several strategies for positioning. These strategies include adjustable friction torque and tube lengths. First, a list of requirements is made.

3.1.1 Requirements

Requirements can be divided into a few categories: what the system *must* achieve, what the system *should* achieve, and what it *could* achieve.

There is only one requirement that *must* be achieved, which is that the system is CT- and MRI-compatible. This means that no magnetic or conducting materials can be used. Materials with a high atomic number have to be avoided as well, as they absorb X-rays. The other requirements, *shoulds* and *coulds*, are named design goals.

Design goals are not as strict. For instance, the torque specification is not set in stone, as several alterations to the NPS can be made to accommodate a lower torque output by the vane motor. In practice, this means that an off-the-shelf actuator will be picked to do experiments on. Within this constraint, the maximum achievable performance for torque, rotational velocity, and precision will be evaluated for several scenarios. The design goals are listed below.

Precision The motor should achieve a precision of 1° . This is based on the minimum tumour size that has to be punctured, which is 5 mm in diameter. The NPS' end-effector precision is 0.01° . The end-effector is linked by a worm to the actuators with a 1:100 transmission. The actuators themselves thus achieve a precision of 1° . The chosen actuator should be able to achieve this precision as well.

Torque The motor should achieve a static torque of 7.32 Nmm and a dynamic torque of 5.5 Nmm. This is based on the torque that the worm experiences when actuating the NPS' segments.

This torque originates from several sources. To aid the explanation below, an impression of part of the NPS is shown in Figure 3.1. A simplified iconic diagram of the vane motor and NPS is shown in Figure 3.2.

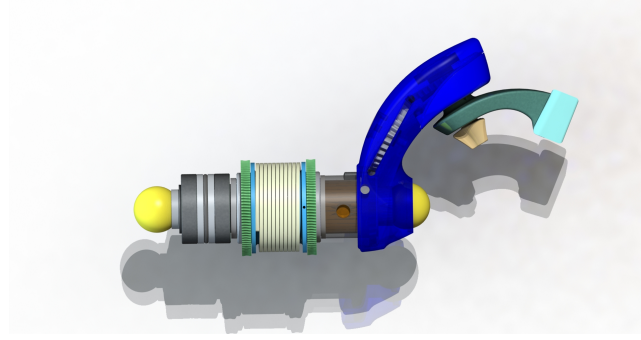


Figure 3.1: Impression of the NPS' segments and drive axis, including worm wheels (green) and pre-load torsion spring (white).

The vane motor is connected to a worm, which forms a 100:1 transmission with a worm wheel. This worm wheel is connected to a shaft which in turn rotates the NPS' segments. These segments produce a friction torque T_f on the worm wheel axis.

A pre-tensioned torsion spring (Figure 3.1) serves to keep the worm wheels aligned with the actuators' worms. The maximum torque, T_s in Figure 3.2, produced by this spring is 300 Nmm.

Finally, there is a load torque which is determined by the amount of moving mass. This torque is also after the transmission, so the equivalent inertia is $1/10000^{th}$ of the original inertia.

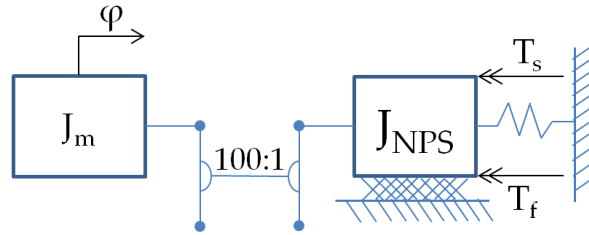


Figure 3.2: Simplified iconic diagram of vane motor and NPS. The vane motor has inertia J_m and generalised coordinate ϕ indicating its rotation. A worm is attached to the vane motor which forms a 100:1 transmission with a worm wheel, which is connected to the rest of the NPS, which has an inertia of J_{NPS} . Gear transmissions in the NPS lead to a friction torque T_f . A pre-loaded torsion spring designed to keep the worm wheel in place produces torque T_s .

The total torque was determined by actuating the worm wheel with a Maxon RE-max 24 222053+201940 motor, which has a motor constant of 24.4 Nmm/A. Movement started at a current of 0.3 A, which means the static torque equals 7.32 Nmm. Dynamic torque was found by determining the steady-state current

when the worm wheel is rotating, found to be 0.22 A. This leads to a dynamic torque of 5.5 Nmm.

Rotational velocity The motor should achieve a speed of 40 RPM. This is based on the worst-case scenario where the NPS has to perform a homing procedure of its entire range of motion (340 degrees per DOF), after which it gets a set-point of 340 degrees again. The desired positioning time is 5 seconds, which means that a rotational velocity of 2.5 rad/s is needed at the end-effector. The current design incorporates a transmission of 1:100 between end-effector and actuator, yielding a rotational velocity of 250 rad/s or 14,000 deg/s of the actuator.

3.2 Materials & methods

This section covers the vane motor's operating principle and introduces several models from literature. Using these, a novel system design is made based on arguments from a physics and control standpoint.

3.2.1 Vane motor operating principle

A vane motor's operation is not unlike that of a waterwheel. In this case, pressurized air enters a chamber that consists of two eccentrically placed cylinders, the stator and rotor. See also Figure 3.3. The chamber is divided into compartments by vanes, which are pushed against the stator by either centrifugal force, pressurized air, or springs at the base. Air enters one compartment and as the rotor rotates, the chamber is sealed and expands until it reaches the outlet (here at the bottom). Here, air is exhausted to a lower, typically atmospheric, pressure. The air which is still in the compartment is then slightly compressed as it completes its rotation.

The pressure difference produces a force on the vane between the two adjacent compartments, which constantly varies throughout one rotation. The sum of all the vane forces then produces a torque on the rotor, causing the vane motor to accelerate. Vane motors have a simple operating principle, are compact, and thrive at high rotational velocities. For this reason, they are often used in speed control applications, such as a dentist's drill. In this thesis however, the positioning accuracy of the vane motor is important.

As stated in Section 2.2.3, there are several challenges in accurate positioning of a vane motor. The first is that there is a large amount of friction present in the system, especially at the vanes' contact point with the rotor. This introduces some stick/slip-behaviour to the system which makes positioning at low speeds harder to accomplish.

Next, since there are several distinct positions in which the combined torque on the rotor is lowest, there is a possibility that the system will exhibit cogging behaviour at these positions. One such position is when a compartment has just exhausted its air to the outlet, as the work done by that chamber suddenly drops to zero.

Another challenge lies in the fact that air is compressible. Between 2 and 6 bar of air pressure, the compressibility factor of air, an indication of how much

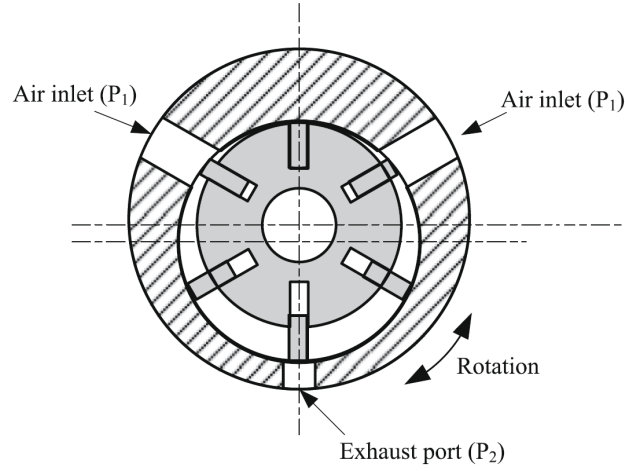


Figure 3.3: Working principle of a vane motor [15].

the gas behaves as an ideal gas, is between 0.97 and 0.99. The compressible flow can be interpreted as a stiffness between actuator and end-effector. The system should then be designed such that the influence of the effects mentioned above is as small as possible.

3.2.2 Dynamic model

This section will discuss the dynamic model of the vane motor system, which includes, aside from the vane motor itself, the tubing to and from the actuator, and the control valves. It will take several models from literature on the geometry and kinetics of the vane motor, and combine them with a model for the filling and evacuating of pressurized air tubes. The model is then implemented in Simulink to gain insight about the vane motor system's inner workings and to aid in the future design of a vane motor.

Vane motor dynamics

The vane motor model should capture the change in pressure during rotation of the rotor, the torque produced by a pressure difference between compartments, and the resulting acceleration, rotational velocity, and angle of the rotor as a result of this torque. The first step in describing the relation between pressure and torque is analysing the vane motor's geometry. Part of this model is taken from [4], where a rotational accuracy of around 1 degree was achieved. The part about work per cycle and mass flow is partly taken from [51], which is a general introduction to vane motors. No performance is mentioned in that paper.

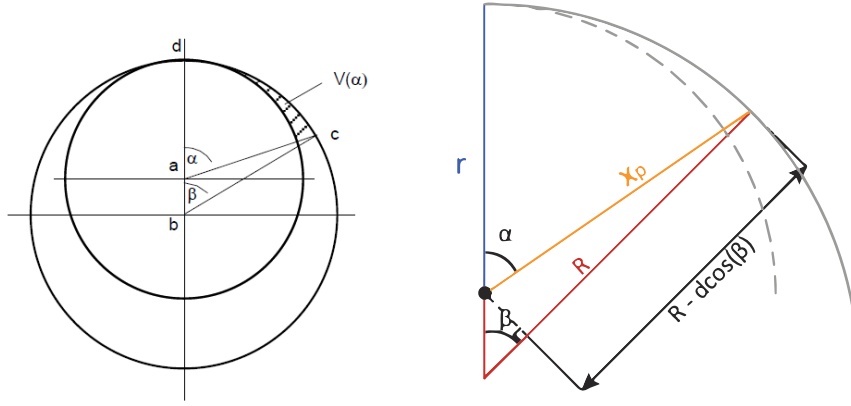
Geometry Consider a motor with N_v vanes, which has a length of L . The angle between vanes is $\gamma = \frac{2\pi}{N_v}$. Let R be the stator's radius and r the rotor's radius. The eccentricity d is defined as $R - r$. The stator and rotor angles, α and β respectively, are shown in Figure 3.4a.

Their relation is defined as [51]:

$$\beta = \alpha - \arcsin\left(\frac{d\sin(\alpha)}{R}\right) \quad (3.1)$$

The effective surface area of the vanes is a function of angle. It depends on the working radius x_p , which is defined as [4] (see Figure 3.4b)

$$x_p = \sqrt{d^2\sin^2\beta + (R - d\cos\beta)^2} \quad (3.2)$$



(a) Rotor (α) and stator (β) angles of a vane (b) Working radius x_p follows from Pythagoras' theorem.

Figure 3.4: Vane angles and working radius.

The volume V_α can be found by a subtraction of integrals and is equal to:

$$V_\alpha = \frac{L}{2}(R^2\beta - r^2\alpha - dR\sin(\beta)) \quad (3.3)$$

There are three different scenarios for the working volume V_w , i.e. the volume in one compartment:

$$V_w = \begin{cases} V_\alpha & 0 < \alpha \leq \gamma \\ V_\alpha - V_{\alpha-i\gamma} & \gamma < \alpha \leq 2\pi \\ V_{2\pi} - V_{\alpha-\gamma} & 2\pi < \alpha \leq 2\pi + \gamma \end{cases} \quad (3.4)$$

Here, $V_{\alpha-i\gamma}$, $i = 1, 2, \dots, N_v$ equates to the volume of i completely filled chambers, and $V_{2\pi}$ equals the volume of all vanes.

Figure 3.5 shows an example. Suppose one is interested in computing the volume of compartment *III*, shaded with diagonal stripes. Since $\alpha > \gamma$, working volume V_w equals:

$$V_w = V_\alpha - V_{\alpha-i\gamma} = V_{I+II+III} - V_{I+II} \quad (3.5)$$

These equations can then be used to compute the dead, filled and expanded volumes, see Figure 3.6. The ratio of expanded to filled volume determines the expansion ratio,

$$\epsilon = \frac{V_{exp}}{V_{fill}} \quad (3.6)$$

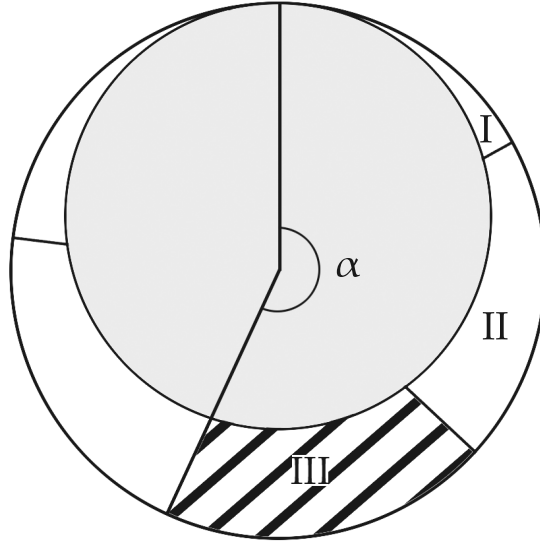


Figure 3.5: Example of chamber volumes for a vane motor with 5 vanes.

which can be interpreted as the extent to which internal energy of the incoming air is used. If $\epsilon = 1.0$, then the pressure remains constant from intake until the exhaust is reached. This means that there is no extra work done by pressure differences between compartments. A high expansion ratio, on the other hand, means that much of the internal energy is used to power the engine. While this leads to higher torques, the water in the supplied air can freeze, causing potential harm to the engine [51].

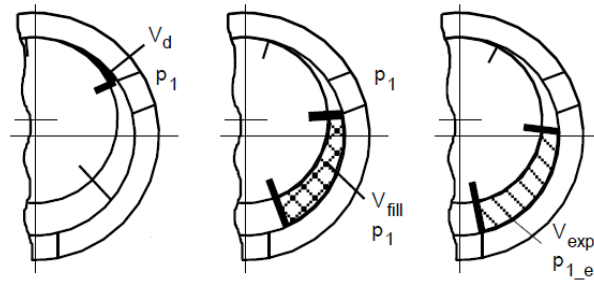


Figure 3.6: Dead, filled and expanded volumes of a compartment [51].

Work per cycle In order to compute the amount of work done per revolution, a PV diagram is constructed. It is shown in Figure 3.7.

Air enters the motor at a , with pressure P_u . As the compartment rotates, it is filled until its volume reaches V_{fill} , still at P_u . From b to c , the air expands to V_{exp} and P_{ue} , which can be calculated by assuming a polytropic process:

$$P_{ue} = P_u \left(\frac{V_{exp}}{V_{fill}} \right)^n = P_u \epsilon^n \quad (3.7)$$

where n is the polytropic index, typically taken to be 1.3 for air.

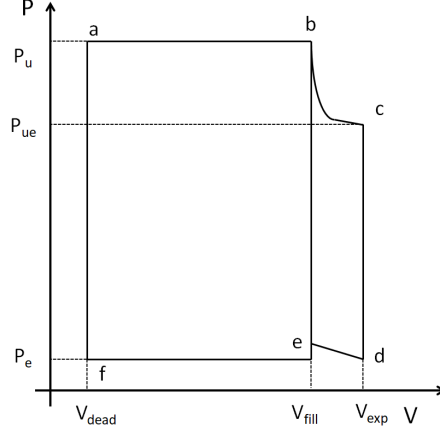


Figure 3.7: Pressure-volume diagram of the vane motor [51].

Now the air is exhausted to the surroundings, dropping its pressure to P_u . From d to e , the reverse process of $b \rightarrow c$ occurs; air, now at ambient pressure, is slightly compressed to P_{e2} :

$$P_{e2} = P_e \epsilon^{-n} \quad (3.8)$$

This compression produces negative work. At e , most of the air is released to the other port, leaving only the dead volume V_d in the compartment at state f .

The total work done by one compartment during one stroke of the engine is given by

$$\begin{aligned} W &= W_{\text{expansion}} + W_{\text{displacement}} + W_{\text{compression}} \\ &= - \int_c^b p dV - \int b^d p dV - \int_e^d p dV \\ &= \frac{P_u V_{\text{fill}}}{n-1} (\epsilon^{1-n} - 1) + (P_u - P_d)(V_{\text{fill}} - V_d) + \frac{P_d V_{\text{exp}}}{n-1} (\epsilon^{n-1} - 1) \end{aligned} \quad (3.9)$$

Mass flow to and from vane motor In a real setup, air entering the chamber does not necessarily have the same pressure as that leaving the compressor. This is due to pressure losses over long and narrow ducts. To simulate this, lumped volumes are placed at inlet and outlets. The mass and pressure in these volumes can then be computed by a differential equation and the ideal gas equation, respectively. A model is shown in Figure 3.8. Looking at volume 1, the equations are:

$$\dot{m}_1 = -\dot{m}_{\text{from}1} + \dot{m}_{\text{to}1} \quad (3.10)$$

$$P_u = \frac{m_1 R_s T_0}{V_1} \quad (3.11)$$

where R_s is the specific gas constant. With clockwise rotation, the mass flow from volume 1 can be computed as

$$\dot{m}_{from1} = \frac{P_u \omega V_{disp,nom}}{2\pi R T_0} \quad (3.12)$$

where $V_{disp,nom} = V_{fill} N_v$ is the nominal displaced volume per stroke. The incoming mass flow rate is a function of input pressure. Similarly, the mass flow rate to port 2 can be determined to be

$$\dot{m}_{to2} = \frac{p_e \omega \epsilon V_{disp,nom}}{2\pi R T_0} \quad (3.13)$$

Assuming zero losses, mass flow rate to the exhaust volume can be computed as

$$\dot{m}_{toe} = \dot{m}_{from1} - \dot{m}_{to2} \quad (3.14)$$

In a real set-up, however, additional power losses will occur due to leakage and friction.

Motor kinematics

Now that the pressure in each of the compartments is known throughout the entire cycle, the effect of the pressure difference on motor acceleration and rotation can be elucidated.

The motor will experience friction, both dry (Coulomb/Stribeck) and viscous. Applying Newton's second law to the vane motor yields

$$T - fS(\dot{\phi}) - c\dot{\phi} = J\ddot{\phi} \quad (3.15)$$

where T is the driving torque, f is the stiction coefficient, c the friction coefficient and $S(\dot{\phi})$ is defined as

$$S(\dot{\phi}) = \begin{cases} 0 & \dot{\phi} = 0 \\ \text{sign}(\dot{\phi}) & \dot{\phi} \neq 0 \end{cases} \quad (3.16)$$

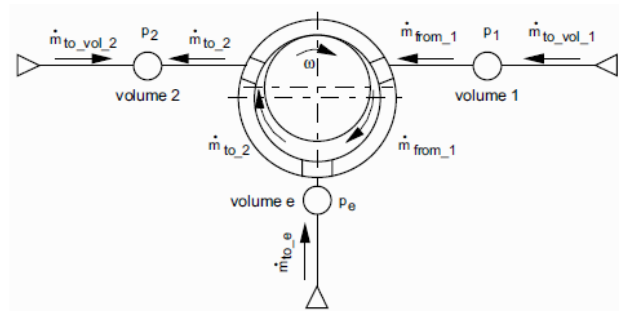


Figure 3.8: Mass-flow model of the vane motor [51].

Driving torque T is defined as the sum of torques acting on each individual vane: $T = \sum_{v=1}^{N_v} T_v$. The torque on each vane is simply pressure difference times effective vane area:

$$\begin{aligned} T_v &= (\Delta P) A_v \\ &= (P_a - P_b)(x_p - r)L(x_p - r)/2 \\ &= (P_a - P_b)(d^2 \cos(2\phi) + 2Rd \cos(\phi) + R^2 - r^2)L/2 \end{aligned} \quad (3.17)$$

where P_a and P_b are the pressures in neighbouring compartments. Including friction in the moment balance and dividing by the effective inertia, the acceleration is obtained. This can be integrated twice to produce the rotation angle of the vane motor, which can be fed back to compute the pressures inside each compartment.

3.2.3 Tubes and valves model

Tubes take time to fill and empty, which, especially at high rotational velocities or valve switching frequencies, can have a large effect on the dynamics of the whole system. The tubes produce several effects. First, flow along the surface of the tubes will be restricted, introducing a resistance. This resistance is dependent on the Reynolds number of the flow [52]. The Reynolds number is a dimensionless quantity that expresses the degree to which a flow is laminar or turbulent and is defined as

$$Re = \frac{\rho u L}{\mu}, \quad (3.18)$$

where ρ is the fluid density, u is the mean fluid velocity, μ is the dynamic viscosity of the fluid (in $Pa \cdot s$), and L is the characteristic length of the fluid container, which is the smallest relevant dimension (in this case the radius). The Reynolds number is then used to calculate the tube resistance R_t . If the flow is laminar ($Re < 2000$), $R_t = \frac{32\mu}{D_{tube}^2}$. If the flow is wholly turbulent ($Re > 4000$), then $R_t = 0.158 \frac{\mu}{D_{tube}^2} Re^{3/4}$. In the crossover region, interpolation is used.

The flow resistance is then used to compute the attenuation in mass flow, which is

$$\psi = e^{-\frac{R_t R_s T}{2P} \frac{L_t}{c}}, \quad (3.19)$$

where L_t is the length of the tube and c is the speed of sound in the fluid. The second effect of the tube length is that mass flow at the far end of the tube is delayed with respect to mass flow at the start of the tube. The delay is equal to the time it takes an acoustic wave to travel the length of the tube, which is L_t/c .

Valves The system has three valves in it: two 3/2 NC valves controlling the inflow of air, and one proportional valve at the exhaust to throttle the output flow. The inlet valves merely control the direction of airflow. Therefore, the only important characteristic is the switching time, which introduces an upper limit on the control bandwidth. The proportional valve controls the volume flow through its orifice. It is the relation between input voltage and volume flow that needs to be captured correctly. This relation is usually available from the manufacturer and can also be determined experimentally through system identification.

3.2.4 System design

To enable the vane motor to be positioned accurately, the flow of air needs to be controlled by valves. Since the vane motor needs to be bi-directional, one such valve should be able to switch between the two different inlets of the vane motor. Should the motor be positioned within the desired accuracy, the flow of air will need to be stopped entirely. This means that a valve with three outlets and two positions (a 3/2-valve) is needed, which is normally in the closed position in order to ensure that the NPS does not accidentally rotate should there be a power failure.

Next, there should be a mechanism which controls the amount of air which enters and leaves the vane motor. This can be done by throttling either the incoming or the outgoing flow. The difference between the two is the pressures which are present in the vane motor. If throttling is done on the input, air entering the vane motor will be between atmospheric and supply pressure, while the air exiting the vane motor will be at exhaust pressure. On the other hand, if throttling is done at the output, the incoming air will always be at supply pressure, while air leaving the vane motor will be between atmospheric and expanded supply pressure, which depends on motor geometry.

In position control, systems which have a low stiffness are often more difficult to control accurately than systems which have higher stiffness's. The mechanical concept of stiffness relates force to position. In bond graph theory, force is the mechanical effort variable, while position is the integral of the mechanical flow variable, speed. Translating this to the hydraulic domain, force becomes pressure, and position becomes volume. The concept of stiffness translates to the inverse of a hydraulic capacitance:

$$k = \frac{F}{x} = \frac{e}{\int f dt} = \frac{P}{V} = \frac{1}{C} \quad (3.20)$$

Therefore, a higher system stiffness is achieved if the pressure is as high as possible. For this reason, it is chosen to throttle the output flow with a valve. This valve can be either directional (binary on/off) or proportional, which can restrict the flow anywhere between fully and not at all. For a proportional valve, any type of continuous control such as PID or state feedback control can be used, while for the directional valves only a PWM signal can be transmitted. It is possible to reconstruct an analogue signal with a pulse-width modulated signal, but this requires a higher control frequency than using a proportional valve. In Figure 3.9, the system layout with a proportional valve on the exhaust can be seen.

Regarding the inlet that is not currently in use, several options exist for its pressure. One is to have the other inlet at atmospheric pressure. This however means that a torque is still generated between the exhaust and the other inlet, even if the exhaust valve is completely closed. In order to be able to bring the motor to a complete halt, the inlet that is not in use should either be at the same pressure as the exhaust or completely closed.

If it is completely closed, compression at the end of the cycle would induce additional torque. On the other hand, if the pressure is the same as at the exhaust valve, no extra (negative) work is done at the end of the cycle. This

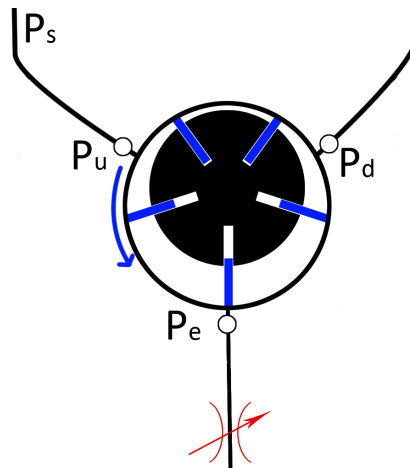


Figure 3.9: Layout of the pneumatic system with two directional inlet valves and one proportional outlet valve. P_s indicated supply pressure, P_u is the upstream pressure at the vane motor entrance, P_e is the exhaust pressure, and P_d is the downstream inlet pressure.

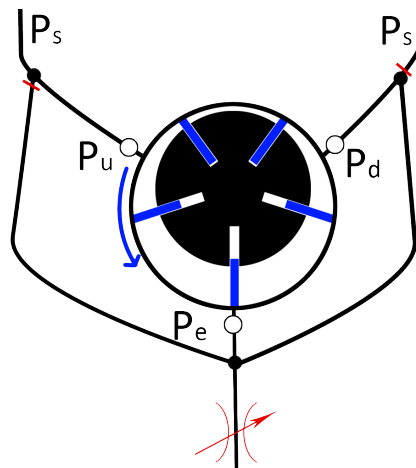


Figure 3.10: Layout of the pneumatic system with an added tube between the exhaust and downstream inlet port. Flow restriction shown with red lines.

removes negative work from the cycle and simplifies the dynamic analysis as well. The principle is illustrated in Figure 3.10.

These tubes as discussed above, along with the valves and the vane motor, comprise the system which will be analysed.

3.3 Results

3.3.1 System design

A system was designed comprising a vane motor, two binary valves, and a proportional valve (Figure 3.11).

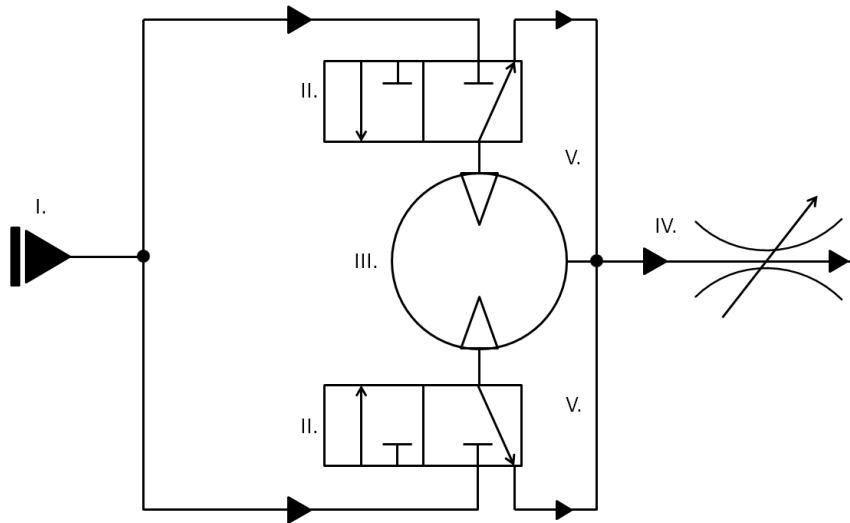


Figure 3.11: Schematic representation of the pneumatic system designed in this chapter. Triangles indicate direction of flow. (I) Compressed air enters the system. (II) Two 3/2 NC valves direct airflow towards the vane motor. (III) Vane motor. (IV) Proportional flow control valve on the exhaust. (V) Tubes from inlets to exhaust, as in Figure 3.10.

Flow starts at (I), where compressed air enters the system. This air is directed towards the two 3/2 NC directional control valves at (II), which connect the vane motor (III) either to the supply pressure, or to the exhaust pressure via (V). At (IV), a proportional flow control valve determines the mass flow and thus the pressure in the exhaust tube.

Set-up

The system design's components are translated into a lab set-up. All the components are listed here, for completeness and reproducibility. The set-up can be found in Figure 3.12.

Air supply at constant pressure (2-6 bar) enters the directional valves through tubes with an outer diameter of 4 mm, which control the direction at which the air enters the vane motor. The vane motor produces a rotation, logged by

Table 3.1: Overview of the Beckhoff components used in experiments.

Component	Function
EL1828	Connects Beckhoff system to personal computer.
EL2828	Processes directional valve signals.
EL7342	Processes encoder signals.
ES3064	Processes signals from load cell and pressure sensors.
EL4034	Generates control signal for proportional valve.
EL2024	Generates 0V or 24V signal for disc brake.
ES9505	Transformer to 5V for several components.

an Avago HEDS-9140#A encoder using a Avago HEDS-5140#A06 codewheel, which has 500 counts per rotation in quadrature.

The vane motor also delivers a torque, which is logged by a SMD 120g occlusion load cell. This load cell has a maximum pressure/force range of 120g, and is operated at 10V. The load cell produces a very small voltage on the order of several mV, so a National Instruments INA826EVM amplification board was used. This board has a variable gain which can be adjusted by adding an external resistor. For this set-up, a resistor of $82\ \Omega$ was used. The torque generated by the vane motor passes through a Prony brake, which provides an extra friction torque that can be controlled by tightening or loosening two plates of brass around the driven shaft. The friction torque delivered by this Prony brake was adjusted to reflect the torque of the worm in the current NPS setup. Static and kinetic friction were identified by using a Maxon RE-max motor and noting the currents running through the motor as the worm started rotating (static friction) and rotated at a constant speed (kinetic friction). The associated torques are 7.32 mNm and 5.37 mNm, respectively. At the end of the shaft, an aluminium wheel was fitted with an inertia of $9.5 \cdot 10^{-6}\ \text{kgm}^2$ in order to lower accelerations. In the NPS, the actuator would rotate the worm, which is part of the 1:100 transmission. The worm, which is on the actuator side of the transmission, has an inertia of around $1 \cdot 10^{-6}$.

Air then leaves the vane motor and goes to the proportional flow control valve, a Camozzi LRWA0-34-2-A-10. This valve controls volume flow through its orifice by a supply voltage, and has a maximum throughput of 610 NL/min. The exhaust of this valve is connected to the atmosphere.

All signals are processed by Beckhoff EtherCAT terminals: the components are listed in Table 3.1. The valves and vane motor will be discussed separately.

One of the challenges of using a pneumatic motor inside an MRI-scanner is that standard off-the-shelf solenoid valves can not be present in the system. There are two solutions to this. First, an MRI-compatible valve can be designed. This valve could use hydraulic or pneumatic force to open and close itself.

The other option is to place the valves outside of the MRI bore, at a distance of about 5 meters. This introduces a delay into the system. Since both the directional inlet valves and the proportional exhaust valve are at a distance of 5 meters from the bore, the delay from inlet to exhaust is doubled.

Vane motor The most important part of the system set-up, the vane motor has the highest influence on precision. The vane motor used in this research is an

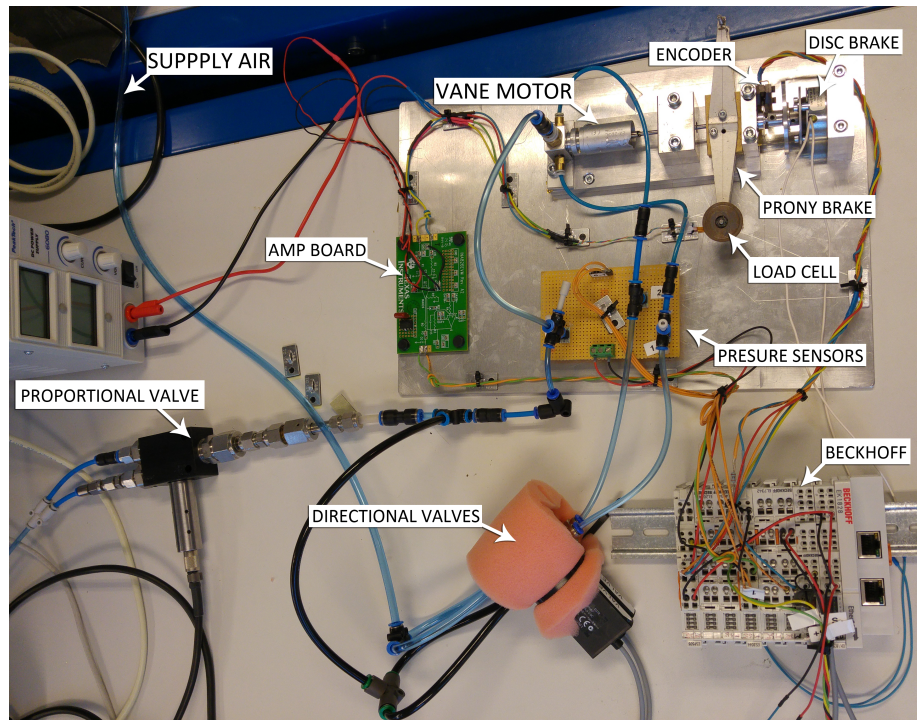


Figure 3.12: Overview of the lab set-up.

NSK EX-203c, which is developed for use in dentistry. It has a rated speed of 22000 rpm at 0.25 MPa drive pressure with an air consumption of 42 NL/min. At 0.4 MPa drive air, it has 72 NL/min air consumption and rotates at 27000 rpm. A torque specification is not listed, but motors of similar size (see also Table 2.1) produce torques of around 60 Nmm. The vane motor has five chambers.

Valves Two directional valves control the direction of incoming air flow. Both valves are Festo MHE3-MS1H-3/2G-M7-K, a 3/2 NC valve which has a response time of 3 ms and a flow rate of 200 L/min. One tube leaves the valve to the vane motor, while the other goes to the atmosphere. At the exhaust end of the vane motor there is a Camozzi LRWA0-34-A-10 3/2 proportional valve, which has a 0-10 V input and maximum flow rate of 610 L/min. At an input of 5 VDC, the valve is completely closed. At lower voltages the valve exhausts to output 1 (which is closed), while at voltages higher than 5 VDC the valve outputs – proportionally – to the atmosphere.

Disc brake A disc braking mechanism is also added to the system for control purposes. One of the controllers designed in Chapter 5 utilizes this brake to reach a setpoint by braking at the correct time. The brake used is a Huco M.1701.2121 FSB001.

3.4 Discussion

The objective of this chapter is two-fold. Firstly, to design a system of tubes and valves around the vane motor, such that precision positioning is possible (Section 3.2.4). Valves and tubes were placed such that the stiffness between control input and output is as high as possible. Extra tubes were added between the exhaust and downstream inlet in order to decrease negative work and to simplify the torque consideration. A lab set-up consisting of all the elements described has been realised (Section 3.3.1).

Secondly, a model that represents the system's inner workings was created by combining several models from literature, among others on vane motor dynamics and tube filling dynamics, into a non-linear Simulink model (Section 3.2.2). This model can be used to gain insight into the vane motor system's dynamics, and to optimise the geometry of an MR-compatible vane motor to be designed in the future. Predictions by the Simulink model and their results on the lab set-up are discussed in Section 4.3.

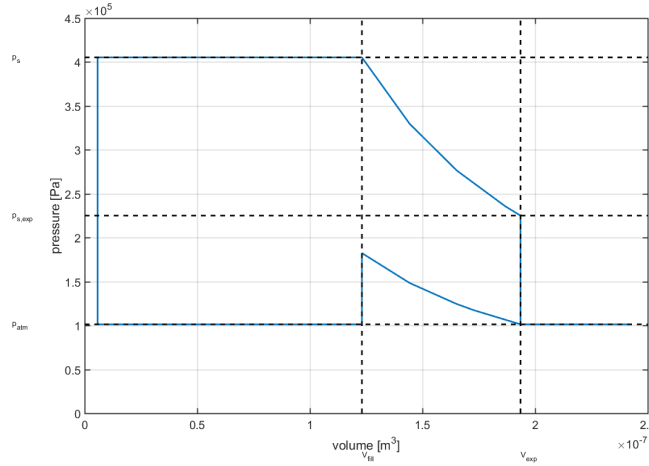


Figure 3.13: Pressure-volume diagram, from the Simulink model. The cycle starts at P_{atm} , on the bottom-left. After passing the inlet, pressure rises to P_s . The compartment expands to V_{fill} as it is closed, then undergoes a polytropic expansion to V_{exp} and $P_{s,exp}$ as the vane motor rotates. When encountering the outlet, pressure drops to P_{atm} . Finally, since the motor is symmetric, a small amount of compression occurs before the compartment reaches the downstream inlet and remaining air is exhausted.

A P-V diagram is shown in Figure 3.13 as in Figure 3.7.

The same expansion/compression dynamics are found in the Simulink model as in [51]. The extra horizontal line is due to the volume of the chamber increasing with angle, though the chamber pressure stays constant. Since it is a line, not an enclosed volume, no work is added during that part of the cycle. It should be noted that the precise shape of the P-V diagram is very dependent on motor geometry; specifically, the eccentricity d and the locations of the in- and outlets is important for the amount of expansion.

Chapter 4

System identification

4.1 Objective

The objective of this chapter is to identify the characteristics of the experiment setup (see Section 3.3.1). These include the relation between motor speed and torque, the time to fill and empty the tubes, pressures in both exhaust and downstream outlets, and the dynamic response in the form of an input/output model.

The torque generated by the vane motor depends on the difference in chamber pressures. Once such a chamber reaches the inlet or the exhaust, its pressure suddenly changes. This causes the torque to also make a discontinuous jump, which introduces a non-linearity to the system. The amount of fluctuation in the relation depends on motor geometry. Determining this relation is an objective of this chapter.

4.2 Materials & methods

This section discusses the methods used to investigate the system characteristics and the system's dynamic response. Each of these is treated in its own subsection.

4.2.1 Torque-angle relation

As explained, the motor's chambers cause non-linearities in torque. To determine this relation, the Prony brake friction is increased to above the maximum motor torque, resulting in a static situation. Supply pressure is fixed to a value of 4 bar. The motor angle is then increased stepwise, manually, and measured by the encoder. In between, the load cell force is measured. This process is repeated until the motor has experienced two full rotations.

4.2.2 Motor speed and torque

Two important parameters of any rotary motor are its torque and rotational velocity. For electric motors, they are often related as shown in Figure 4.1. This curve is characteristic of a motor's performance.

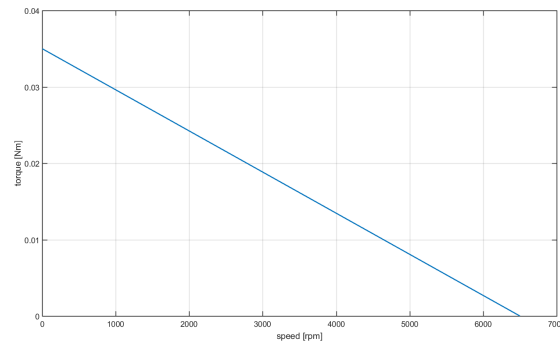


Figure 4.1: Example of a torque-speed relation for an electric motor.

A load cell is used to measure the friction torque applied by the Prony brake to the drive axle. Excess torque produced by the input pressure is converted to motor friction and acceleration. Under a constant input pressure, a (semi-) constant speed is eventually achieved.

To determine the torque-speed relation, a constant supply pressure is produced. The friction torque absorbed by the Prony brake is varied by loosening or tightening the bolts. Any excess power produced by the motor is turned into rotational motion. Each configuration of the Prony brake results in a point on the torque-speed curve. Five different configurations, resulting in five different friction torques, are used, ranging from completely tightened to completely loosened. For each configuration, five measurements are used. This is done for pressures of 3 bar and 4 bar.

4.2.3 Pressure dynamics

Tubing between vane motor and valves inevitably introduces a delay into the system. Experiments are done by putting tubes of different lengths under pressure. After a set amount of time, pressure is released by exhausting air through the inlet. The time it takes the far end of the tube to reach the same pressure as the near end of the tube is determined for tubes of 1 m, 2 m, and 5 m.

In Section 3.3.1, it was theorised that adding an extra tube between the exhaust and downstream port would simplify the torque consideration, since no extra work would be done at the final portion of the cycle when the exhaust and downstream pressures are equal. The effectiveness of this tube is determined by exciting the vane motor with a chirp signal, and measuring the pressures at both the exhaust and downstream ports. If these pressures are exactly equal, no work is done by compression between the exhaust and downstream ports.

4.2.4 Dynamic response

To gain insight in a system's open-loop dynamics, a system identification is performed. By exciting the system with an input signal possessing certain properties and measuring the corresponding output, parametric input/output models that represent the real system can be fitted, in the form of e.g. a transfer function or a state-space representation. This section discusses such models.

Tubes of 5 m long are used, in the system configuration as designed in Chapter 3. Results are compared to results with tubes of 0.7 m.

Input and output The first step is to define the input and output of the vane motor system. A block diagram of the open-loop system is shown in Figure 4.2. The vane motor itself (Figure 4.3) has pressure difference ΔP as input, and rotation ϕ as output. Since pressure difference is linked to torque, which is proportional to the second time derivative of the output, this subsystem is expected to have a transfer function of the form $\frac{1}{Js^2+ds+k}$.

The tubes introduce a delay in airflow. Flow at the tube's end is dependent on tube length, as well as air and flow properties (Section 3.2.3). As pressure (difference) is the input for the vane motor, volume flow needs to be integrated. In the frequency domain, this integration acts as a low-pass filter.

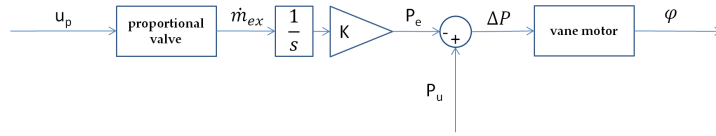


Figure 4.2: Block diagram of open-loop vane motor system. Input voltage to proportional valve, u_p , being converted to a mass flow at the exhaust. This mass flow is integrated to determine exhaust pressure. The difference between this pressure and supply pressure produces a torque, which the vane motor converts to rotation ϕ .

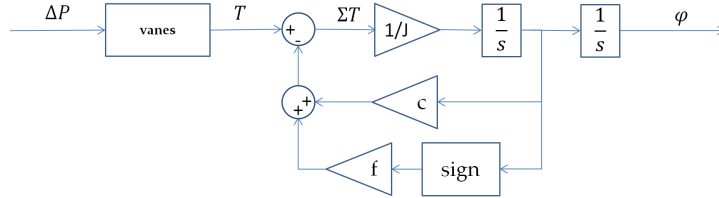


Figure 4.3: Block diagram of the vane motor. Pressure difference ΔP is converted into torque by the chambers (see Section 3.2.2). This torque is then summed with friction and stiction torques, and divided by inertia to produce acceleration. This acceleration is then integrated twice into angle ϕ .

Regarding the valves, a transfer function of input voltage u_p to vane motor exhaust pressure P_{ex} is needed. From the datasheet supplied by manufacturer Camozzi, a graph is available showing a third-order relation between voltage and volume flow. Since the objective is to control the pressure at the outlet of the vane motor, an identification will be done on the transfer function $G(s) = \frac{\phi(s)}{u_p(s)}$. The directional valves where air passes through before entering the vane motor are only to switch direction of the air flow. The relevant dynamics at play here are in the switching time, which at 3 ms is probably faster than the system bandwidth.

Input voltage is measured by Beckhoff itself, which sends the command signal to the proportional valve, and is between 5V and 10 V. Rotation ϕ is measured by the encoder (see also Figure 3.12).

Input signals Regarding the input signal, there are several options: multi-sine, frequency sweep (chirp), and PRBS. In a multi-sine, harmonics of several different frequencies are used to excite the system (Figure 4.4). This input signal is suited to capture the response at certain frequencies, such as a (anti-)resonance. However, since this only portrays the response at certain frequencies, its use for representing a complete transfer function is limited.

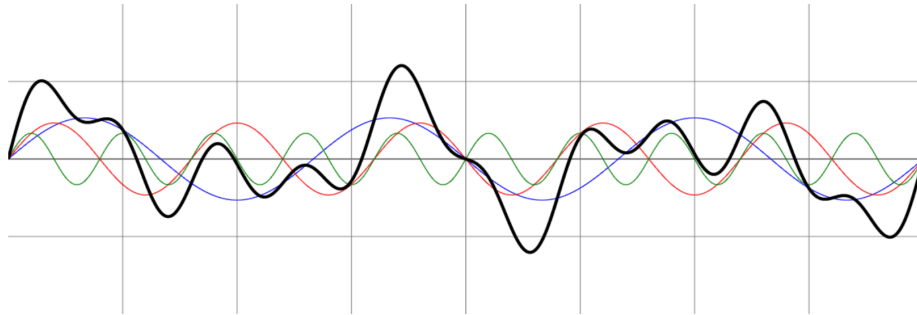


Figure 4.4: Example of a multi-sine. The red, green, and blue harmonic sine waves are summed to produce an input signal which has frequency content at three distinct frequencies [53].

Another popular choice of input signal is a frequency sweep, or chirp signal (Figure 4.5). This signal excites a continuous band in the frequency spectrum, and so provides the response for this entire band. It can be used to get the system's transfer function, provided all the relevant frequencies in the system are excited.

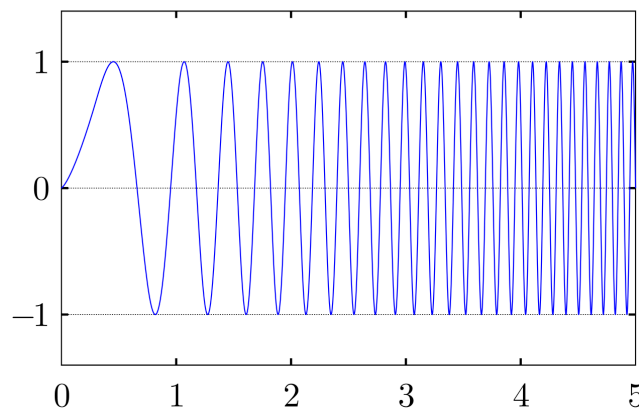


Figure 4.5: Example of a chirp signal, where the frequency of the sine wave increases over time [54]

Finally, a Pseudo-Random Binary Signal (PRBS) is another type of input signal that is often seen in system identification. The PRBS switches between -1 and +1 in a (pseudo-) random fashion. A PRBS can be generated in multiple ways. MATLAB does this by using bit-shift operations, which produces a perfectly flat spectrum over the specified frequency range.

Both the chirp signal and the PRBS are used in this thesis, as they excite a broad spectrum of frequencies. In order to produce a consistent result, identification using a chirp signal is done as follows. Ten consecutive chirp signals are run with a small rest period of five seconds in between. Afterwards, the first two chirps are discarded to remove any transient effects. The other eight are averaged out to produce the final input/output data.

Sample time The sample time should be taken so that at least all of the relevant dynamics are captured. From the Shannon-Nyquist sampling theorem, the sample time should be at least twice as high as the highest frequency to be measured. A lower limit for sample time stems from the type of state-space models that can be identified. In some model types, such as Auto-Regressive model with exogenous input (ARX), the resulting model is determined by minimising error between real and predicted response at all frequencies [55]. If a high sampling rate is taken, the weighting will take into account high frequencies as well, where noise dominates. This can give a skewed representation of the system. This can be avoided by digital filters, which can reduce the sampling frequencies for identification.

To acquire the data a high sampling rate of 1 kHz is used, which is also the upper sampling frequency limit in Beckhoff. In order to lessen the focus on high-frequency behaviour, the signals are resampled to 500 Hz.

State-space models One representation of a Linear Time-Invariant system is a state-space model. In continuous-time, a state-space representation relates the states, $\mathbf{x}$, of a system to their time derivatives by a set of first-order differential equations. A discrete-time implementation follows the following equations:

$$\mathbf{x}_{k+1} = A\mathbf{x}_k + B\mathbf{u}_k \quad (4.1)$$

$$y_k = C\mathbf{x}_k + D\mathbf{u}_k, \quad (4.2)$$

where A is the system matrix, B is the input matrix, $\mathbf{u}_k$ is the current control input, C is the output matrix, y_k is the measured output signal and D (often empty) is the feedthrough matrix.

In order to identify a state-space model, one needs only to select a system's order, which is equal to the dimensionality of $\mathbf{x}$. This means the system does not need explicit definitions for the model equations, as the prediction error methods, discussed below, do. Once the order is set, the state-space parameters are optimised and one identified system is produced. This means that nuance in the delay of the system or the amount of zeroes cannot be explicitly defined.

Prediction error methods More distinction in system models can be applied by the so-called prediction error (PE) methods [55]. These methods attempt to fit a transfer function directly to the input/output data. While state-space

representations can be transformed to a transfer function, the result is not always optimal.

In PE methods, the output y is related to the input u and error/noise e :

$$y(t) = G(z)u(t) + H(z)e(t), \quad (4.3)$$

for some transfer functions G, H . These polynomials can be split into numerator and denominator, and the way this is parametrised determines the model type. For instance, in an AR(MA)X model, both $G(z)$ and $H(z)$ share the same denominator, which means that the noise model follows the same dynamics as the system model. In ARMAX models, $H(z)$ has its own numerator polynomial, which allows for some more nuance in identification. This compared to ARX models, where $H(z)$'s numerator is equal to 1. Output Error (OE) models, finally, follow the equation $y(t) = \frac{B(z^{-1})}{F(z^{-1})}u(t) + e(t)$, which assumes $H(z) = 1$, a white noise model.

The downside of PE methods is that a model structure must be defined beforehand. When using a structure that does not represent the system accurately, these methods can give misleading results.

Both state-space models and PE models are used in this thesis. For both type, input/output models of varying complexities are generated. For each fitted model, residual analysis is performed, which relates the expected model output to the real data and is a way to show the goodness-of-fit of a model.

4.3 Results

4.3.1 Torque-angle relation

Results on the non-linear torque-angle relation are discussed from both a modelling and an experiment point of view.

Model results In Figure 4.6, the modelled relation between rotation angle and pressure or torque is illustrated. In the top figure, the same expansion/compression dynamics can be seen as in Figures 3.13 and 3.7. In the bottom figure, the sum of all vane torques (dashed line) can be seen to vary significantly with angle. This can be explained as follows. The highest difference in chamber pressures occurs whenever one chamber (A) reaches the exhaust. The previous chamber (B) will then have a pressure slightly lower than supply pressure (depending on geometry), thus producing a high torque. After this, chamber B expands, producing the negative slope in torque. Then the pressure in chamber B also drops to exhaust pressure, which explains the steep drop in torque.

Again, changing the angles of the in- and outlets, as well as the number of vanes and the radii of rotor and stator have an impact on the energy delivered per stroke and on the torque profile. For instance, if inlet and outlet are closer together, then the expansion of air will be smaller and in the extreme case, the pressure-angle plot will have a block profile.

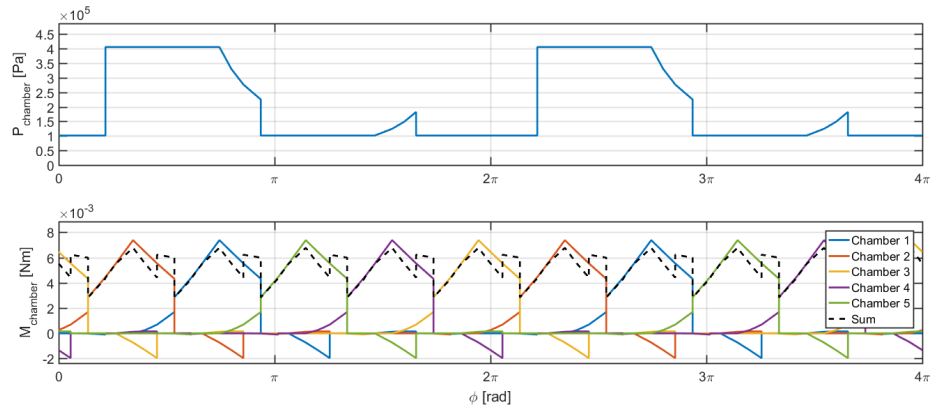


Figure 4.6: Relation between angle and pressure (top) or torque (bottom) as modelled in Simulink.

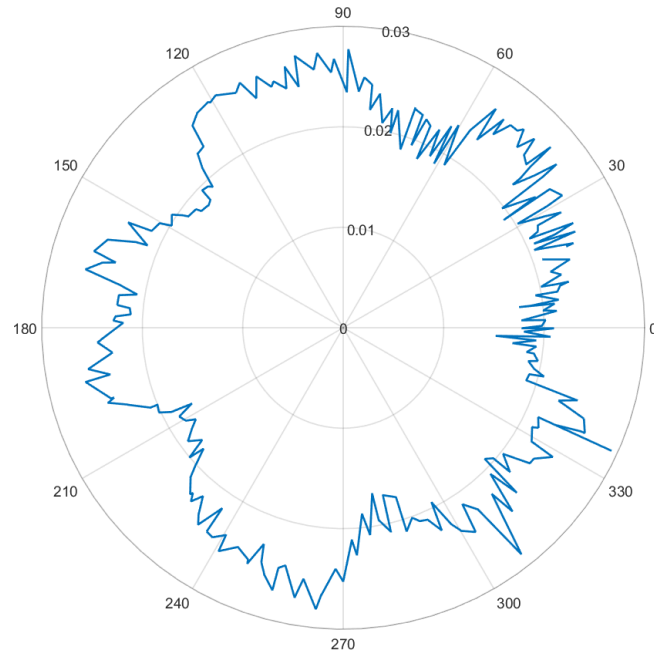


Figure 4.7: Torque experienced by the Prony brake as a function of angle (blue). Radial axis shows torque in Nm, azimuthal axis shows angle in degrees.

Experimental results The relation between torque and angle is determined as described in Section 4.2. The results are plotted in Figure 4.7. It can clearly be seen that at certain intervals, the torque drops sharply. This happens every 72 degrees, which corresponds to the angle between two adjacent vanes ($360^\circ/5 = 72^\circ$). Two full rotations are shown, with the angles transformed to the interval $[0^\circ, 360^\circ]$.

4.3.2 Motor speed and torque

Torque is plotted against speed in Figure 4.8. A curve with a shape similar to that in Figure 4.1 can be seen, although the experimental data is quite scattered.

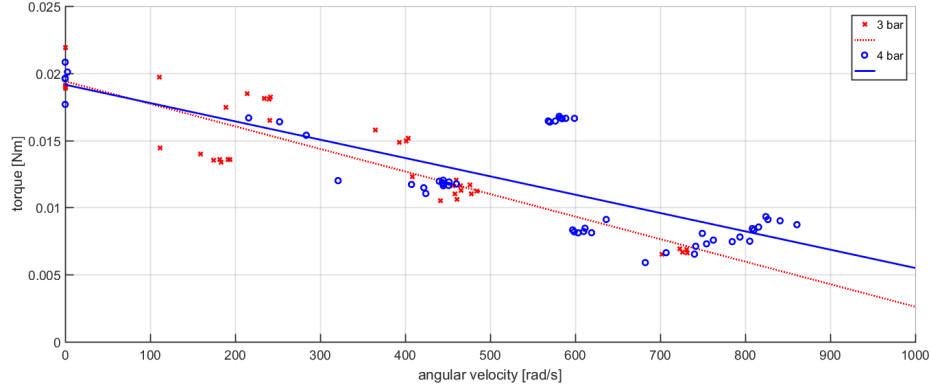


Figure 4.8: Torque as a function of angular velocity, for pressures of 3 and 4 bar. Markers indicate experiment results, lines a linear fit.

4.3.3 Pressure in tubes

Time to fill and evacuate the tubes is modelled in Section 3.2.2 as an attenuated, time-delayed response to an initial mass flow.

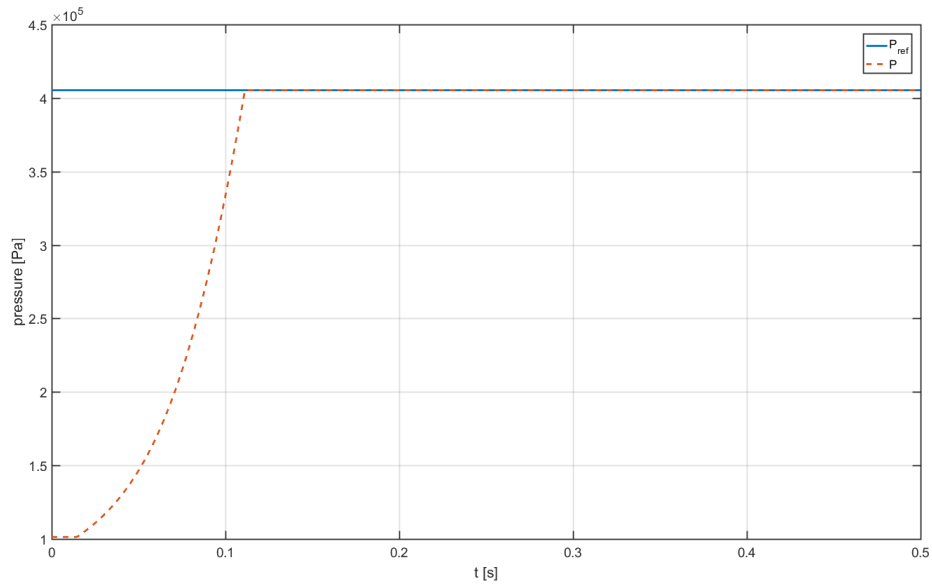


Figure 4.9: Reference and actual tube pressure as a function of time. $L_t = 5m$, $d_t = 0.004m$, $\dot{\phi} = 100rad/s$.

In Figure 4.9, the results of a tube filling simulation can be seen. It can be seen that for a length of 5 m and a rotational speed of 100 rad/s, it takes around

0.1 seconds for the end of the tube to reach reference pressure.

Results for lengths of 1 m, 2 m, and 5 m, and for pressures of approximately 2 bar and 4 bar, are shown in Figure 4.10.

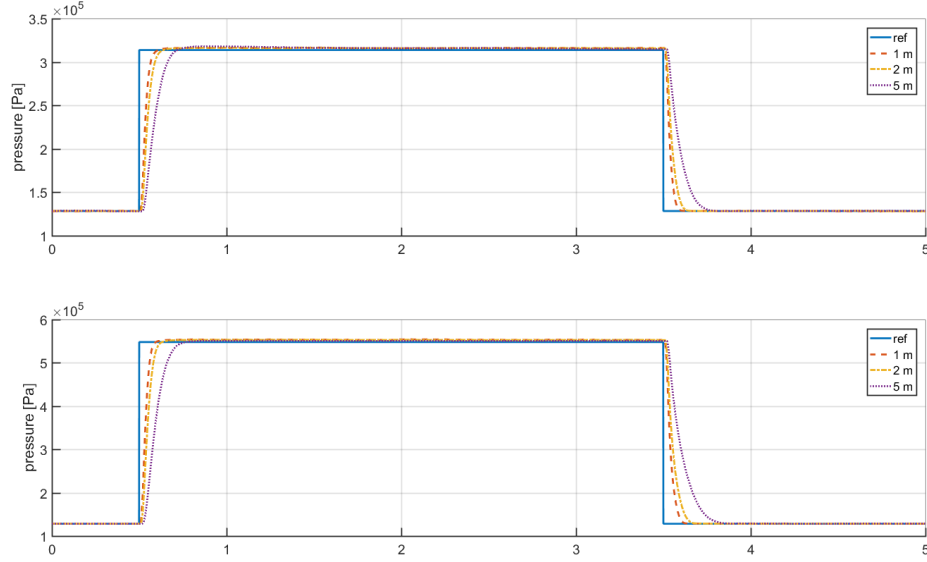


Figure 4.10: Pressure build-up over time. Blue, straight line indicates reference pressure. Orange dashed line shows response for a tube length of 1 m. Yellow dash-dot line indicates a tube length of 2 m. Purple dotted line shows response for a tube length of 5 m. In the top figure, a reference pressure of approximately 2 bar was given. In the bottom figure, this pressure is approximately 4 bar.

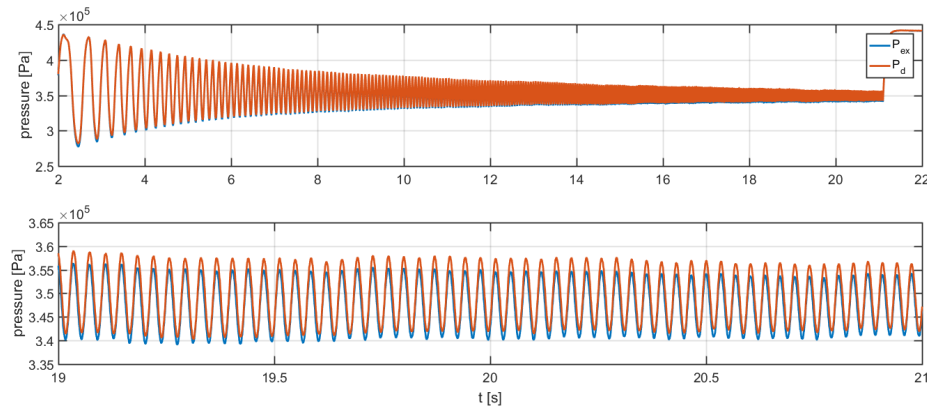


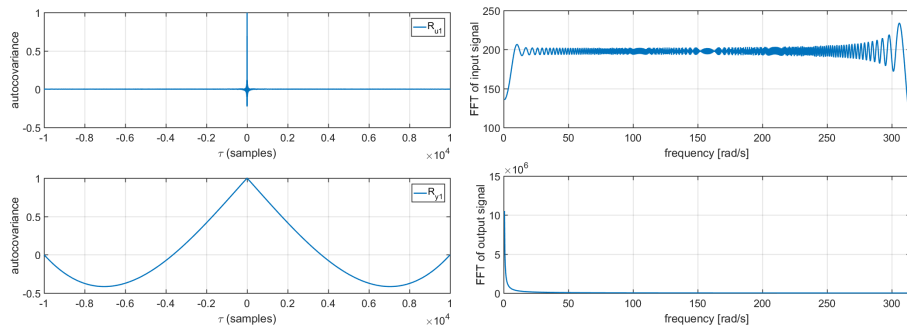
Figure 4.11: Exhaust and downstream pressures during a chirp input signal (0.01 - 30 Hz). Bottom plot is a zoomed-in portion of the top plot.

In Figure 4.11, pressures in both the exhaust and downstream inlet are plotted during a chirp signal that ranges from 0.01 Hz to 300 Hz. The bottom plot is a zoomed-in version of the top plot. It can be seen that, though the

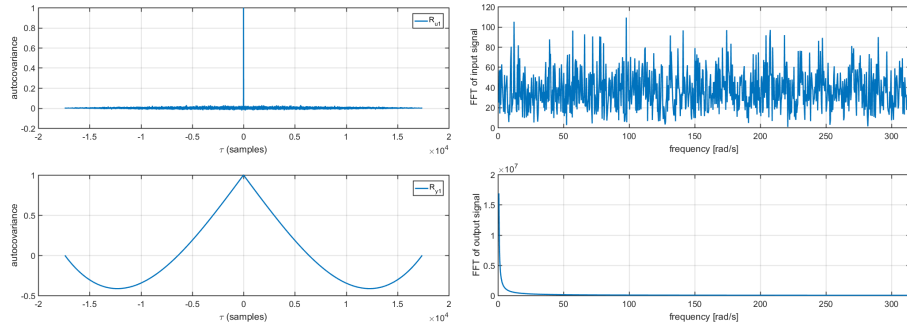
pressures are not entirely equal, they are within 1% gauge pressure of each other.

4.3.4 Dynamic response

Input signals First, the whiteness of the signals is investigated. This is done by making an auto-covariance plot, as seen in Figures 4.12a and 4.12c. For a white signal, there is no auto-correlation with other samples. If there were, this would indicate some relation between one sample and the others. As can be seen, both input signals are reasonably white. The output signals exhibit high auto-correlation, indicating a hidden dependency. This could also be an indicator of non-linearities in the system.



(a) Auto-covariance of the chirp signal. (b) Frequency content of the chirp signal.



(c) Auto-covariance of the PRBS. (d) Frequency content of the PRBS.

Figure 4.12: Analysis of chirp (top) and PRBS (bottom) input signals.

In Figures 4.12b and 4.12d, the frequency content of both input and output signals is shown. When exciting a system that is linear, one would expect that an input signal of e.g. 5 Hz creates a response at 5 Hz as well. By looking at the figures, it does not seem that frequencies that are not excited by the input are present in the output.

Since the frequency content of the PRBS is not consistent, i.e. the signal is not white, the chirp signal is used as an input signal for the remainder of this chapter.

State-space models Three state-space models are identified: one 2nd order, one 5th order and one 12th order. The second order system provides a simple approximation which can be compared to a standard mass-spring-damper system. The fifth order system shows a significantly better fit than a fourth order system, and the twelfth order system was chosen to see if more complexity introduces a better fit.

The Bode plots of all three systems are shown in Figure 4.13. The fifth and twelfth order models are shown to capture a resonance at around 150 rad/s and an anti-resonance at 250 rad/s that the second order model does not. The low frequencies of the (anti-)resonance suggest that they are caused by the air-induced stiffness.

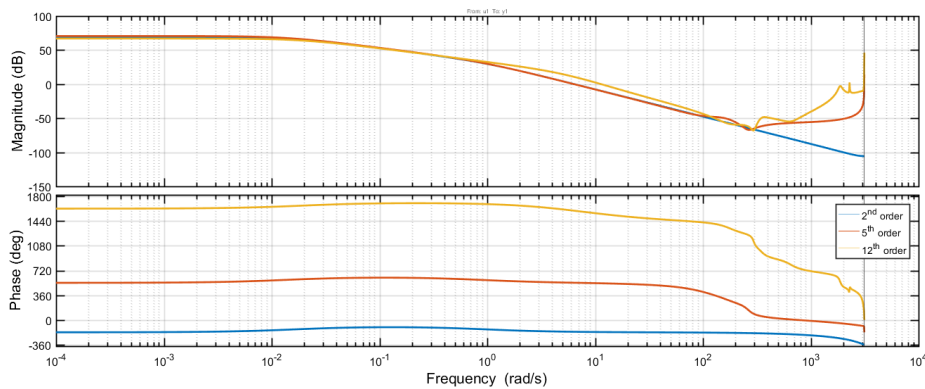


Figure 4.13: Bode plots of the identified state-space models.

The residuals are plotted in Figure 4.14. As can be seen on the left side in the auto-correlation plot, only the twelfth-order system performs has acceptable residuals.

Prediction error methods Several ARX, ARMAX, and OE models were made with varying complexities. Their Bode plots are shown in Figure 4.15 and the residuals of the ARMAX models are shown in Figure 4.16. It can be seen that the residuals are out of bounds for even the most complex model.

Comparison with short tubes As a comparison, a Bode plot as in Figure 4.13 is constructed for tubes of 0.7 m to compare responses. The result is shown in Figure 4.17. It can be seen that the bandwidth for shorter tubes is higher and the anti-resonance near 250 rad/s in Figure 4.13 is not present.

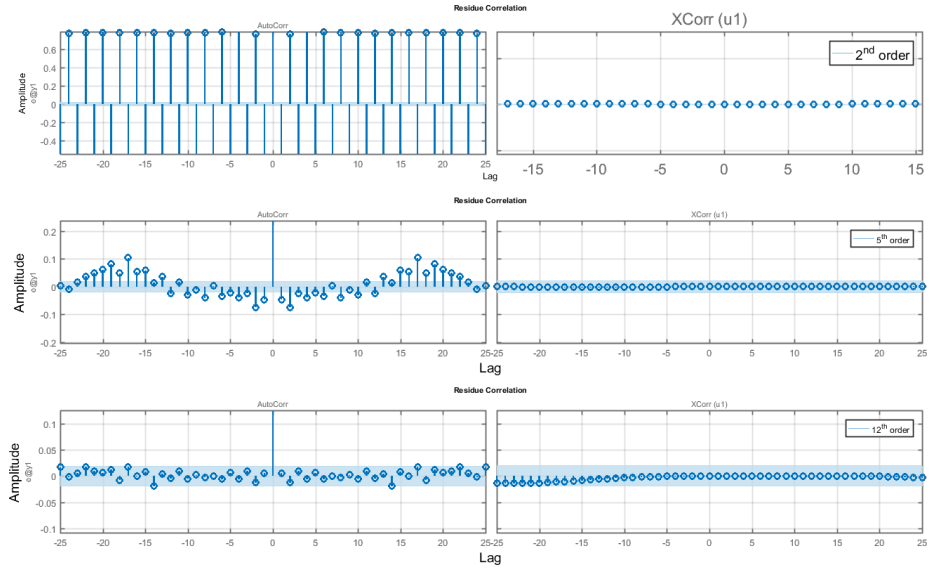


Figure 4.14: Residuals plot of the identified state-space models.

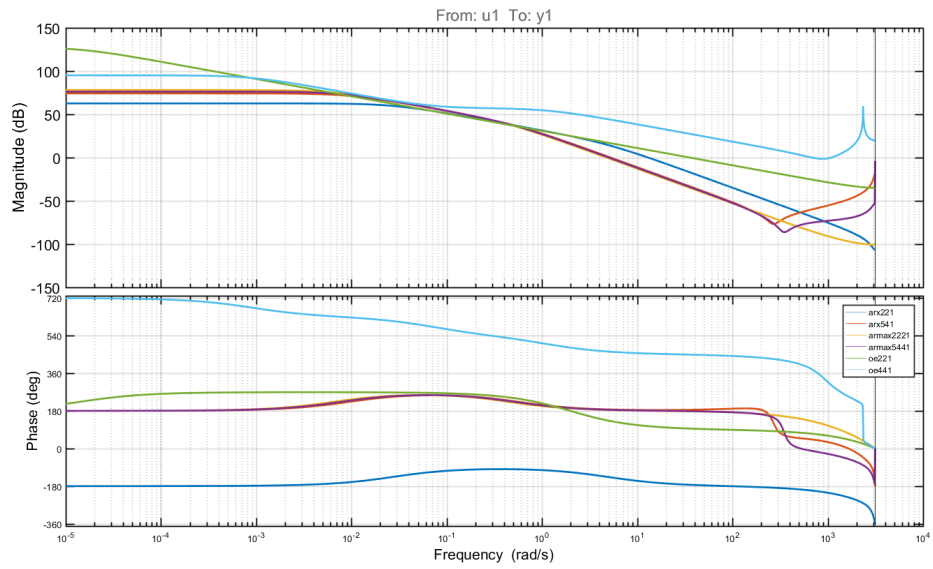


Figure 4.15: Bode plots of the identified state-space models.

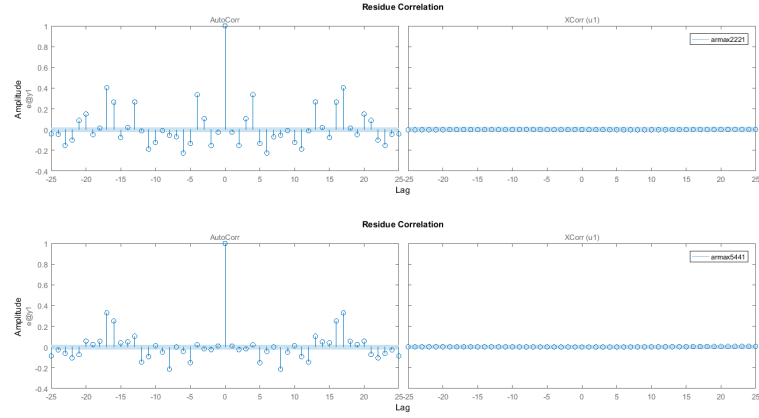


Figure 4.16: Residuals plot of the identified state-space models.

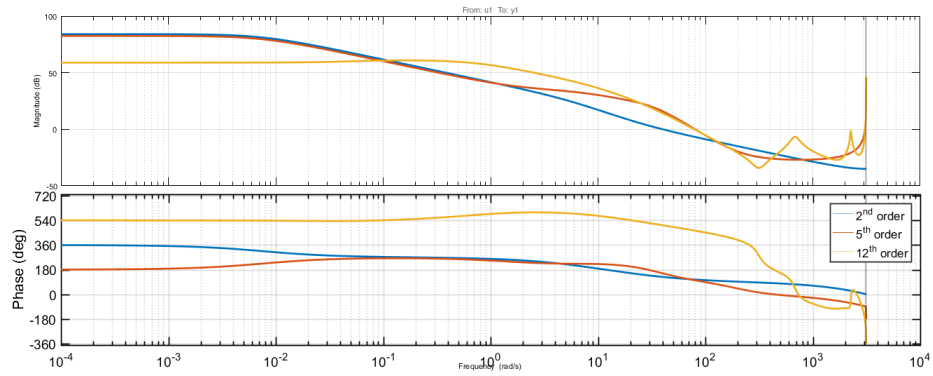


Figure 4.17: Bode plots of the identified state-space models, where tubes of 0.7 m are used between actuator and control valves.

4.4 Discussion

Torque-angle and torque-speed relations While the torque in Figure 4.7 can be seen to drop every 72° as predicted by the model in Figure 4.6, the data itself exhibits noisiness. The load cell converts displacement of the sensor into a force

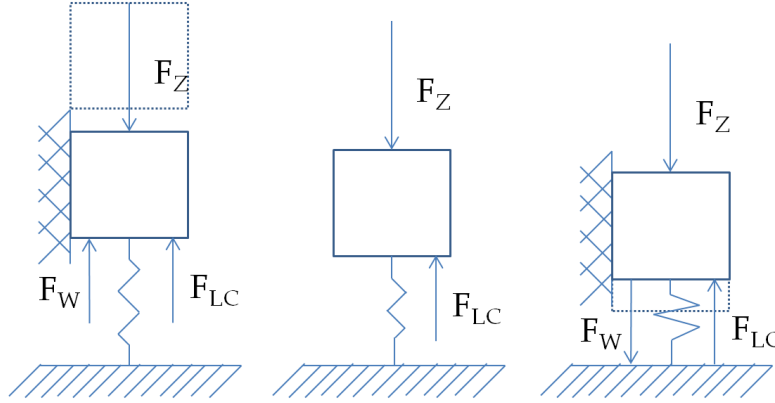


Figure 4.18: Left: Prony brake mass approaching load cell from above. Middle: Prony brake mass in ideal situation where no friction is present. Right: Prony brake mass approaching load cell equilibrium from below, i.e. from being pressed too far down.

via a known stiffness. Therefore, it acts like a spring. The Prony brake produces static friction. Looking at Figure 4.18, it can be seen that if the Prony brake approaches from above, the friction force acts upwards. Since the forces must be in static equilibrium, $F_Z = F_W + F_{LC}$. This results in an equilibrium position that is slightly higher than in the case where there is no friction (and $F_Z = F_{LC}$). Similarly, if the Prony brake is pressed down excessively, friction force acts in the direction of F_Z : $F_Z = F_{LC} - F_W$. This results in a lower equilibrium position. The difference in displacement $\Delta x = F_W / k_{LC}$. This difference in force can result in measurements being slightly offset.

Considering the torque-speed data in Figure 4.8, the same explanation holds as above. Furthermore, since the area of contact between the Prony brake and the vane motor shaft is not perfectly round or rigid, fluctuations in torque occur during measurements. This results in the high deviation in experimental data in Figure 4.8.

Pressure dynamics From Figure 4.11, it can be seen that the time it takes to fill and evacuate a tube increases with tube length. For a tube of 5 meters, around 0.12 seconds is needed before the end pressure has reached 90% of reference pressure. This imposes a limit on the control frequency, which will be addressed in Chapter 5. The time constants are in line with those in [20]. The reference pressure does not seem to have an influence on settling time.

Dynamic response Looking at the residuals in Figures 4.14 and 4.16, it can be seen that almost all residuals are out of bounds, except for the twelfth-order state-space model. This can be explained by the system exhibiting non-linear behaviour, e.g. in stick-slip dynamics and the torque-pressure relation.

Residuals of higher-order models are lower than those of the less complex models. This can be explained by the fact that extra free parameters can always be used to explain a part of the data better. Residuals for OE and ARX models are higher than those of state-space models. This makes sense, since they require more explicit model equations.

Chapter 5

Controller design

5.1 Objective

The objective of this chapter is to design a controller for precise positioning of vane motors.

5.2 Materials & methods

In literature, many control methods for pneumatic systems are presented, from simple PID controllers [4] to non-linear sliding mode controllers [4] and neural network approaches [56]. In this thesis, three controllers are developed: PID, state feedback, and brake. The state feedback controller is only discussed and designed, not implemented.

5.2.1 PID control

"Based on a survey of over eleven thousand controllers in the refining, chemicals and pulp and paper industries, 97% of regulatory controllers utilize PID feedback."

This quote by Desborough Honeywell illustrates that PID control has become a staple in industry. The strength of PID control lies in its simplicity and flexibility. A schematic diagram showing the working principle of a PID controller is shown in Figure 5.1.

The controller output (CO) is based on a factor proportional to the error (P), its integral (I) and its derivative (D). This error is the difference between a reference signal and a process variable (PV). In theory, proportional action is enough to position a system with zero stiffness. In practice, a proportional action alone results in oscillation around an equilibrium. Derivative action is used to add damping to the system, such that this oscillation is reduced over time. In systems with friction, this often equates to the PV coming to a standstill near the setpoint. Integral action causes a small error in PV to accumulate, leading to a CO that increases over time. By using integral action, steady-state error can be reduced.

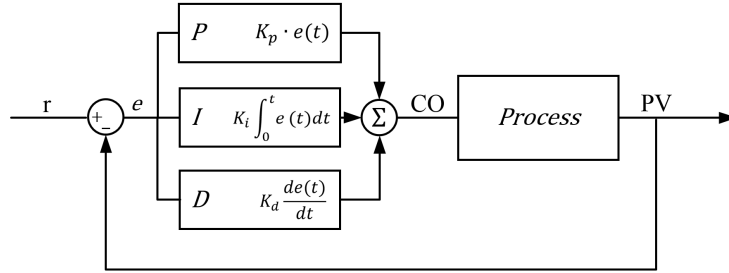


Figure 5.1: Diagram of a PID controller.

A PID controller can be written as:

$$K(t) = K_p e + K_I \int e dt + K_D \frac{de}{dt} \quad (5.1)$$

where e represents the error, $PV - r$. A PID controller's transfer function (including a low-pass filter instead of a pure derivative action) is:

$$K(s) = \frac{s\tau_i + 1}{s\tau_i} \frac{s\tau_z + 1}{s\tau_p + 1} \quad (5.2)$$

Though other parametrisations exist, the order of both numerator and denominator is characteristic of the PID controller.

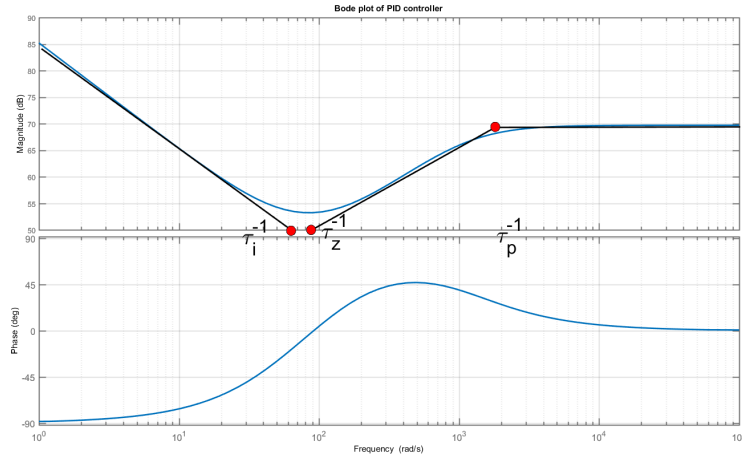


Figure 5.2: Frequency response of a PID controller annotated with the tuning parameters.

Figure 5.2 shows the frequency response of a generic PID controller. The points $1/\tau_i$, $1/\tau_p$, $1/\tau_z$ indicate a change in slope of the frequency response. These values can be tuned: for example, more phase lead can be added by increasing the distance between τ_z and τ_p .

Many techniques exist for selecting appropriate values for proportional, integral, and derivative gains in a PID-controller. Some of these, such as the Ziegler-Nichols method, rely on heuristics and observations to tune the control

parameters. Other methods, such as the 'van Dijk' method, use a (simplified) model of the system in order to determine the parameters [57]. In this thesis, the Ziegler-Nichols method will be used.

Ziegler-Nichols tuning Ziegler-Nichols tuning is done by first keeping the integral and derivative gains K_I and K_D at zero and giving the system a step input. The proportional gain K_P is increased until the controller output overcomes static friction and the system starts moving. As proportional action is equivalent to a spring, the controller causes the system to oscillate. Dynamic friction induces damping of the oscillation and eventually the system comes to a halt again. The value of K_P which causes a stable oscillation is referred to as the "ultimate gain" K_u , and the period of this oscillation is called T_u .

Whenever friction is present, the oscillation will eventually damp out. If the time it takes before the oscillation is damped out is large, it will appear to be stable. There is no optimum point where the oscillation is deemed stable, making Ziegler-Nichols tuning an approximate method.

Implementation In the case of the vane motor system, the PID control output acts on the proportional valve, which works at input voltages ranging from 5 V to 10 V. This valve merely decides the intensity of the motion, not its direction; this is controlled by the directional valves at the inlet.

PID controller values are determined by increasing proportional gain of the control signal, which is sent to the valves. The PV, vane motor angle, is measured by an encoder (Section 3.3.1). Once the angle appears to be in stable oscillation, the ultimate gain K_u is found. The period T_u is determined by applying a Fast Fourier Transform (FFT) to the PV.

Some small alterations to the standard PID controller are made. The first is in the integral action. When using a normal I-action with slow-responding systems, the integral action tends to accumulate heavily over time. This can result in the controller output saturating, which means the system will overshoot its target, after which the CO will saturate again. This phenomenon is called integral windup.

In order to counter-act this, anti-integral-windup is added to the controller. Any time the integral action would cause the CO to increase beyond the saturation point, the CO is set to this saturation point instead. Then, the value of the I-action is back-calculated to be equal to the saturated output minus the P- and D-actions. In formulas, this looks as follows.

The standard formula for a PID controller is given in Equation 5.1. If the integral action $K_I \int e \, dt$ grows too large, it is back-calculated as

$$\int e \, dt = \frac{1}{K_I} \left[K_{max} - K_P e - K_D \frac{de}{dt} \right] \quad (5.3)$$

The other alteration lies in the D-action. The difference operation which is used in the discrete-time implementation of the PID controller is not causal and is sensitive to noise. There are several solutions to reduce the noisiness of the derivative action, one of which is to use a fixed-width window. Here, instead of basing the derivative on one sample, it is based on n samples:

$$\Delta x = \frac{1}{n} ((x_k - x_{k-1}) + (x_{k-1} - x_{k-2}) + \dots + (x_{k-n+1} - x_{k-n})) \quad (5.4)$$

for some k, n . Rewriting this equation gives $\Delta x = \frac{x_k - x_{k-n}}{n}$. In this thesis, a fixed-width window of 10 samples is used.

5.2.2 State feedback controller

In a state-space model, the time derivative of a state vector, $\dot{\mathbf{x}}$ is related to the state vector $\mathbf{x}$ itself and a control vector $\mathbf{u}$:

$$\dot{\mathbf{x}} = A\mathbf{x} + B\mathbf{u} \quad (5.5)$$

$$\mathbf{y} = C\mathbf{x} + D\mathbf{u}, \quad (5.6)$$

where y is the system's output. A state feedback controller designed for such a representation attempts to bring the time derivatives of the states to zero. This is done by calculating a controller gain matrix K , based on the plant (Figure 5.3). This matrix can be computed in multiple ways, e.g. using pole placement or LQR.

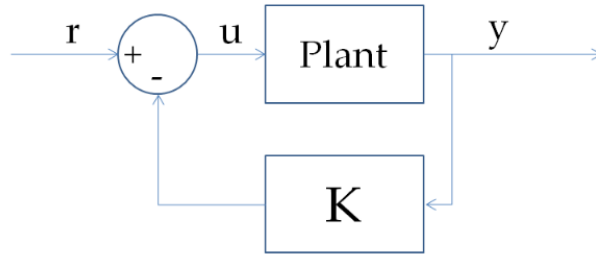


Figure 5.3: Block diagram of a state feedback controller.

Implementation The first thing to do is determining the states of the system. The inertia stores kinetic energy by undergoing an angular acceleration. This component adds two states, angle ϕ and angular velocity $\dot{\phi}$, to the system, as angular acceleration is integrated twice to produce angle ϕ . Next, work is done by the air at the exhaust port, which produces a torque via a pressure difference with the surrounding chambers. Therefore, exhaust pressure P_e also contributes to the dynamics of the system (see also Figure 3.10). The upstream and downstream pressures, P_u and P_d respectively, do not add extra information to the state-space representation. The upstream pressure in this case is constant and equal to supply pressure, while the downstream pressure is equal to P_e . This is due to the extra tube that was placed between the exhaust port and the downstream inlet.

The system is controlled by varying the amount of airflow through the proportional valve's orifice, $\dot{V}_p$. This volume flow is related to the proportional valve's input voltage. Since u_p influences the time derivative of pressure, which is linked to the torque, the control signal u_p can be understood as acting on the vane motor's jerk, the third time derivative of position.

As said, the goal of a state feedback controller is to steer the time derivatives of the states towards zero. Since the position (error) needs to disappear as well,

the integral of position over time (referred to as absement), $\int \phi$, is appended to the state vector:

$$\mathbf{x} = \begin{pmatrix} \int \phi \\ \phi \\ \dot{\phi} \\ P_e \end{pmatrix} \quad (5.7)$$

For the first two entries, finding an expression for the time derivatives is trivial: $(\int \phi) = \phi$, $\dot{\phi} = \dot{\phi}$. Angular acceleration follows from the torque balance on the vane motor (see Section 3.2.2):

$$\ddot{\phi} = \frac{1}{J} \left(T(\phi, P_e) - c\dot{\phi} - f \frac{\dot{\phi}}{|\dot{\phi}|} \right) \quad (5.8)$$

where the final term equals $S(\dot{\phi})$ in Equation 3.15.

Finally, the exhaust pressure P_e is related to the integral of the net mass flow at the exhaust:

$$P_e = \frac{R_s T}{V_e} \int \dot{m}_e dt \quad (5.9)$$

where $\dot{m}_e$ is the difference between incoming and outgoing mass flow, and R_s indicates the specific gas constant. The incoming mass flow (see Section 3.2.2) equals:

$$\dot{m}_{e,in} = \frac{P_{ue} \dot{\phi} V_d}{2\pi R T} \quad (5.10)$$

where P_{ue} indicates *expanded* upstream pressure, i.e. compartment pressure just before the exhaust is reached, and V_d is short for $V_{disp,nom}$ in Section 3.2.2, the nominal displaced volume in one stroke.

The outgoing mass flow is determined by the flow through the proportional valve's orifice and is a function of input voltage u_p and pressure difference over the valve ΔP :

$$\dot{m}_{e,out} = \rho \dot{V}_{out} = \frac{P_e}{R_s T} \dot{V}_{out}, \quad \dot{V}_{out} = f(u_p, \Delta P) \quad (5.11)$$

Note that it is assumed here that the flow is incompressible. Instead of u_p , now $u_p P_e$ is the control variable. This gives $\dot{m}_{e,out} = f(u_p P_e)$.

This results in the following state-space equations:

$$\begin{pmatrix} \dot{\phi} \\ \ddot{\phi} \\ \dot{P}_e \end{pmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & \frac{1}{J}(-c - d(\dot{\phi})) & \frac{1}{J}T(\phi, P_e) \\ 0 & 0 & \frac{P_{ue} V_d}{2\pi R T} & 0 \end{bmatrix} \begin{pmatrix} \int \phi \\ \phi \\ \dot{\phi} \\ P_e \end{pmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ -f(u_p P_e) \end{bmatrix} u_p \quad (5.12)$$

Torque-pressure relation The expression for torque is discontinuous, since chamber pressures make a jump at the in- and outlets as shown in Figure 4.7.

A continuous approximation can be acquired by considering two contributions that turn pressure difference into a torque; (i) volume work due to expansion, and (ii) torque generated by the pressure difference between two chambers.

The first factor can be determined by assuming any infinitesimal volume work due to expansion is turned into acceleration of the inertia:

$$PdV = Jd\ddot{\phi} \quad (5.13)$$

When summing this contribution over the entire cycle, the end result is:

$$J\Delta(\ddot{\phi}) \propto P_u V_{fill} - P_{ue} V_{exp} \quad (5.14)$$

The other factor is determined by considering the torque generated by a pressure difference between the exhausted and filled chambers. An approximation for this is:

$$T = (P_u - P_e) \bar{A}_{eff} \quad (5.15)$$

where $\bar{A}_{eff}$ is the average surface area of the vane between inlet and exhaust.

These two contributions result in the following state-space representation:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}u + \mathbf{G}\mathbf{v} \quad (5.16)$$

where the third term includes non-controllable external disturbances, and $\mathbf{v} = P_u$. These are the terms which relate to upstream pressure P_u , as this pressure does not change.

The A and G matrices then change to

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & \frac{1}{J}(-c - d(\dot{\phi})) & -\frac{1}{J}(V_e + \bar{A}_{eff}) \\ 0 & 0 & \frac{P_{ue}V_d}{2\pi V_e} & 0 \end{bmatrix}, \quad (5.17)$$

$$\mathbf{G} = \begin{bmatrix} 0 \\ 0 \\ 1 + V_u \\ 0 \end{bmatrix} \quad (5.18)$$

Controllability A full state feedback controller is also called a pole placement controller, since it allows a control engineer to place the closed-loop poles of the system at pre-determined locations in the s-plane. Since the poles directly influence the closed-loop response, the system can be given any desired dynamics.

However, this works only if the system is fully controllable. Controllability is defined via the Gram controllability matrix Q :

$$Q = [B \ AB \ A^2B \ \dots A^{n-1}B] \quad (5.19)$$

A system is said to be *completely controllable* if, starting from any state, the origin $\mathbf{x} = \mathbf{0}$ can be reached in finite time and by applying only valid control actions. If a system is not fully controllable, then there is a certain subspace whose values can not be changed by the control action. If this is the case, some subset of states can still be reached. It is then important that the non-reachable subspace has stable eigenvalues.

5.2.3 Brake control

The final controller that is implemented utilizes a braking system. If the vane motor rotates at a relatively constant speed, the moment where it reaches the setpoint can be computed. If the delay of the braking system is correctly taken into account, and is consistent, then theoretically the vane motor could be stopped at the setpoint itself. In practice however, velocity will not be constant due to a varying friction torque. Part of this can be compensated by applying velocity control to the system.

Ideally, the velocity is low enough so that a small spread in braking time does not affect the end position much, but high enough so that it is not affected significantly by changing friction. First of all, experiments are done in order to determine the variance in braking distance for different speeds. If this variance is small, then, if the speed is accurately known, the stopping position can be computed.

In order to assess the feasibility of the brake system, the vane motor is supplied with air at constant pressure, which causes it to rotate at a relatively steady speed. After 8 seconds, the brake is applied. The distance that the vane motor rotates between when the brake signal is given and when the vane motor comes to a halt is logged. Average speed 0.1 seconds prior to braking is also extracted from the data.

5.3 Results

The results presented in this chapter are the controllers discussed in the previous section.

PID control The PID controller is treated first. Its design is largely covered in Section 5.2.1.

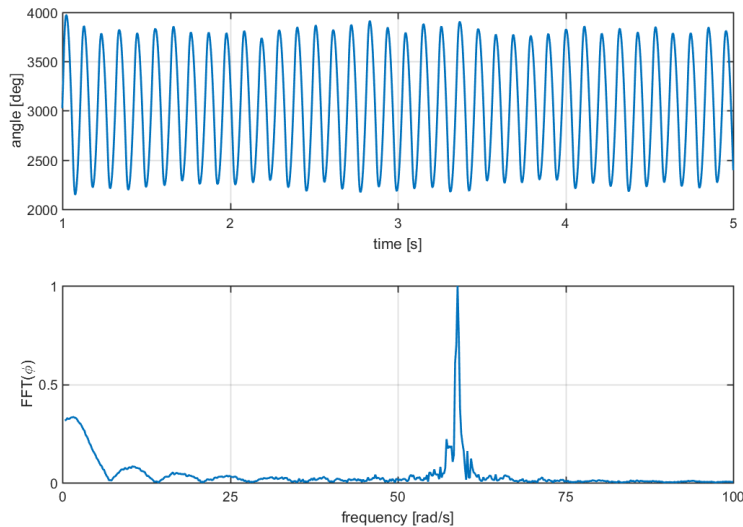


Figure 5.4: Time (top) and frequency (bottom) content analysis of stable oscillation obtained in Ziegler-Nichols tuning.

Table 5.1: Different rules-of-thumb for tuning a PID controller using the Ziegler-Nichols method.

Method	K_P	K_I	K_D
Classic	$0.6K_u$	$T_u/2$	$T_u/8$
"No overshoot"	$0.22K_u$	$2/T_u$	$T_u/3$
Lu, Hwang [4]	$0.6K_u$	$2K_P/T_u$	$K_P T_u/8$
This thesis	$0.4K_u$	$8K_P/T_u$	$K_P T_u/5$

In Figure 5.4, a stable oscillation is obtained from the experiment described in Section 5.2.1. From the frequency spectrum analysis in the bottom figure, it is noted that the system oscillates stably at a frequency of 60 rad/s. Now that both the period of oscillation and the ultimate gain are determined, the constants K_P , K_I , K_D can be calculated according to many rules-of-thumb, see Table 5.1. The parameters used in [4] were used as a starting point. These values were then altered until a satisfactory response was obtained.

Brake control Considering the brake controller, preliminary tests were done to determine the distance to standstill at different speeds. The results are plotted in Figure 5.5.

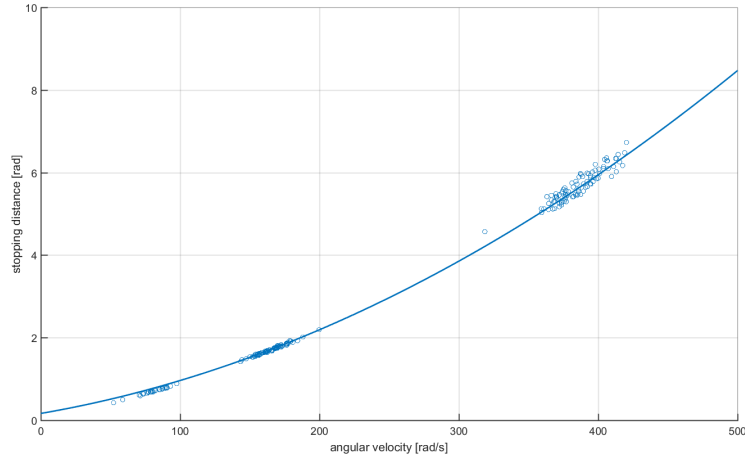


Figure 5.5: Braking distance as a function of rotation speed. Circles indicate experimental results, line indicates a quadratic fit.

It can be seen that there is a quadratic relation between motor speed and braking distance.

5.4 Discussion

State feedback control The state feedback controller requires good knowledge of the system to be controlled. This includes a relation between input voltage u_p and volume flow V_{ex} of the proportional valve. This relation is often dependent

on the air pressure and flow laminarity in the valve. The manufacturer has been contacted to provide such a relation. They have not yet responded with this relation.

Alternatively, mass flow rate through the valve can be approximated from theory. In general, mass flow through an orifice is given by

$$\dot{m} = \frac{C_d}{\sqrt{1 - \left(\frac{d}{D}\right)^4}} \epsilon \frac{\pi}{4} d^2 \sqrt{2\rho_u \Delta P} \quad (5.20)$$

Here, C_d is a discharge coefficient which depends on orifice geometry, d is the orifice diameter, D the pipe diameter, ΔP the differential pressure across the orifice, ϵ the expansibility factor (itself a function of d , D , ΔP , and polytropic index n), and ρ_u the upstream fluid density. Maximum flow rate could be measured by a flow meter, from which the orifice diameter can be determined for various input voltages. When orifice diameter is known as a function of input voltage, mass flow can be approximated in real-time during operation.

Simulations were done in order to determine the system's controllability. The parameters that have the highest uncertainty are c and f , the viscous and Coulomb friction coefficients, respectively. Tests regarding the controllability were done for a range of these parameters corresponding to realistic values of the friction coefficients. From simulations, it was found that the system is not fully controllable for a viscous friction coefficient of more than 0.0538 Ns. The reachable subspace, which is defined by the controllability matrix' image, varies with the friction coefficient.

Brake control There is a quadratic relation between motor speed and braking distance. This can be explained by the fact that the kinetic energy stored in the inertia scales quadratically with speed. In Figure 5.6, a close-up of left part of

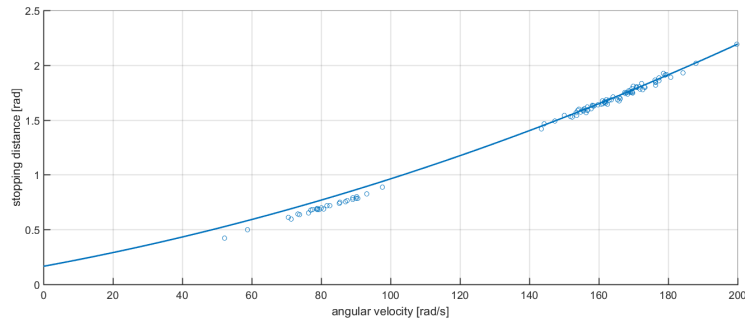


Figure 5.6: Zoomed-in part of Figure 5.5.

Figure 5.5 can be seen. There is a spread of around ± 0.06 rad around the fitted line. At higher speeds, this deviation becomes larger. If the vane motor can be rotated at a low speed, and this speed can be precisely determined, then there is a possibility that the vane motor can be precisely positioned. More experiments need to be done to determine the feasibility of the brake controller.

Chapter 6

Performance analysis

6.1 Objective

The objective of this chapter is to determine the performance of the vane motor servo system in terms of precision, speed, and torque.

6.2 Materials & methods

To determine the system's performance, several experiments are performed, each of which is discussed below.

6.2.1 Experiments

The precision is evaluated for a host of scenarios, described below. Each experiment is designed as follows. A setpoint is generated and the controller starts to position the vane motor. Once error is within margins (1° unless stated otherwise), control stops. After a few seconds of rest, a new setpoint is generated and the cycle starts anew.

During each experiment, the precision is determined by the error (as measured by the encoder) after 10 seconds. For all the experiments listed below, the PID controller designed in Section 5.2.1 is used.

Constant setpoints The precision of the system may depend on the motor configuration corresponding to a specific target position, due to the vane motor's non-linear pressure-torque relation. It may also depend on the distance from start to target position. In a first experiment, the motor is controlled to move from one target position (0°) to another (3000°), back and forth.

Random setpoints It is possible that, when using the same setpoints every time, the system positions itself in a configuration that is favourable. The vane motor should instead be able to be precisely positioned at any angle. Therefore, experiments with random setpoints are also performed. For one set of experiments, these setpoints are generated between 0° and 360° to see

Table 6.1: Various statistics regarding performed experiments.

#	Tube length ¹ [m]	Accepted error [°]	T [Nm]	n [-]	Success ² [%]	Settling time ³ [s]
1	0.7	1	0.0073 ± 0.0012	148	100 (-)	1.96 ± 1.15
2	0.7	1	0.0078 ± 0.0011	141	100 (-)	2.09 ± 1.19
3	0.7	1	0.0108 ± 0.0024	165	100 (-)	1.37 ± 1.48
4	0.7	1	0.0091 ± 0.0019	155	100 (-)	1.83 ± 0.75
5	0.7	0.5	0.0067 ± 0.0012	158	100 (-)	1.67 ± 1.89
6	0.7	0.5	0.0067 ± 0.0015	151	100 (-)	1.71 ± 1.27
7	5	1	0.0076 ± 0.0012	133	98.5 (99.2)	2.35 ± 0.94
8	5	1	0.0079 ± 0.001	131	99.2 (100)	2.37 ± 0.92
9	5	1	0.0066 ± 0.0016	121	95.9 (96.7)	2.43 ± 1.34
10	5	1	0.0062 ± 0.0016	147	84.4 (92.5)	4.82 ± 2.50
11	5	1	0.008 ± 0.0018	207	88 (96.1)	3.91 ± 2.50
12	5	1	0.0059 ± 0.0011	99	86.9 (98)	4.48 ± 1.88

behaviour of the controller over short distances. For another set of experiments, the setpoints are generated between 0° and 36000° .

Longer tubes This experiment determines how precise the vane motor can be positioned while the valves are outside of the MRI bore. In the set-up shown in Figure 3.12, distance between the control valves and the vane motor is around 70 cm. Experiments are performed with 5 meters of tubing between the vane motor and the valves.

6.3 Results

Results of positioning experiments for different scenarios discussed in Section 6.2.1 are presented. Ten experiments were performed, each with different settings. An overview is found in Table 6.1.

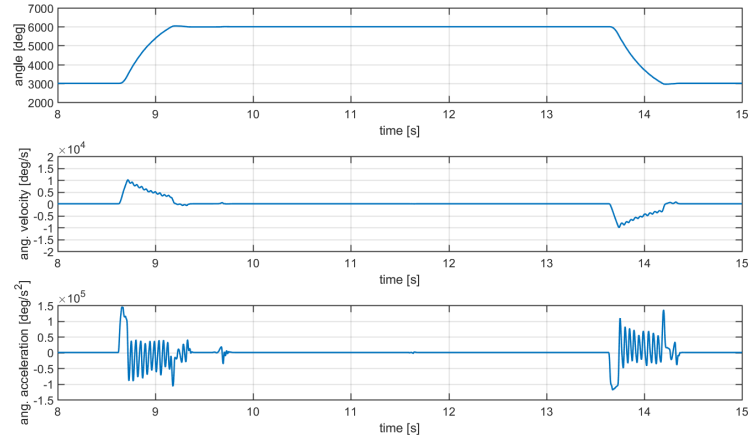
In Figure 6.1, position, velocity, and acceleration are shown during operation, with the latter two reconstructed by a state variable filter. This is done for both short and long tubes.

The average error for experiments 1-3, along with its standard deviation are plotted in Figure 6.2; it can be seen from the top plots that there is a small overshoot. This overshoot is compensated within the first second, however. The amount of overshoot during the first seconds is also fairly predictable, judging from the small error bars around it. The settling time, defined here as the time to reach steady-state error, is seen to be around 2 seconds.

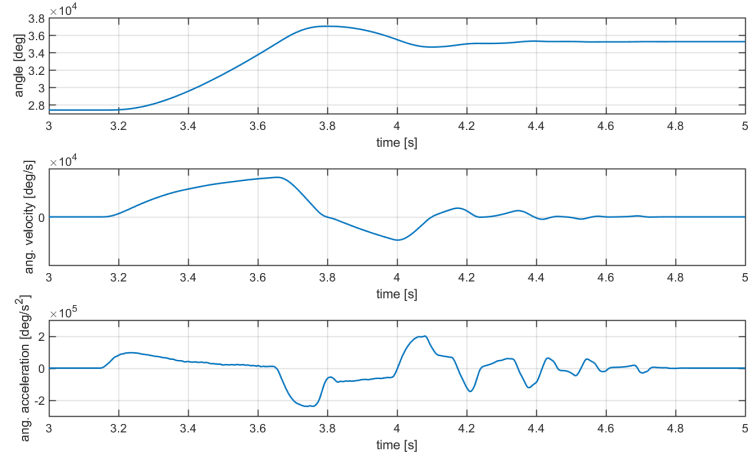
The plots on the bottom show the error near steady-state, along with the standard deviation. After 2 seconds, all errors are within bounds.

A histogram showing the distribution of steady-state errors is shown in Figure 6.3. While not all setpoints are achieved within ten seconds, it can be seen that, eventually, all steady-state errors are within margin. Barring a few discrepancies, the distribution of steady-state error seems uniform.

¹ from valves to motor. ² after 10 seconds (after 20 seconds). ³ only successful runs.



(a) Experiment 4.



(b) Experiment 7.

Figure 6.1: Position (top), velocity (middle), and acceleration (bottom) over time. Velocity and acceleration signals were not measured, but constructed using a state variable filter. Top set of plots for tubes of 0.7 m, bottom set of plots for tubes of 5 m.

Plots showing the percentage of setpoints reached as a function of time is listed for four experiments in Figure 6.4. In experiment 7, where longer tubes are used, it can be seen that around 90% of setpoints are achieved in the allowed time of 10 seconds.

6.4 Discussion

Successfulness Given an unlimited amount of time, all setpoints were achieved eventually. For some, this took more than a minute. In the allowed time of 10 seconds, the successfulness is lower than 100% for experiments where long

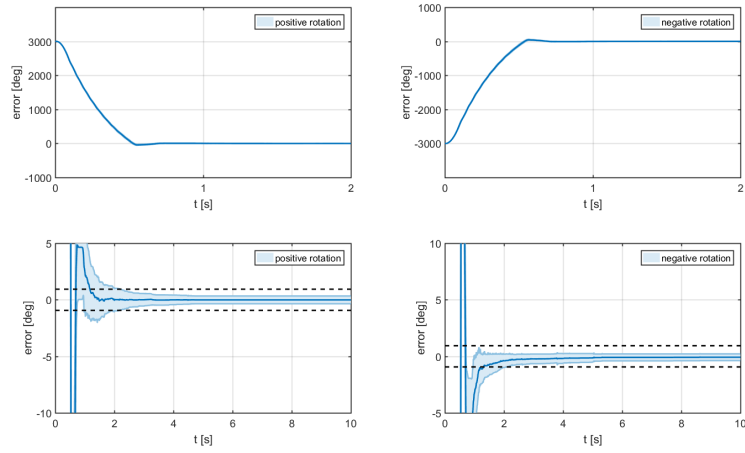


Figure 6.2: Error for positive (left, $3000^\circ \rightarrow 6000^\circ$) and negative (right, $6000^\circ \rightarrow 3000^\circ$) positioning operations, averaged over experiments 1-3. In the bottom plots, a detail of the average error is shown including the error resolution requirement from Section 3.1.1 (dashed lines).

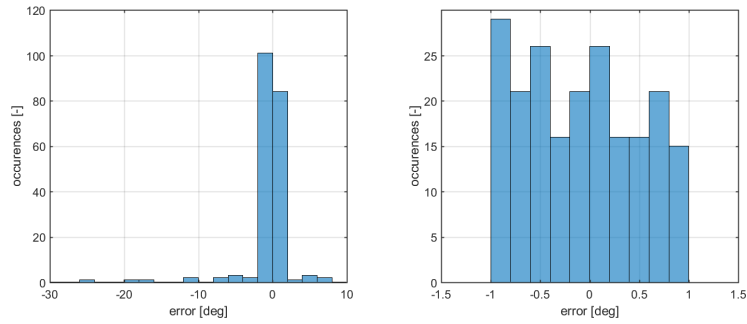


Figure 6.3: Distribution of steady-state error for experiment 11, after 10 seconds (left) and steady-state (right).

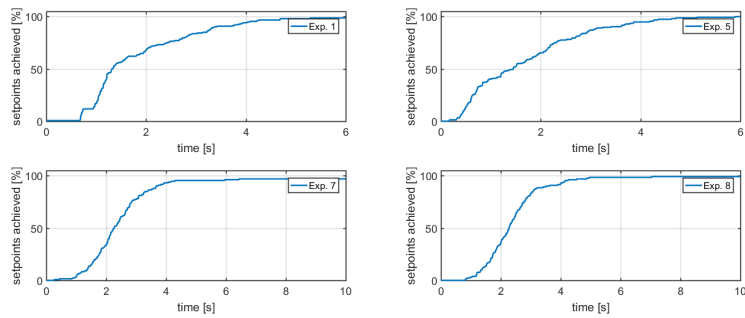


Figure 6.4: Percentage of setpoints achieved as a function of time, for various experiments (see Table 6.1).

tubes are used. This is due to static friction, which sometimes places the vane motor in a limit cycle. Often, limit cycling is overcome by temporarily opening

the proportional valve to give the system a nudge in torque. Limit cycles are harder to break out of in some cases, which appears to be configuration-dependent. This could be attributed to the cogging effect which occurs due to the non-linear torque-pressure relation.

Response Looking at Figure 6.1, a few things are worth noting. In the top set of plots, where short tubes are used, the velocity plot shows an oscillation when decreasing speed. This is due to the D-action and results in a smooth response for position. The derivative action's effect is even more noticeable in the acceleration plot.

Looking at the bottom set of plots, where tubes of 5 m are used, the response looks completely different: less chattering, constant readjusting, and less predictable acceleration behaviour. This can be attributed to the delay caused by the tubes. Since there is approximately a 0.15 second delay between opening a valve and the pressure having settled at the motor, overshoot happens more quickly than with shorter tubes. Once enough torque is produced such that the motor starts moving, the delay again easily causes it to overshoot. In the bottom-left plot of Figure 6.2, the vane motor can be seen to stand still for a small amount of time, between 0.8 s and 1 s. This happens when the response has overshoot its target and has to reverse directions. The direction of airflow reverses and the proportional valve is closed. The I-action accumulates, and after some time produces a torque that exceeds the static friction torque. This causes the motor to move again.

The designed PID controller is quite aggressive, which can be attributed to the I-action. This was chosen in order to get a fast response of the system. However, this also means overshoot is higher than with a less aggressive controller.

Overshoot can be a problem in some applications. For the NPS, when an overshoot occurs, there is a possibility that the segments move against their limits. This simply stops motion of the segments and does not affect the system in a negative way. In applications where overshoot is undesired, controller gains can be lowered to achieve a satisfactory, albeit slower, response.

Friction The Prony brake absorbs a certain friction torque, which is determined by how far the bolts are tightened. An increase in this friction can cause the system to become stuck in stick-slip, leading to limit cycling. The motor sometimes becomes stuck in static friction. Even with the exhaust valve completely open, not enough torque is produced to gain acceleration. The solution is to increase inlet pressure of the system. However, the maximum allowed pressure of the vane motor is specified to be 4 bar. Measurements with a higher inlet pressure risk damaging the vane motor and are therefore not executed.

Chapter 7

Conclusions & outlook

7.1 Conclusions

The objective of this thesis has been to *develop a vane motor servo system for precision positioning of a needle guide in a CT- or MR-environment*. A choice of actuator was made based on a literature study (Chapter 2). The pneumatic vane motor was chosen because of its high power-to-weight ratio, its inherent MR-safeness, and its simple operating principle.

In Chapter 3, a novel system of control valves and tubes was designed in order to facilitate position control. Several models from literature were combined to form a single, non-linear Simulink model. A set-up of the designed system was realised.

Chapter 4 used the model constructed in Chapter 3 in order to identify system characteristics which could make positioning a vane motor challenging, such as the configuration-dependent torque and high friction. A system identification was done in Section 4.2.4 in order to uncover the system's dynamics. It was found that the system behaves as a 5th order system, though none of the fitted linear models produced satisfactory residuals. This indicates that non-linearities are present in the system, supporting the literature study performed in Chapter 2.

Chapter 5 used the dynamical model of the system to design controllers. Three controllers were designed. First, a PID controller with integral anti-wind-up. A full state feedback controller that uses the vane motor's position, velocity, and absement, as well as the exhaust pressure was also designed. Finally, a brake was added to the system that can force the vane motor to a standstill.

The PID controller was implemented on the actuator system and the precision design goal was evaluated for different scenarios. For "short" tube lengths of 0.7 m between actuator and control valves, it was found that the PID controller achieves the requirement in 100% of the cases. The average precision across experiments 1 through 6 is -0.0187 ± 0.3295 degrees. In experiments 5 and 6, a precision of 0.5° was achieved for all cases.

For tubes of 5 meters, the design goal of 1° precision is achieved in 10 seconds for 90% of the operations. When friction is lowered, all setpoints are achieved within 7 seconds. The average precision across experiments 6 through 12 is 0.0483 ± 0.6946 degrees (Table 6.1, Figure 6.4).

These results are an improvement over [4], where a pneumatic vane motor achieved an accuracy of 1.1° when controlled by a PID controller. The authors used a significantly larger motor and short tubes (length not specified).

In conclusion, this thesis has demonstrated the feasibility of a pneumatic servo motor for precision positioning in a CT- and MR-environment. Due to its performance and simplicity, the pneumatic vane motor system as designed in this thesis opens up a host of applications in medical robotics.

7.2 Discussion

Discussion of most results is already treated in their respective chapters. This section addresses some topics that pertain to multiple chapters.

Prony brake Most of experiments performed in this thesis utilised the Prony brake. While this mechanism allows for easy adjusting of the system's friction, the results are not always consistent. This stems from the construction of the Prony brake itself (Figure 7.1). An imperfect contact between the Prony brake's plates and the shaft can cause a varying torque. The bolts used to align the Prony brake (in the middle) also have an influence on the absorbed friction torque.

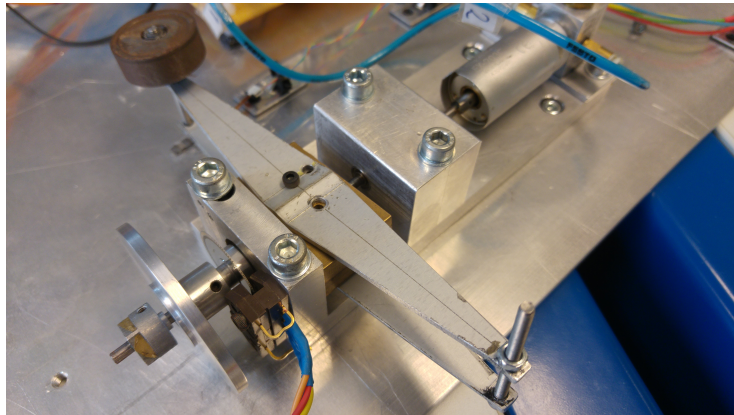


Figure 7.1: Close-up of the Prony brake, which consists of two plates pressed against the shaft. By tightening or loosening the bolt, the amount of friction can be varied.

Limit cycling As discussed in the paragraph on friction in Section 6.4, a system with higher friction is often difficult to control because limit cycling occurs. Figure 7.2 illustrates this phenomenon.

When applying position control on the vane motor, the amount of friction and the cogging effect described in Section 3.2.1 complicate the final part of the positioning. One problem is that, when the system overshoots and the direction has to be reversed, both friction and cogging will make the vane motor behave discontinuously. For example, consider the situation where the system has overshoot its setpoint and has to reverse directions to compensate. When the

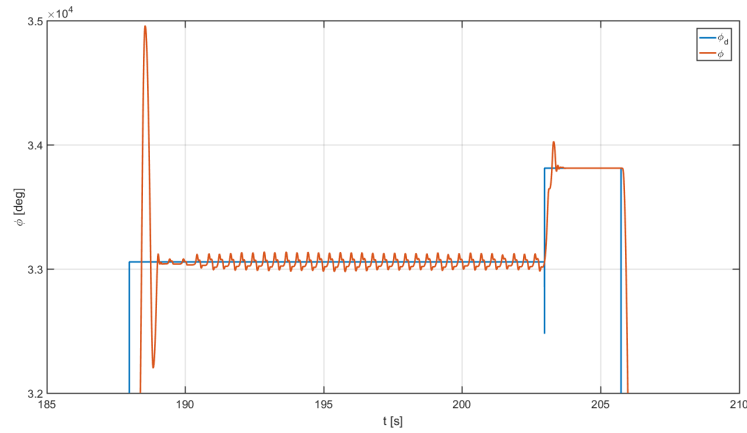


Figure 7.2: Phenomenon of limit cycling seen during positioning. Blue line indicates desired angle, orange line shows actual angle.

inlet chambers' pressures switch (e.g. 3 bar $\leftrightarrow$ 4 bar), torque on the vanes will make a discontinuous jump, which is difficult to control.

The solution is to place an additional tube directly from the air inlet, which is always at inlet pressure, to the exhaust tube before the proportional valve. By controlling the flow of air through this tube, the pressure at the exhaust can be set to supply pressure, so that the net torque between inlet and outlet is zero. In theory, this makes the transition from positive to negative direction smoother. This configuration is shown in Figure 7.3.

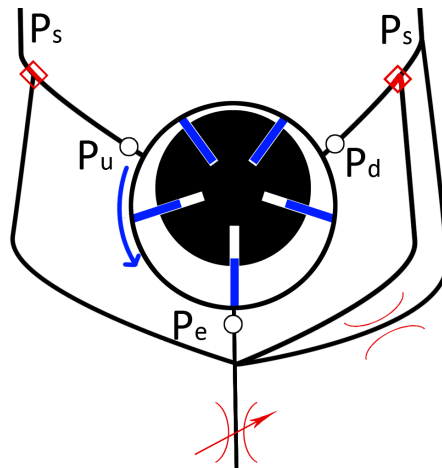


Figure 7.3: Layout of the pneumatic system with an added tube and throttle between the exhaust and supply pressure.

7.3 Outlook

This section discusses some subjects of the vane motor that are not handled in the main body of this thesis, but are of interest to any future work. First some general remarks are given about the vane motor-based system, after which two subjects are discussed in-depth: the mechanical design of the vane motor and control techniques.

In this thesis, a proportional valve was used since it allowed for continuous control techniques, which made it simpler to implement controllers. However, binary valves can also be used to throttle the output. In principle, any continuous signal can be reconstructed by applying pulse-width modulation to a binary valve up to its switching frequency. Since binary valves are a lot cheaper than proportional valves, this may be of interest when implementing the vane motor system in a commercial product.

To increase the range of speeds and torques that can be produced by the vane motor, an extra valve can be added before the directional valves that can be used to control supply pressure.

Integrated design of the vane motor This thesis has proven that a pneumatic vane motor, which can be made MRI-safe, can be positioned precisely. The next step is to optimize the vane motor's geometry itself and design a CT- and MRI-safe version of the vane motor. The dimensions have a great influence on the vane motor's dynamics. For instance, by increasing the difference between rotor and stator radius, the amount of expansion increases, which adds extra work to the cycle. Increasing the distance between inlet and outlet has the same effect. This means that the right part of the line in Figure 3.13 becomes more stretched. Another parameter to vary is the amount of vanes. More vanes add additional friction to the system, but can result in more work during a cycle as expansion also increases. The non-linear Simulink model constructed in Chapter 3 can be used to simulate several geometries.

Redesign of the vane motor for the NPS needs to take into account the geometry of the NPS as well. There is a space reserved of about $35 \times \text{Ø}16$ mm for the actuator, which has to include the vane motor itself, an optical encoder, and a connection to the worm. The whole mechanism has to be made airtight so as to not lose power.

A preliminary design was made and is shown in Figure 7.4. The rotor and worm are made out of one part. The chamber is made airtight with a labyrinth seal (see Figure 7.5). Air enters and leaves the vane motor through holes in the side of the stator, since this is more compatible with the NPS' geometry.

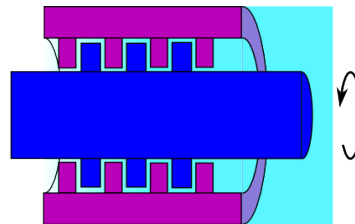
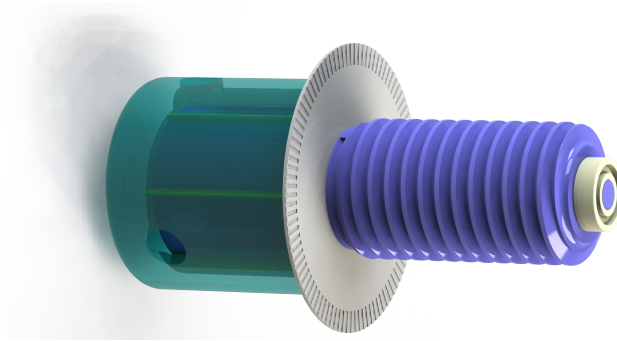
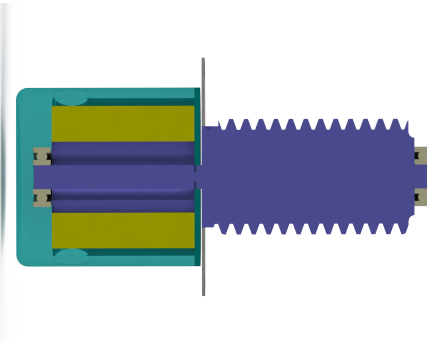


Figure 7.5: Depiction of a labyrinth seal. The size and geometry of the area between stator (pink) and rotor (blue) prohibits airflow, making a relatively airtight fit [58].



(a) Impression of a vane motor as designed for the NPS, integrated with an optical encoder disc (white) and worm gear (dark blue).



(b) Section view of the vane motor. Worm and rotor in blue, vanes in yellow, housing including labyrinth seal in teal, bearings in beige, encoder wheel in grey.

Figure 7.4: Impression of a vane motor as designed for the NPS, 3D model (top) and section view (bottom).

Control PID control was shown to produce mostly satisfactory results. However, with a more sophisticated controller, higher precision may be achieved. One of these is the state feedback controller (Section 5.2.2). If the linearisation made proves to be inaccurate, the controller can be expanded to a non-linear version.

Another control method which is found in literature is the sliding mode controller. In [4], a precision of around 0.35 degrees was achieved by applying such a controller. Sliding mode control is a form of variable structure control, where a non-linear system's dynamics are altered by applying a discontinuous control signal. This causes the system to 'slide' along a pre-defined surface.

This type of control can be very robust and is therefore often used in control of non-linear systems [4].

References

- [1] M. M. Arnolli, N. C. Hanumara, M. Franken, D. M. Brouwer, and I. A. M. J. Broeders, "An overview of systems for CT- and MRI-guided percutaneous needle placement in the thorax and abdomen," *The International Journal of Medical Robotics and Computer Assisted Surgery*, vol. 11, no. 4, pp. 458–475, dec 2015.
- [2] R. Gassert, R. Moser, E. Burdet, and H. Bleuler, "MRI/fMRI-compatible robotic system with force feedback for interaction with human motion," *IEEE/ASME Transactions on Mechatronics*, vol. 11, no. 2, pp. 216–224, apr 2006.
- [3] N. V. Tsekos, A. Khanicheh, E. Christoforou, and C. Mavroidis, "Magnetic Resonance Compatible Robotic and Mechatronics Systems for Image-Guided Interventions and Rehabilitation: A Review Study," *Annual Review of Biomedical Engineering*, vol. 9, no. 1, pp. 351–387, aug 2007.
- [4] C.-H. Lu and Y.-R. Hwang, "Modeling of an air motor servo system and robust sliding mode controller design," *Journal of Mechanical Science and Technology*, vol. 26, no. 4, pp. 1161–1169, apr 2012.
- [5] S. Pandian, F. Takemura, Y. Hayakawa, and S. Kawamura, "Control performance of an air motor-can air motors replace electric motors?" in *Proceedings 1999 IEEE International Conference on Robotics and Automation (Cat. No.99CH36288C)*, vol. 1. IEEE, 1999, pp. 518–524.
- [6] N. Yu and R. Riener, "Review on MR-Compatible Robotic Systems," *Biomedical Robotics and Biomechanics*, 2006.
- [7] Netherlands National Institute for Public Health and the Environment, "Trends in the number of CT procedures (in Dutch)," <http://www.rivm.nl/Onderwerpen/M/Medische{ }Stralingstoepassingen/Trends{ }en{ }stand{ }van{ }zaken/Diagnostiek/Computer{ }Tomografie/Trends{ }in{ }het{ }aantal{ }CT{ }onderzoeken>, (Accessed 2017-01-23).
- [8] Netherlands National Institute for Public Health and the Environment, "Echography and MRI (in Dutch)," <http://www.rivm.nl/Onderwerpen/M/Medische{ }Stralingstoepassingen/Trends{ }en{ }stand{ }van{ }zaken/Diagnostiek/Echografie{ }en{ }MRI>, (Accessed 2017-01-23).
- [9] "MR Safety Testing," <http://www.magneticresonancesafetytesting.com/genpg.asp?pgname=Terminology>, (Accessed 2017-07-03).

- [10] D. Stoianovici, A. Patriciu, D. Petrisor, D. Mazilu, and L. Kavoussi, "A New Type of Motor: Pneumatic Step Motor." *IEEE/ASME transactions on mechatronics : a joint publication of the IEEE Industrial Electronics Society and the ASME Dynamic Systems and Control Division*, vol. 12, no. 1, pp. 98–106, feb 2007.
- [11] J. Beringer, "Particle Data Group: Passage of particle through matter 1," *Phys. Rev.*, vol. D86, no. 010001, 2012.
- [12] "CT scan procedure - schoolphysics," http://www.schoolphysics.co.uk/age16-19/Medicalphysics/text/CT_{_}scanning/index.html, (Accessed 2017-07-28).
- [13] R. W. Briggs, I. Dy-Liacco, M. P. Malcolm, H. Lee, K. K. Peck, K. S. Gopinath, N. C. Himes, D. A. Soltysik, P. Browne, and R. Tran-Son-Tay, "A pneumatic vibrotactile stimulation device for fMRI," *Magnetic Resonance in Medicine*, vol. 51, no. 3, pp. 640–643, mar 2004.
- [14] A. Krieger, R. C. Susil, C. Ménard, J. A. Coleman, G. Fichtinger, E. Atalar, and L. L. Whitcomb, "Design of a novel MRI compatible manipulator for image guided prostate interventions." *IEEE transactions on bio-medical engineering*, vol. 52, no. 2, pp. 306–13, feb 2005.
- [15] J. Naranjo, E. Kussul, and G. Ascanio, "A new pneumatic vanes motor," *Mechatronics*, vol. 20, no. 3, pp. 424–427, apr 2010.
- [16] DEPRAG, "No Metal: Amazing air motors made from ceramic materials," <http://www.deprag.com/en/company/press-release/presse200911a.html>, (Accessed 2017-01-23).
- [17] D. Ben-Dov and S. Salcudean, "A force-controlled pneumatic actuator," *IEEE Transactions on Robotics and Automation*, vol. 11, no. 6, pp. 906–911, 1995.
- [18] E. Richer and Y. Hurmuzlu, "A High Performance Pneumatic Force Actuator System: Part II Nonlinear Controller Design," *Journal of Dynamic Systems, Measurement, and Control*, vol. 122, no. 3, p. 426, 2000.
- [19] V. Groenhuis and S. Stramigioli, "Laser-Cutting Pneumatics," *IEEE/ASME Transactions on Mechatronics*, vol. 21, no. 3, pp. 1604–1611, jun 2016. [Online]. Available: <http://ieeexplore.ieee.org/document/7353189/>
- [20] V. Groenhuis and S. Stramigioli, "Rapid Prototyping High/Performance Metal Free Pneumatic Stepper Motors," 2017.
- [21] G. Ganesh, R. Gassert, E. Burdet, and H. Bleuler, "Dynamics and control of an MRI compatible master-slave system with hydrostatic transmission," in *IEEE International Conference on Robotics and Automation, 2004. Proceedings. ICRA '04. 2004.* IEEE, 2004, pp. 1288–1294 Vol.2.
- [22] N. Yu, C. Hollnagel, A. Blickenstorfer, S. Kollias, and R. Riener, "fMRI-Compatible Robotic Interfaces with Fluidic Actuation," *Robotics: Science and Systems IV*, 2009.

- [23] Nanomotion, "Nanomotion HR Servo Motors," <https://www.nanomotion.com/product-type/nanomotion-motors/>, (Accessed 2017-02-07).
- [24] P. Moreira, G. Van De Steeg, T. Krabben, J. Zandman, E. E. G. Hekman, F. Van Der Heijden, R. Borra, and S. Misra, "The MIRIAM Robot: A Novel Robotic System for MR-Guided Needle Insertion in the Prostate," *Journal of Medical Robotics Research*, 2016.
- [25] F. Sergi, A. C. Erwin, and M. K. OMalley, "Interaction Control Capabilities of an MR-Compatible Compliant Actuator for Wrist Sensorimotor Protocols During fMRI," *IEEE/ASME Transactions on Mechatronics*, vol. 20, no. 6, pp. 2678–2690, dec 2015.
- [26] *Piezomotor - Wikipedia*.
- [27] G. S. Fischer, A. Krieger, I. Iordachita, C. Csoma, L. L. Whitcomb, and G. Fichtinger, "MRI Compatibility of Robot Actuation Techniques A Comparative Study," *Medical Image Computing and Assisted Intervention*, vol. 11, no. 2, pp. 509–517, 2008.
- [28] O. Wendt, J. Oellinger, T. C. Lüth, R. Felix, and U. Boenick, "The effects of the use of piezoelectric motors in a 1.5-Tesla high-field magnetic resonance imaging system (MRI)," *Biomedizinische Technik. Biomedical engineering*, vol. 45, no. 1-2, pp. 20–5, 2000.
- [29] N. V. Tsekos, A. Ozcan, and E. Christoforou, "A Prototype Manipulator for Magnetic Resonance-Guided Interventions Inside Standard Cylindrical Magnetic Resonance Imaging Scanners," *Journal of Biomechanical Engineering*, vol. 127, no. 6, p. 972, 2005.
- [30] University of Freiburg, "Imaging of Neuro disease Using High Field MR and Contrastophores," http://portal.uni-freiburg.de/inumac/copy_{-}of_{-}inumac-1, (Accessed 2017-02-02).
- [31] Y. Bar-Cohen, "Artificial Muscles using Electroactive Polymers (EAP): Capabilities, Challenges and Potential," pp. 125–224, 2005.
- [32] F. Carpi, A. Khanicheh, C. Mavroidis, and D. De Rossi, "MRI Compatibility of Silicone-Made Contractile Dielectric Elastomer Actuators," *IEEE/ASME Transactions on Mechatronics*, vol. 13, no. 3, pp. 370–374, jun 2008.
- [33] S. Hau, A. York, and S. Seelecke, "High-Force Dielectric Electroactive Polymer (DEAP) membrane actuator," Y. Bar-Cohen and F. Vidal, Eds. International Society for Optics and Photonics, apr 2016, p. 97980I.
- [34] "EAP Composite - Wikipedia," <https://upload.wikimedia.org/wikipedia/commons/c/c3/EAP-composite.png>, (Accessed 2017-07-28).
- [35] "DC motor - Wikipedia," https://en.wikipedia.org/wiki/DC_{-}motor_{-}#/media/File:Electric_{-}motor_{-}cycle_{-}2.png, (Accessed 2017-07-28).
- [36] GAST, "1AM-NRV-39A Product Specifications," <http://gastmfg.com/uploads/data-sheets/STD111.pdf>, (Accessed 2017-02-07).

- [37] NSK, "EX-203C Set," <http://www.nsktech.com.au/uploads/70472/ufiles/EX-103{ }203.pdf>, 2009, (Accessed 2017-02-01).
- [38] Morita, "AIR TORX Powerful torque Air Motor," <http://www.jmorita-mfg.com>, (Accessed 2017-02-06).
- [39] Deprag, "Advanced Line Stainless Steel Air Motors," <http://www.deprag.com/fileadmin/bilder{ }content/emedi a/broschueren{ }pics/emedi a{ }druckluftmotoren/D6400/D6400en.pdf>, (Accessed 2017-02-06).
- [40] Modec, "MT05 Air Motor," <http://www.modec.fr/wp-content/uploads/2016/04/mt05{ }2009.pdf>, (Accessed 2017-02-06).
- [41] Desoutter, "Reversible Air motors," <http://www.desouttertools.com/tools/12/air-motors/1203/reversible>, (Accessed 2017-02-06).
- [42] HydroLeduc, "Microhydraulics," <http://www.hydroleduc.com/images/links/catalogues/hydroleducdocumentation{ }en/microhydraulics/hydroleduc{ }micro{ }en.pdf>, (Accessed 2017-02-07).
- [43] MOOG, "A085 Series Servoactuators," <http://www.moog.com/content/dam/moog/literature/ICD/a085hydraulicactuators.pdf>, (Accessed 2017-02-07).
- [44] Bansbach, "Micro-Hydraulics," <http://www.bansbach.com/index.php/en/products/micro-hydraulics>, (Accessed 2017-02-07).
- [45] Micromatic, "Rotary Actuators," <http://www.flodraulic.com/assets/actuator-product-catalog-web.pdf>, (Accessed 2017-02-07).
- [46] Pi, "Fast Linear Actuator with PIShift Piezomotor," <http://www.pi-usa.us/products/PDF{ }Data/N412{ }Miniature{ }Piezo{ }Motor{ }Actuator.pdf>, (Accessed 2017-02-07).
- [47] Pi, "Piezo Motor Positioners," <http://piezo-motor.net/pdf/Linear{ }Stage{ }Ultrasonic{ }Motor{ }Small{ }Rotation{ }Stage.pdf>, (Accessed 2017-02-07).
- [48] DSM, "Custom Piezoelectric Motors," <http://www.dynamic-structures.com/custom-piezo-motors/>, (Accessed 2017-02-07).
- [49] M. motors, "RE-max Series," <http://storkdrives.com/wp-content/uploads/2013/10/maxon-RE-max-catalog-data1.pdf>, (Accessed 2017-02-07).
- [50] M. motors, "DCX Series," <http://pdf.directindustry.com/pdf/maxon-motor/maxon-x-drives/7173-363519.html>, (Accessed 2017-02-07).
- [51] P. Beater, "Modelling and Control of Pneumatic Vane Motors," *International Journal of Fluid Power*, vol. 5, no. 1, pp. 7–16, jan 2004.
- [52] E. Richer and Y. Hurmuzlu, "A High Performance Pneumatic Force Actuator System Part 1 - Nonlinear Mathematical Model," *ASME Journal of Dynamic Systems Measurement and Control*, vol. 122, no. 3, pp. 416–425, 2000.

- [53] "Multisine - littledebugger," <https://www.littledebugger.com/2017/02/14/sum-sines-html5javascript/>, (Accessed 2017-07-28).
- [54] "Linear chirp - Wikipedia," <https://upload.wikimedia.org/wikipedia/commons/thumb/c/cf/Linear-chirp.svg/512px-Linear-chirp.svg.png>, (Accessed 2017-07-28).
- [55] R. Aarts, "System Identification & Parameter Estimation handout," 2013.
- [56] S.-Y. Chen, Y.-H. Hung, and S.-S. Gong, "Speed Control of Vane-Type Air Motor Servo System Using Proportional-Integral-Derivative-Based Fuzzy Neural Network," *International Journal of Fuzzy Systems*, vol. 18, no. 6, pp. 1065–1079, dec 2016.
- [57] J. Van Dijk and R. Aarts, "Analytical one parameter method for PID motion controller settings," *IFAC Proceedings Volumes (IFAC-PapersOnline)*, vol. 2, no. PART 1, pp. 223–228, 2012.
- [58] "Labyrinth seal - Wikipedia," <https://upload.wikimedia.org/wikipedia/commons/a/a7/Labyrinth-seal.png>, (Accessed 2017-07-28).