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Chapter 1

Introduction

High precision machines such as EUV wafer scanners and E-beam measurement systems require a high vacuum level. Contamination of this vacuum due to moving cables and bearings of the positioning stages within are an issue. An inverted planar motor solves this contamination issue but leads to a complex system due to position dependant commutation and a large number of coils \[7\]. Therefore an alternative stage design is made at MI-Partners which has a low degree of complexity and does not cause contamination of the vacuum.

1.1 Novel wafer positioning system

The novel design consists of a clean/precision vacuum and a dirty/non-precision vacuum which are separated by a wall. The wafer chuck floats inside the clean vacuum just above the wall. The design uses a Short Stroke-Long Stroke (SS-LS) stage configuration where the wafer chuck is the SS. The LS is placed underneath the wall and can be a conventional xy stage as no processing occurs in the dirty vacuum. The purpose of this vacuum is to minimize the load on the wall. The actuation between the SS and LS is done through the wall.

![Diagram of novel wafer positioning system]

Figure 1.1: Novel wafer positioning system
CHAPTER 1. INTRODUCTION

To achieve stable levitation of the chuck a controller is required. All six degrees of freedom (DoF) will be controlled to create an accurate positioning system. By limiting the displacement of the SS relative to the LS electrical commutation is avoided. Wafers of 450 mm, which are to become the next industry standard, are to be carried on the wafer chuck. This is quite a big increase from the 300 mm wafers which is today’s standard. Because the system is a maglev stage which is actuated through a wall the machine will be called through wall maglev stage (TWMS).

As with every novel concept there are some challenges to overcome. The main challenges of this novel concept are:

- Actuation through the wall which results in a large gap size
- Adequate measuring with no sensors on the chuck
- Starting the wafer stage when the SS is not above the LS

This research focuses on the first challenge, the actuation through the wall.

1.2 Problem statement

The through wall actuation will be done by separate actuators for separate directions. Even though in principle it is possible to make one actuator that can generate a force in multiple directions, it is chosen not to do so because it greatly complicates the design, while separate actuators are more versatile in application. The actuators that generate forces in x/y are called in-plane actuators (IPA) and the vertical actuators active gravity compensators (AGC). The minimal amount of AGCs that is required for stable levitation is three. However the SS will be actuated using four AGCs, because previous research at MI-Partners has shown that over-actuation can improve the performance by reducing the excitation of the torsional mode, which typically limits the performance of large chucks.

The main goal of this research is to come up with an electromagnetic design of the IPA and the AGC. These actuators will then be fabricated if they satisfy the specifications. The characteristics will be measured and compared to the simulations. The final goal of this research is to implement the actuators in a demonstrator. This demonstrator will be a simplified version of the TWMS. In this research the vacuum is ignored because a vacuum has no effect on the electromagnetics and also no vacuum will be applied in the demonstrator.

1.3 Outline

In chapter 2 the TWMS is described in more detail. Then the assumptions on the wafer chuck dimensions and a basic layout of the actuator positions is given. And
finally the specifications that are valid for both the IPA and the AGC are given.

In chapter 3 the first of the two actuators, the IPA, is described. An optimal design is calculated after which a much simpler version is proposed for fabrication. Six of these actuators are fabricated after which they are measured on a measurement setup. Then the performance is compared to the simulations. Finally a conclusion is made on whether the IPA is good enough for implementation in the demonstrator.

In chapter 4 the second actuator, the AGC, is described. The influence of the magnet dimensions is analysed after which a configuration using only stock magnets is picked. A coil is added to provide for the active force. Six of these actuators are fabricated and the performance is measured and compared to simulations. Then a conclusion is made on whether the AGC can be implemented in the demonstrator.

Chapter 5 first briefly describes the mechanical design of the demonstrator. After this the control architecture is given and the dynamic behaviour of the SS is analysed. Then a set-point is made and the current limitations of the system are given.

Finally in chapter 6 the concluding remarks and recommendations are given.
Chapter 2

Through wall maglev stage

In this chapter the layout of the TWMS is explained. Then the basic assumptions of the wafer chuck are given. At the end of this chapter the specifications which apply to both the IPA and the AGC are discussed.

2.1 Machine layout

The layout of the TWMS is shown in Figure 2.1. The objective of the system is to accurately position the SS wafer stage. To accomplish this the SS is actuated through the wall with Lorentz actuators that have their coil part on the LS and the magnets on the SS. The SS is measured in 6 DoF relative to a metrology frame which is connected with spring-dampers to the frame. The springs are used to place the eigenfrequency of the suspension mode as low as possible and the dampers are used to dampen it. This decoupling from the frame is required because the frame...
is excited by the LS actuator as well as the SS actuators. The SS controller will use the SS position information to control the SS actuators. The stiffness between the SS and the LS should be kept to a minimum because this stiffness will directly transfer any vibrations from the LS to the SS, deteriorating the performance. This is accomplished by using Lorentz actuators because these generally have a low stiffness. Gravity compensation will be applied because Lorentz actuators suffer from ohmic losses. The stiffness of the gravity compensation should also be kept to a minimum.

The LS consists of a conventional stacked XY stage. The measurement of the LS can be done relative to the SS or to the frame. In the demonstrator it will be done relative to the frame because this is simpler to implement. The frame has to be connected rigidly to the ground such that the excitation of the frame due to the actuator forces is limited.

### 2.2 Wafer chuck assumptions

Basic assumptions have to be made on the size and mass of the wafer chuck. The mass of the wafer chuck is an important factor for the AGCs and the size of the chuck limits the size of the actuators. The assumptions of the wafer chuck used in this research are shown in Figure 2.2. As explained in the introduction the stage has to be capable of carrying 450 mm wafers. Outer dimensions of 500 mm x 500 mm are assumed such that there is enough room for a 450 mm wafer. The thickness of the chuck is assumed to be 30 mm based on basic FEM calculations of the eigenfrequencies of a simple chuck layout. These calculations also showed that the mass of the chuck can be made below 10 kg excluding the magnets. The total mass of the magnets including their housing is assumed to be 10 kg resulting in a total mass of 20 kg. Aluminium is used as chuck material because it is light, cheap and easy to machine.

![Figure 2.2: Assumptions on the wafer chuck of the TWMS.](image)
2.3 Through wall actuation

As explained in the introduction the actuation of the TWMS will be done by IPAs and AGCs. In this section the factors that are important for both of these actuators are described. The first factor is the air gap which influences the efficiency of both actuators. Then the layout of the actuators on the chuck is explained and finally the specifications that apply to both the IPA and the AGC are described.

2.3.1 Air gap

The actuation of the SS through the wall results in a relatively large air gap for the actuators. The gap should be as small as possible because the magnetic field diminishes according to an inverse square law over the distance. An estimation of the minimum gap size is needed for designing the actuators. The gap size is dictated by four factors: the thickness of the wall, its deflection, mechanical tolerances and the spacing required to prevent contact of the SS and LS with the wall.

To attain a minimal gap size the sum of the wall thickness plus its deflection should be minimized. The minimal thickness of the wall is dictated by its deflection as strength is not an issue. The deflection of a fully clamped square plate under gravitational load can be calculated as follows [11]:

\[ w_{\text{max}} = 0.0138 \frac{\rho g L^4}{E t^2}, \]  

(2.1)

where \( \rho \) is the density, \( g \) the gravitational acceleration, \( L \) the length, \( E \) the young’s modulus and \( t \) the thickness. It turns out that the \( \rho/E \) factor is quite similar for a lot of common materials like glass, aluminium, titanium and steel. This factor of \( 4 \times 10^{-8} \text{ kgm/N} \) will therefore be used in the calculations. A square plate with sides of 1 m is assumed. The wall thickness plus deflection over the wall thickness is shown in Figure 2.3. The optimal wall thickness can be calculated as \( \frac{dw_{\text{max}}}{dt} = 0 \). This results in an optimal wall thickness of \( t = \sqrt{0.0276 \rho g L^4 E^{-1}} = 2.2 \text{ mm} \). However a full perfectly clamped plate was assumed, which is difficult to realize. Therefore a plate with a thickness of 3 mm will be assumed, while keeping the deflection at 0.6 mm.

A spacing above and underneath the wall of 0.5 mm should provide sufficient room to avoid contact. Finally worst case tolerances of 0.4 mm are added to come to a total gap size of 5 mm, see Table 2.1. An air gap of 5 mm will therefore be assumed in the design of the actuators.

If a smaller gap size is required material that has a lower \( \rho/E \) factor could be used, for example silicon carbide. Other options would be to use a slight pressure difference, to apply pretension or to support the wall using the LS.
Figure 2.3: Influence of the wall thickness on wall thickness plus deflection. An optimum is found for a wall thickness of 2.2 mm. However a wall thickness of 3.0 mm is assumed because the calculation assumes perfect clamping which is impossible.

Table 2.1: Contribution of the various parts to the total gap size.

<table>
<thead>
<tr>
<th>Part</th>
<th>Value (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spacing underneath wall</td>
<td>0.5</td>
</tr>
<tr>
<td>Wall deflection</td>
<td>0.6</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>3.0</td>
</tr>
<tr>
<td>Spacing above wall</td>
<td>0.5</td>
</tr>
<tr>
<td>Mechanical tolerances</td>
<td>0.4</td>
</tr>
<tr>
<td><strong>Total gap size</strong></td>
<td><strong>5.0</strong></td>
</tr>
</tbody>
</table>
2.3. THROUGH WALL ACTUATION

Figure 2.4: Layout of the actuators on the SS.

2.3.2 Actuator layout on chuck

The layout of the actuators on the chuck is important because it influences the way the eigenmodes are excited [13]. Especially in the over-actuation principle that is applied in the TWMS the location of the actuators plays a major role. The SS wafer chuck of 500 mm by 500 mm is divided into nine areas. The AGCs are placed in the corners because this limits the excitation of the torsion mode and saddle shape [3]. Four IPAs are placed in between the AGCs. The centre of the stage is left open and can be used for wireless energy transfer in the future.

2.3.3 Specifications of the through wall actuation

The specifications that are important for both of the actuators are shown in Table 2.2. The in-plane acceleration should be at least 5 ms\(^{-2}\) to make the system quick enough. The vertical acceleration is less stringent because the vertical displacement will be used for slower processes like focusing the process on the wafer. The in-plane stroke of ±1 mm is to allow for errors in the LS position. The voltage and current specifications are such that standard linear current amplifiers can be used.

The maximum stiffness between the SS and LS depends on the allowable transmission of vibrations. Assumed is a 20 Hz disturbance frequency \(f_d\) and a required reduction factor \(r\) of 100. The maximal allowable eigenfrequency of the suspension mode then becomes:

\[
f = \frac{f_d}{\sqrt{r}} = \frac{20}{\sqrt{100}} = 2 \text{ Hz} \quad (2.2)
\]

The maximum stiffness can now be calculated:

\[
k = (2\pi f)^2 m = 3 \text{ N/mm}, \quad (2.3)
\]
Table 2.2: Specifications that apply to both the IPA and the AGC.

where $k$ is the stiffness, $f$ the eigenfrequency and $m$ the mass of the chuck.

The required bandwidth of the controller is determined by vibrations in the metrology frame which are assumed to show acceleration levels $a$ of up to 1 mms\(^{-2}\). These accelerations have to be followed by the SS. The maximum error $e$ of this tracking is set at 10 nm after which the minimal bandwidth can be calculated as:

$$f_{bw} > \frac{1}{2\pi} \sqrt{\frac{a}{e}} = \frac{1}{2\pi} \sqrt{\frac{10^{-3}}{10^{-8}}} = 50 \text{ Hz}$$

(2.4)
Chapter 3

In-Plane Actuator

As was explained in Chapter 2, the in-plane actuation is done by four identical In-Plane Actuators (IPA). In this chapter, the design of this actuator is explained, the realisation is shown, and the measured characteristics are given.

3.1 Basic layout

The basic layout of the IPA is shown in Figure 3.1. The IPA consists of coils on the LS and magnets on the SS. The magnets will most likely be put in a Halbach array. A Halbach array is a special arrangement of permanent magnets that augments the magnetic field on one side of the array while cancelling the field on the other side. This configuration was first discovered by Mallinson in 1973 [8], but was named after Klaus Halbach who used such a configuration to focus accelerator particle beams [5]. A Halbach array seems a good choice for the IPA because it can make a relatively efficient actuator with little magnetic field above the chuck. Any magnetic field above the chuck might interfere with the process on the wafer and is therefore undesirable.

![Figure 3.1: Basic layout of the In-Plane Actuator (IPA).](image)
Table 3.1: Comparison between copper coils and flat wound aluminium coils. Flat would aluminium are better because of the higher fill factor and better thermal conductivity.

The IPA shall contain no iron on the LS side. Even though iron might help to aim the magnetic field it will always attract the SS downwards which is detrimental for a maglev stage. Iron can be however be applied on the magnet side of the actuator because this will not generate the attractive force between the LS and SS. This iron might help to aim the magnetic field and/or prevent the magnetic field from interfering with the process on the wafer.

3.1.1 IPA coils

For the IPA two coils types have been taken into consideration: copper coils and flat wound aluminium coils. A comparison between those two types is given in Table 3.1. Even though copper is a good thermal conductor a copper coil generally is not because of the insulation layers on the wires. Heat that is generated in the centre of the coil will have to go through several layers of insulation before reaching the outside of the coil. The aluminium flat wound on the other hand has just a single layer of insulation between the centre and the top and bottom. This results in a better heat conduction and with active cooling high current densities can be reached. The downside of the aluminium is that the resistivity is a factor 1.65 higher than copper which will increase the ohmic losses. The factor 3.3 lower density makes the aluminium coils a good choice is moving coil applications. Due to the use of flat sheets instead of round wires the fill factor of flat wound coils can reach about 0.9 instead of about 0.6 for round wired coils. The fill factor is important for the optimal utilization of the magnetic field.

Clearly the flat wound aluminium coils are the preferred choice. However for this research copper coils that are already available at MI-Partners will be used due to money and time constraints. The dimensions of these coils are shown in Figure 3.2 and the specifications in Table 3.2. The coils fit in the maximum dimensions and have a race track shape which is good for the IPA as a large portion of the coil can be used for actuation. Furthermore the fill factor of these coils is about 0.8 which is higher then typical because hexagonal winding is used [14]. Two of these coils will be placed side by side connected in series such that the current goes in the same direction where they touch. The two coils are required to generate sufficient force. The use of water cooling will be avoided because it is undesirable to have any water near the expensive electronics. This means that the system should be able to
operate without active cooling. Of course in an industrial application water cooling should be applied because the heat is detrimental for high precision positioning.

The limiting factor for force generation in the IPA is the heat generation in the coil due to ohmic losses. Because no water cooling is applied in the actuator the maximum continuous currents are limited by the conduction, convection and radiation of heat from the coils. No deep analysis of these effects will be conducted because in an industrial application one would surely apply water cooling. This is especially the case for a system in a vacuum because then only conduction and radiation remain.

An estimation of a safe current density has to be made because this is needed in the calculation of the IPA force. A surface area of 100 mm x 120 mm on the top and bottom is assumed to be the area which has to transfer the heat. The heat transfer is assumed to be dominated by the transfer between the IPA and the surrounding air. A radiation heat transfer coefficient $h_r$ of 5 W/m$^2$K is assumed and a convective heat transfer coefficient $h_c$ of 5 W/m$^2$K which is a low estimation for free convection to air \[10\]. The total heat transfer coefficient $h$ is therefore 10 W/m$^2$K. The heat that can be transferred when the operating temperature of the coil is 80$^\circ$C is then:

$$\dot{Q} = hAdT = 14.4 \text{ W}$$ \hspace{1cm} (3.1)

The maximum continuous current can now be calculated because the resistance of the coils is known:

$$I_{\text{max}} = \sqrt{\frac{\dot{Q}}{R}} = 1.5 \text{ A}$$ \hspace{1cm} (3.2)
### Table 3.3: Specifications of the IPA.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum dimensions x-y</td>
<td>167 x 167</td>
<td>mm²</td>
</tr>
<tr>
<td>Stroke x-y</td>
<td>±1</td>
<td>mm</td>
</tr>
<tr>
<td>Stroke z</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td>Air gap</td>
<td>5</td>
<td>mm</td>
</tr>
<tr>
<td>Force x</td>
<td>50</td>
<td>N</td>
</tr>
<tr>
<td>Parasitic force z</td>
<td>&lt; 10%</td>
<td>of force x</td>
</tr>
<tr>
<td>Mass on SS</td>
<td>2</td>
<td>kg</td>
</tr>
<tr>
<td>Maximum stiffness</td>
<td>0.5</td>
<td>N/mm</td>
</tr>
<tr>
<td>Maximum voltage</td>
<td>48</td>
<td>V</td>
</tr>
<tr>
<td>Maximum current</td>
<td>7</td>
<td>A</td>
</tr>
</tbody>
</table>

The current of 1.5 A corresponds with a current density of 2.5 A/mm² for the coil. A maximum current density of 5 A/mm² can be applied for one fourth of the time because doubling the current gives four times the heat generation. The heat capacity of the coils is about 175 J/K and with a maximum ohmic losses of 60 W the temperature would rise only with only 0.35 K/s. Therefore there is no risk of a quickly heating and only sustained application of the maximum current can cause problems. A maximum current density of 5 A/mm² will be assumed in all simulations because it is considered safe for short time operation.

### 3.2 Specifications

The design of the IPA has to meet certain specifications, see Table 3.3. Most of the specifications follow directly out of the through wall actuation specifications in Table 2.2. The total stiffness between the SS and LS was set at 3 N/mm. This stiffness is divided over four AGC and two IPA because the stiffness of an actuator like the IPA is almost non-existing in directions orthogonal to the actuation direction. This results in a maximum stiffness of 0.5 N/mm per IPA. The total mass of the magnets including their housing of 10 kg is divided into 8 kg for the IPAs and 2 kg for the AGCs, resulting in a maximum mass of 2 kg per IPA. The force in x is calculated by multiplying the acceleration of the stage with its mass divided by two IPAs. The parasitic force z should be less 10% of the force in x to provide sufficient decoupling.

### 3.3 Dimensioning the magnets

As explained the most likely design will be a Halbach array. Instead of just assuming such a configuration an optimization of the propulsion force is done by adjusting the magnetization directions. First the model in which this is done will be explained.
3.3. DIMENSIONING THE MAGNETS

The magnets will be dimensioned with the x position and z position in millimeters. The magnetization direction of these blocks will be optimized for maximum propulsion force. Figure 3.4 shows how a magnet with magnetization in any direction can be calculated using a horizontal and a vertical magnet.

After this the resulting optimal configuration is given and an abstraction to simpler designs is made. Finally a design which can be made using only stock magnets is analysed. In all simulations a $B_r$ of 1.4 T and a current density of 5 A/mm$^2$ are assumed. The depth of the array is taken as 100 mm such that the straight part of the coil is longer than the array.

3.3.1 Optimization model

For the optimization a 2D slice of the configuration is used as depicted in Figure 3.3. The magnetization of the magnet area is to be optimized. To do so the magnet area is divided into 1 mm$^2$ blocks of which each block has its own magnetization direction. For each block the magnetic field is calculated with horizontal and vertical magnetization using 3D analytic formulas. The forces in the coil are then calculated by taking the cross product between the current density and the magnetic field. The torques around (0,0) are calculated as well by taking the cross product of the forces and the position vector. Finally the total force and torque is calculated by numerically integrating the forces and torques over the coil. These forces correspond to the horizontal and vertical magnetized block; other magnetization directions can be calculated using combinations of the horizontal and vertical magnetized block. This is depicted in Figure 3.4. The forces and torque of each of the magnet blocks is calculated for 0 to 359 degrees in steps of 1 degree. The total calculation time to do this for all of the magnet block and all directions is under one minute.

The force and torque of the total magnet area is simply the sum of the forces and torques of the individual magnets blocks, choosing a magnetization direction for
CHAPTER 3. IN-PLANE ACTUATOR

Figure 3.5: Magnet configuration optimized for maximum propulsion force with 360 degrees magnetization freedom.

Figure 3.6: Contribution of each magnet block to the total propulsion force of 85.6 N for the configuration shown in Figure 3.5.

each block. This way of simulating the magnet array as a sum of its components is valid because the superposition principle holds under the assumption that the relative permeability is one. It would not be valid for systems which contain iron or when demagnetization becomes a problem.

3.3.2 Optimal configuration

The optimal configuration is the configuration for which the highest force in the actuation direction is observed. The magnet configuration for which this occurs is shown in Figure 3.5. The contribution of each of the 1 mm\(^2\) blocks is shown in Figure 3.6. The total force in x direction is 85.6 N, in z 0 N and the torque is 0.9 Nm. This optimal configuration is good for comparison of other configurations but cannot be manufactured in any sensible way. Therefore even simpler configurations will be analysed and compared to this configuration.
3.3. DIMENSIONING THE MAGNETS

3.3.3 Optimal configuration limited to eight magnetization directions

The optimal configuration for maximum propulsion force using eight magnetization directions instead of a continuously varying magnetization direction is shown in Figure 3.7. The contribution to the total propulsion force of 83.5 N are given in Figure 3.8. The torque went down from 0.9 Nm to 0.8 Nm. Only 2.5% of the propulsion force is lost and this array can be manufactured, but 13 complicated custom magnets are needed. Therefore simpler configurations will be analysed.

3.3.4 Optimal configuration limited to horizontal and vertical magnetization

If the configuration is limited to only magnets with horizontal and vertical magnetization it can be manufactured relatively easy compared to the previous configurations, but still custom magnets are needed. The configuration with a maximum propulsion force under this limitation is shown in Figure 3.9. The corresponding propulsion force contributions are given in Figure 3.10. The force for this configur-
CHAPTER 3. IN-PLANE ACTUATOR

Figure 3.9: Magnet configuration optimized for maximum propulsion force using horizontal and vertical magnetization.

Figure 3.10: Contribution of each magnet block to the total propulsion force of 77.7 N for the configuration shown in Figure 3.9.

ation is 77.7 N, in z 0 N and the torque is 0.8 Nm. The force went down by 9% compared to the optimal configuration.

### 3.3.5 Configuration using only stock magnets

A configuration that can be fabricated using only stock magnets is shown in Figure 3.11. The force contribution for this magnet configuration is shown in Figure 3.12. The total force in x direction for this configuration is 70.8 N, in z 0 N and the torque is 0.0 Nm. The propulsion force went down by 17% compared to the optimal configuration, which is still acceptable. In this configuration clear discontinuities are visible in the propulsion force contribution graph which depict areas where the magnetization direction is far from optimal. A comparison between the different magnet configurations is given in Table 3.4.
3.3. DIMENSIONING THE MAGNETS

Figure 3.11: Comparable magnet configuration that can be made using just stock magnets.

Figure 3.12: Contribution of each magnet blocks to the total propulsion force of 70.8 N for the configuration shown in Figure 3.11. The contribution of each 15 mm x 15 mm magnet block is given underneath the magnets for comparison.

<table>
<thead>
<tr>
<th></th>
<th>$F_x$ [N]</th>
<th>$F_z$ [N]</th>
<th>$T_y$ [Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimal full freedom</td>
<td>85.6</td>
<td>0</td>
<td>0.9</td>
</tr>
<tr>
<td>Optimal eight directions</td>
<td>83.5</td>
<td>0</td>
<td>0.8</td>
</tr>
<tr>
<td>Optimal four directions</td>
<td>77.7</td>
<td>0</td>
<td>0.8</td>
</tr>
<tr>
<td>Stock magnets</td>
<td>70.8</td>
<td>0</td>
<td>0.0</td>
</tr>
</tbody>
</table>

Table 3.4: Total forces for the various configurations (2d calculation).
CHAPTER 3. IN-PLANE ACTUATOR

Figure 3.13: Final magnet array using sixteen stock magnets.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width</td>
<td>15 mm</td>
</tr>
<tr>
<td>Height</td>
<td>15 mm</td>
</tr>
<tr>
<td>Length</td>
<td>50 mm</td>
</tr>
<tr>
<td>Tolerances</td>
<td>±0.1 mm</td>
</tr>
<tr>
<td>Material</td>
<td>NdFeB</td>
</tr>
<tr>
<td>Coating</td>
<td>Nickel-plated (Ni-Cu-Ni)</td>
</tr>
<tr>
<td>Magnetisation</td>
<td>N48</td>
</tr>
<tr>
<td>Article ID</td>
<td>Q-50-15-15-N</td>
</tr>
</tbody>
</table>

Table 3.5: Specifications of the IPA magnets.

3.4 Final configuration

The final magnet configuration is shown in Figure 3.13. To make the array out of just one type of stock magnets the array consists of sixteen magnets instead of five. The array consists of two identical rows of eight magnets of which each row contains three pairs of magnets that are combined to create bigger magnets. The configuration includes a gap between the two rows to prevent demagnetization during manufacturing as will be explained in section 3.5. The specifications of the magnets are shown in Table 3.5. These magnets have a very high magnetization classified as N48 which means the maximum energy product of this magnet is 48 MGOe. This corresponds with a remnant flux density $B_r$ of 1.37 T to 1.42 T. The strongest magnets available at this moment are N52, but they are not so common.

3.4.1 Back iron

Adding back iron to magnets is a common way to increase the force. A big drawback in a moving magnet actuator is that iron has a high density. To see whether back iron should be added to the actuator 2D finite elements simulations have been conducted in FEMM [9]. The magnetic field lines of the configuration with and without back iron are shown in Figure 3.14. The first thing that should be noted
3.4. **FINAL CONFIGURATION**

Figure 3.14: The magnetic field lines of the array with and without back iron. Slightly less field exists above the array when the back iron is implemented.

<table>
<thead>
<tr>
<th></th>
<th>$F_x$ [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>No back iron</td>
<td>71.3</td>
</tr>
<tr>
<td>With back iron</td>
<td>70.6</td>
</tr>
</tbody>
</table>

Table 3.6: Total propulsion force with and without back iron. Obviously back iron will not be implemented.

is that the magnetic field is not entirely symmetric. This is due to magnetic field generated by the coils, which is very weak compared to the field of the magnets.

A comparison between the propulsion force with and without back iron is given in Table 3.6. Instead of increasing the force the force is decreased a bit by adding back iron. This can be explained by the fact that the magnets that have a horizontal magnetization are now ‘shorted’ on their top side, leaving less field on the strong side. No back iron will be applied because it is obviously a bad choice.

### 3.4.2 3D simulation of the final configuration

To check the behaviour of the final configuration 3D simulations are conducted. In 3D the forces and torques in all directions can be calculated and the array can be displaced in all six DoF. The model for the 3D simulations is based on the same 3D analytical formulas as in the optimization but now the forces are integrated over all straight coil parts in 3D, see Figure 3.15. The corners are omitted because the influence of these is small.

First the effect of the gap that was introduced in section 3.4 will be investigated. The forces of the configuration with and without gap are given in Table 3.7. The configuration with the gap generates a 1% lower propulsion force which is considered acceptable because the gap solves the demagnetization issues.

The influence of displacement in all directions is also simulated. The results of these simulations are given in Table 3.8. The first thing that should be noted is that the torques are quite small even when the the actuator is displaced 1 mm. Furthermore it is clear that $F_x$ depends most on $z$ while $F_z$ depends most on $x$. 

CHAPTER 3. IN-PLANE ACTUATOR

Figure 3.15: 3D model of the IPA using 3D analytical formulas. Numerical integration is done over the straight parts of the coil.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Without gap (a)</td>
<td>68.4</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
</tr>
<tr>
<td>With gap (b)</td>
<td>67.7</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Table 3.7: Forces and torques for the configuration with and without the gap.

The force and stiffness characteristics of the $F_x$ versus $x$ are given in Figure 3.16. The maximum stiffness of the IPA in x direction is about 0.35 N/mm at maximum displacement, well within the specification of 0.5 N/mm. The parasitic force $F_z$ versus displacement $x$ is shown in Figure 3.17a. The maximum parasitic force falls within the specification of $< 10\%$ of $F_x$. Finally torque $T_y$ versus displacement $x$ is shown in Figure 3.17b. The influence of this torque is neglected as it is small compared to the forces, and the direction is such that is even slightly helps the actuation underneath the centre of gravity.

It seems that $F_z$ depends linearly on x but actually it behaves like negative the sine of x which can be seen when a larger displacement is taken, see Figure 3.18. The $F_x$ force behaves like the cosine of x. This can be explained by the fact that the horizontal and vertical magnetic field produced by an infinite Halbach array are the sine and cosine of the position. On the left and right side of the graph the forces diminish due to the end effects of the array. This is no problem for the IPA because the maximum relative displacement between the SS and LS is 1 mm.

Finally the influence of rotations is analysed. In the TWMS rotations will be small: below 1 mm over the 500 mm width of the SS, corresponding to 2 mrad. A rotation of 5 mrad is simulated in all directions and the results are given in Table 3.9. Clearly the influence of the rotations is very limited and will therefore be neglected.
### 3.4. FINAL CONFIGURATION

<table>
<thead>
<tr>
<th>Position ((x \ y \ z) [\text{mm}])</th>
<th>(F_x [\text{N}])</th>
<th>(F_y [\text{N}])</th>
<th>(F_z [\text{N}])</th>
<th>(T_x [\text{Nm}])</th>
<th>(T_y [\text{Nm}])</th>
<th>(T_z [\text{Nm}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>((0 \ 0 \ 5))</td>
<td>67.7</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
</tr>
<tr>
<td>((0 \ 0 \ 6))</td>
<td>63.2</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>-0.01</td>
<td>0.00</td>
</tr>
<tr>
<td>((1 \ 0 \ 5))</td>
<td>67.5</td>
<td>0.00</td>
<td>-4.49</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
</tr>
<tr>
<td>((-1 \ 0 \ 5))</td>
<td>67.5</td>
<td>0.00</td>
<td>4.49</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
</tr>
<tr>
<td>((0 \ 1 \ 5))</td>
<td>67.6</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>-0.05</td>
</tr>
<tr>
<td>((0 \ -1 \ 5))</td>
<td>67.6</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>0.05</td>
</tr>
</tbody>
</table>

Table 3.8: Forces and torques for various positions of the final configuration. Displacement in \(x\) and \(z\) have a large influence while the displacement in \(y\) has little influence, just as expected.

![Graph 1](image1.png)

(a) Propulsion force

![Graph 2](image2.png)

(b) Propulsion stiffness

Figure 3.16: Propulsion force (a) and the stiffness (b) versus displacement in propulsion direction \(x\).

![Graph 3](image3.png)

(a) Parasitic force

![Graph 4](image4.png)

(b) Torque \(y\)

Figure 3.17: Parasitic force in \(z\) and torque \(y\) versus displacement in propulsion direction \(x\).
CHAPTER 3. IN-PLANE ACTUATOR

Figure 3.18: $F_x$ and $F_z$ versus large displacement in $x$. It can be seen that the forces are spatially shifted sine functions of the position. The amplitude decreases to the sides because of the end effects of the array. When operating only the middle of the graph will be used because $F_x$ is almost constant and $F_z$ is small.

Figure 3.19: Propulsion force versus increasing gap size. It can be seen that the propulsion force quite linear in the over the gap size. This makes compensation for this effect very straightforward.
### 3.5 Construction and Realisation

During construction of the array the magnets will move past each other in ways that might cause demagnetization. This demagnetization of the IPA magnets can happen due to strong demagnetizing fields. In Appendix C the demagnetization curve of N48 magnet material is shown. The irreversible demagnetization occurs for demagnetizing fields which are strong enough to push the curve over the knee at the left side.

The way to construct the array with the least demagnetization would be to first make the rows according to Figure 3.20 by bringing the magnets to each other in the -x direction. This way the strongest demagnetizing field always occurs when the magnet is in its final position. Then to produce the total array two of these rows have to be brought to each other in similar fashion, such that again the strongest demagnetizing field occurs in the final position. This way of manufacturing is optimal for the magnets but it will not be used because it is difficult to create a tool that can do this safely.

A simpler way would be to build the two rows simultaneously as is shown in Figure 3.21. In this way of constructing the even numbered magnets slide past the odd numbered magnets. This sliding introduces a high demagnetizing field in both magnets. This field becomes much higher than $H_{cB}$ and $H_{cJ}$ in the corners of the magnet and will therefore cause demagnetization there. There are two relatively easy methods to avoid this demagnetization. The first method is to use magnets with a higher $H_{cJ}$, for example N48H which has a $H_{cJ}$ of 41% higher than N48. The other method is to leave a small gap between the magnet rows. This method is chosen and a gap of 5 mm is applied because this is enough to prevent the demagnetization.

#### Table 3.9: Forces and torques for various rotations of the final configuration. The minimal influence of 5 mrad rotation shows that the rotations can be neglected.

<table>
<thead>
<tr>
<th>Rotation</th>
<th>$F_x$ [N]</th>
<th>$F_y$ [N]</th>
<th>$F_z$ [N]</th>
<th>$T_x$ [Nm]</th>
<th>$T_y$ [Nm]</th>
<th>$T_z$ [Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_x$ 5 mrad</td>
<td>67.7</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>-0.02</td>
<td>-0.02</td>
</tr>
<tr>
<td>$R_y$ 5 mrad</td>
<td>67.7</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>$R_z$ 5 mrad</td>
<td>67.7</td>
<td>0.29</td>
<td>0.00</td>
<td>-0.02</td>
<td>0.05</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Figure 3.20: Constructing the rows with the least probability of demagnetization by bringing in one magnet at a time from the right. Two of these should then be brought together in a similar fashion without any magnets sliding past each other.
The magnet array is build up in an aluminium housing glueing one magnet at a time. The glue used is a two component epoxy UHU plus Endfest 300 which has a working time of two hours and a final strength of 12 MPa when hardened at room temperature. 12 MPa is enough for this application because the highest expected load is about 0.2 MPa. The magnets are too strong to put in place by hand (350 N repulsion/attraction depending on orientation) so a tool was made to aid in the construction of the array. This magnet array fabrication tool is shown in Appendix A. The tool makes placing the magnets and keeping them in place during curing of the epoxy a simple task. One array takes about two hours to fabricate and another 24 hours is needed for the curing. The tool turned out to be very good because the arrays were built quick and safely. A total number of six IPA’s were fabricated but only four are needed for the demonstrator so there are two spare ones. The magnet array and coil assembly are shown in Figure 3.22.
3.6 Measurements

To confirm whether the IPAs generate the forces that are required they have to be measured. For this measurement a test setup was build which can measure the relative position and the forces simultaneously. The relative position is measured using six eddy current sensors and the forces are measured using a six DoF force cell. The actuator is connected to a current amplifier which is controlled by a xPC system. The position and forces information is sent to this xPC system as well. The measurement setup is shown in Appendix B.

The position of the array relative to the coil has been varied over -1 mm to +1 mm in steps of 0.5 mm in x and y direction. The gap has been varied between 5 mm and 6 mm in steps of 0.5 mm to come to a total of 75 measurement points per actuator. The measurements were done by manually moving the stages to the right position and then start a current of 2.5 A (slightly lower then the 3 A in simulations). After that this current was reversed just to check whether the maximum force is the same in both directions, which of course turned out to be the case. Then the current was turned off and the array was moved to a new position for the next measurement.

The forces can be shown in a 3D vector plot as is done in Figure 3.23. It can be seen that the force points in the right direction but this gives little insight in the exact characteristics nor the amplitude of the force. Therefore 2D plots just as in the simulation are given. The propulsion force versus displacement in the propulsion direction is shown in Figure 3.24. Simulations of the forces with average and minimal
remnant flux density for N48 magnet material have been included. The actuators show about 3% less force than in simulation. This slight difference can be explained by the fact that the simulations did not include the relative permeability of the magnets which slightly reduces the force. IPA 3 seems to be relatively bad but this is most likely due to an error in nulling of the sensors during this measurement, resulting in a wrong position. The stiffness of the IPA turns out to be low (< 0.3 N/mm) just as the simulations predicted.

The second important characteristic is the parasitic force in the z direction. The results of this force versus the displacement in x direction are shown in Figure 3.25. The forces turn out to be very comparable between the different actuators and the simulation as well. The maximum parasitic z force is below 4 N. In simulation changing the gap size results in quite large changes of the propulsion force. This is of course confirmed in measurements because the magnetic field diminishes quickly over distance. The propulsion force versus the gap size is shown in Figure 3.26.

The torques turn out to be small just as was the case in the simulations, see Figure 3.27. Also here different behaviour is observed for IPA 3, most likely due to bad positioning. The torques are negligibly small compared to the forces.

The IPAs work just as in simulation and therefore no problems are expected for the implementation in the demonstrator. For the implementation IPA1 will not be used because it does not have a protective cover while all others do. The characteristics
3.6. MEASUREMENTS

Figure 3.25: Parasitic $z$ force versus displacement for all six IPA compared to the simulations for a current of 2.5 A.

Figure 3.26: Propulsion force versus gap size for all six IPA compared to the simulations for a current of 2.5 A.
Figure 3.27: (a) Torque $x$ versus displacement $x$. (b) Torque $y$ versus displacement $x$. (c) Torque $z$ versus displacement $y$. The torques in all directions are low and will not cause any problems.
of the IPA2, IPA4, IPA5 and IPA6 are most alike and will therefore be used in the demonstrator.

3.7 Summary

An optimal design for maximum propulsion force of the IPA was investigated. A simplified version using only stock magnets was proposed. The loss of propulsion force is about 17% compared to the optimal design. A 5 mm gap is made between the two magnet rows because this prevents demagnetization during fabrication. Six IPAs were manufactured and the characteristics were checked on a measurement setup. All specifications are met and therefore the IPAs are suitable for implementation in the demonstrator.
Chapter 4

Active Gravity Compensator

In this chapter the design of the AGC is explained. This actuator has to compensate the gravity of the SS and generate an active force (AF). The gravity compensating force will be generated by repulsive magnets and the AF by the Lorentz force. The AF is needed to stabilize the system, compensate the remaining gravity, compensate parasitic forces and for vertical acceleration.

4.1 Basic layout

The basic layout of the AGC consists of a ring magnet, a disc magnet and a coil as depicted in Figure 4.1. The combination of magnets is used for gravity compensation (GC) and the coil is used to generate the AF.

![Figure 4.1: Basic layout of the AGC. The AGC consists of a ring magnet on the SS and a disc magnet and coil on the LS. The design of the AGC is fully axisymmetric.](image-url)
CHAPTER 4. ACTIVE GRAVITY COMPENSATOR

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum dimensions on SS (x,y,z)</td>
<td>(160,160,20)</td>
<td>(mm,mm,mm)</td>
</tr>
<tr>
<td>Stroke x and y</td>
<td>±1</td>
<td>mm</td>
</tr>
<tr>
<td>Stroke z</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td>Air gap</td>
<td>5</td>
<td>mm</td>
</tr>
<tr>
<td>GC force</td>
<td>45</td>
<td>N</td>
</tr>
<tr>
<td>Maximum vertical stiffness $k_z$</td>
<td>0.5</td>
<td>N/mm</td>
</tr>
<tr>
<td>Maximum radial stiffness $k_r$</td>
<td>0.5</td>
<td>N/mm</td>
</tr>
<tr>
<td>Active Force</td>
<td>±30</td>
<td>N</td>
</tr>
<tr>
<td>Mass on SS</td>
<td>&lt;0.5</td>
<td>kg</td>
</tr>
<tr>
<td>Maximum voltage</td>
<td>48</td>
<td>V</td>
</tr>
<tr>
<td>Maximum current</td>
<td>7</td>
<td>A</td>
</tr>
</tbody>
</table>

Table 4.1: Specifications of the AGC.

4.2 Specifications

The specifications of the AGC are shown in Table 4.1. The most important specifications of the AGC are the force of the GC and the maximal AF. If these specifications are not met the system cannot function properly. The required force of the gravity compensation is determined by dividing the gravity of the stage over the four AGC multiplied by a factor 0.9 to make sure the SS will always come down when the power is turned off. This results in a gravity compensating force of 45 N per AGC.

The AF specification is based on the active force budget, see Table 4.2. This force budget incorporates all the worst case disturbances that have to be compensated by the AF. The first disturbance is due to the fact that the IPAs actuate 0.02 m below the centre of gravity of the SS. For straight acceleration a compensating torque has to be produced by the AGCs which are about 0.35 m apart. The maximum force that will be needed is calculated as follows:

\[ T = F_x h = 65 \text{ N} \cdot 0.02 \text{ m} = 1.3 \text{ Nm} \]  
\[ F_a = \frac{T}{d} = \frac{1.3 \text{ Nm}}{0.35 \text{ m}} = 4 \text{ N} \]  

The second disturbance is the parasitic z force of the IPA for which a compensating force of 6 N is needed per actuator worst case. The third part is due to the fact that the gravity compensation will only compensate about 90% of the gravity leaving about 5 N. Due to tolerances and uncertainties of the magnets this is increased to 10 N to be on the safe side. And finally a force of 10 N per AGC is needed for an acceleration of 2 m/s\(^2\) of SS in the vertical direction. The total required AF is the sum of the disturbances: 30 N.

To attain a low stiffness the stiffness of the GC plus the inherent stiffness of the AF during normal operation should be minimal. The AF has an inherent positive
### 4.2. SPECIFICATIONS

<table>
<thead>
<tr>
<th>Origin of force</th>
<th>Worst case</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque compensation</td>
<td>4 N</td>
</tr>
<tr>
<td>Parasitic z force IPA</td>
<td>6 N</td>
</tr>
<tr>
<td>Remaining gravity</td>
<td>10 N</td>
</tr>
<tr>
<td>Vertical acceleration</td>
<td>10 N</td>
</tr>
<tr>
<td>Total</td>
<td>30 N</td>
</tr>
</tbody>
</table>

Table 4.2: Active force budget of the AGC.

Figure 4.2: Required force characteristics for zero stiffness AGC. The total force is the sum of the GC force and AF. Zero stiffness occurs where the negative stiffness of the GC is equal to the positive stiffness of the AF.

The radial stiffness is directly linked to the vertical stiffness according to Earnshaw’s theorem [4]. In 1939 Braubek worked out the practical consequences of this theorem [2]. For a system with $\mu_r = 1$ the sum of the stiffness in the three principle directions is zero. Because the AGC has an axisymmetric design the stiffness in $x$ and $y$ are equal to the radial stiffness $k_r$; one can therefore write:

\[
k_x + k_y + k_z = 0 \tag{4.3}
\]

\[
k_r = k_x = k_y = -\frac{k_z}{2} \tag{4.4}
\]

This shows that a low vertical stiffness will result in an even lower radial stiffness.
CHAPTER 4. ACTIVE GRAVITY COMPENSATOR

4.3 Gravity compensation

Although a small negative vertical stiffness is preferred for the GC first a zero stiffness GC is to be designed, after which it can be tuned to the small negative stiffness.

The most simple way of creating a GC system with magnets is just to take two equally sized opposing magnets as in Figure 4.3a. Unfortunately this is not a good way to make a GC system because the stiffness keeps increasing for a decreasing gap. This effect results in a relatively high stiffness. To avoid this high stiffness a slightly different design using three magnets can be applied, see Figure 4.3b. The idea behind this design is that when the magnet is in the middle the positive and negative stiffness cancel out and a zero stiffness results. The force at this position in not zero because the upper and bottom magnet both generate a force in the same direction on the middle magnet. More zero-stiffness GC designs can be found in literature [6, 16].

The zero stiffness magnet configurations that are available are good for most applications but unfortunately cannot be applied in the TWMS. The problem is that the wall prohibits any material connected to the LS to reach above the wall. A design that consists of a part on the SS and a part on the LS with no interconnections is required. Therefore a new zero-stiffness design is proposed which is the combination of a ring and a disc magnet.

The zero-stiffness effect of this configuration can be explained by looking at the two disc-disc combinations, see Figure 4.4. The force characteristics of these disc-disc combinations and the sum of these is shown in Figure 4.5. The zero-stiffness results
4.3. GRAVITY COMPENSATION

Figure 4.4: The ring - disc configuration can be thought of as the sum of two disc - disc combinations. The force characteristics are shown in Figure 4.5.

Figure 4.5: The force characteristics of the configurations shown in Figure 4.4. Zero stiffness behaviour is observed around a gap of 17 mm. This is the result of the increasing negative stiffness of \( F_2 \). A slight difference between \( F_3 \) and \( F_1 + F_2 \) exists because the relative permeability of the magnets is 1.05 instead of 1.

From the fact that the positive stiffness of the first configuration is nearly constant while the negative stiffness of the second configuration increases when closing in. At about 17 mm this negative stiffness becomes just as large as the positive stiffness of the first configuration and a net zero stiffness results.

It should be noted that the splitting of the configuration into two disc-disc configurations is only valid under the assumption that the relative permeability of the magnets is one. Because this is not entirely true the directly calculated characteristic of the force \( F_3 \) is also given for comparison. There is just a slight difference between \( F_3 \) and \( F_1 + F_2 \) showing that summing of the configurations is an accurate way of calculating the total configuration.
4.4 Dimensioning the magnets

In this section the magnets of the AGC will be dimensioned by analysing the influence of varying the dimensions. The magnets should be dimensioned such that the force is about 45 N at the peak, which is the location of the zero stiffness. The gap between the magnets at the zero stiffness is not important because the disc magnet can be placed inside the LS if needed. The only limitation is that the gap should be at least 5 mm.

All magnets are simulated as grade N42 because this is the most common grade used. This corresponds with a $B_r$ of 1.3 T and a relative permeability $\mu_r$ of 1.05 is assumed. Simulations are done using axisymmetric FEM in FEMM \[9\] in combination with MATLAB. The calculation time for these kind of simulations is in the order of a few seconds per position which makes it possible to calculate a lot of configurations in a short time period. In this simulation only vertical displacements are allowed and therefore only the vertical stiffness can be calculated. The radial stiffness follows from equation 4.4. The magnets will be dimensioned using these simulations and thereafter the behaviour for radial displacements will be investigated using other methods.

4.4.1 Influence of the hole diameter

First the influence of diameter of the hole of the ring magnet is investigated. As was shown in the summation of the two disc-disc configuration the hole is main reason for the zero stiffness behaviour to show up. A bigger hole should give more negative stiffness behaviour and a smaller hole less. The hole is simulated as a magnet in the opposite direction just as was done in Figure 4.4. The diameter of the hole is varied between 5 mm and 40 mm to see what is the influence. The disc is kept constant during all simulations to make a fair comparison. The dimensions are given in Figure 4.6.

The force characteristics are given in Figure 4.7. The negative force increases for an increasing hole size. Furthermore the negative stiffness indeed increases with the hole size as was expected.

4.4.2 Influence of the ring diameter

The influence of the diameter of the ring is another important factor. The dimensions used to analyse this influence are shown in Figure 4.8. The ring is modelled without the hole just like was done in the first part of Figure 4.4.

The resulting force characteristics are given in Figure 4.9. It turns out that zero-stiffness behaviour can already be observed for large disc sizes, even without the hole. However this zero-stiffness appears at a too small gap size and therefore cannot be
4.4. DIMENSIONING THE MAGNETS

Figure 4.6: The hole is simulated as an opposing magnet with a diameter equal to the hole diameter. This diameter is varied from 5 mm to 40 mm to investigate the influence it has on the configuration.

Figure 4.7: Force versus gap size for various hole diameters as depicted in Figure 4.6.
4.4.3 Most promising configurations

Out of the 72 possible combinations of hole and ring diameters a selection of the most promising i.e. with a large low stiffness area has been made. The force characteristics of these are given in Figure 4.10.

4.4.4 Influence the magnet thicknesses

Because the characteristics of the most promising configurations do not yet meet the specification of a 45 N peak the thickness of the magnets has to be adjusted. Adjusting the magnet thickness should adjust the total force while keeping similar zero-stiffness behaviour. Both the ring and disc thicknesses are varied to analyse the influence.

The ring with a hole of 15 mm and a outer diameter of 80 mm is taken to see the effect of changing the thickness of the ring magnet. The thickness of the ring magnet is varied between 5 mm and 10 mm.

The influence of the thickness of the disc magnet is also analysed. The dimensions used in the analysis are shown in Figure 4.13. The thickness of the disc magnet is varied between 20 mm and 40 mm.

Indeed the maximum force can be adjusted by adjusting the thickness of either the ring or disc magnet. Decreasing the thickness of either magnet decreases both the stiffness and force. Furthermore it displaces the zero-stiffness position slightly further away.

If no coil is applied the obvious choice would be to decrease the thickness of the
4.4. DIMENSIONING THE MAGNETS

Figure 4.9: Force versus gap size for various ring diameters as depicted in Figure 4.8.

Figure 4.10: Force versus gap size for the most promising combinations of hole and ring diameter.
CHAPTER 4. ACTIVE GRAVITY COMPENSATOR

Figure 4.11: Gap [mm]

Figure 4.12: Force versus gap size for various ring thicknesses as depicted in Figure 4.11.
4.4. DIMENSIONING THE MAGNETS

Disc thickness

\[ 40 \text{ mm} \quad 10 \text{ mm} \]

Gap Force

\[ 80 \text{ mm} \]

Figure 4.13

Force versus gap for various disc thicknesses [mm]

Figure 4.14: Force versus gap size for various disc thicknesses as depicted in Figure 4.13.
CHAPTER 4. ACTIVE GRAVITY COMPENSATOR

<table>
<thead>
<tr>
<th></th>
<th>Disc magnet</th>
<th>Ring magnet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter hole</td>
<td>-</td>
<td>28.5 mm</td>
</tr>
<tr>
<td>Diameter outside</td>
<td>45 mm</td>
<td>85 mm</td>
</tr>
<tr>
<td>Height</td>
<td>30 mm</td>
<td>8 mm</td>
</tr>
<tr>
<td>Tolerances</td>
<td>±0.1 mm</td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>NdFeB</td>
<td></td>
</tr>
<tr>
<td>Coating</td>
<td>Nickel-plated (Ni-Cu-Ni)</td>
<td></td>
</tr>
<tr>
<td>Grade</td>
<td>N45 35M</td>
<td></td>
</tr>
<tr>
<td>Article ID</td>
<td>S-45-30-N</td>
<td>SALE-030</td>
</tr>
</tbody>
</table>

Table 4.3: Final magnet combination for the AGC. Both magnets are supplied by Supermagnete.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter inside</td>
<td>70 mm</td>
</tr>
<tr>
<td>Diameter outside</td>
<td>110 mm</td>
</tr>
<tr>
<td>Height</td>
<td>20 mm</td>
</tr>
</tbody>
</table>

Table 4.4: Dimensions of the coil in the AGC.

ring magnet because that would reduce the mass on the wafer chuck. When a coil is applied the ring magnet has a second task which is to generate a magnetic field in the coil. Then the choice is not obvious and depends on the AF that has to be made.

4.5 Final configuration

Just as for the IPA no custom magnets will be made for the AGC. Instead a configuration of stock magnet of similar dimensions as the most promising configurations was found. The specifications of the ring and disc magnet are given in Table 4.3. The force characteristics of this magnet combination is shown in Figure 4.15 and the stiffness in Figure 4.16. The effect of the tolerances on the magnetization of the magnets is included. This tolerances are not expected to give any problems because the maximum force difference is only ±3%. The zero-stiffness lies at a gap of 17 mm and low stiffness behaviour is observed over a large displacement. The small negative stiffness just left of the zero stiffness peak will be used to compensate the stiffness of the AF.

To generate the AF a coil is needed on the LS. A current running through this coil in combination with the radial part of the magnetic field produced by the ring magnet generates a vertical force. Figure 4.17 shows a simulation in FEMM of the magnetic field generated by the ring magnet. Only the field of the ring magnet is simulated because the field of the disc magnet can only generate internal forces between the coil and the disc. The coil dimensions are given in Table 4.4 and the force and stiffness characteristics in Figure 4.18. The force specification of 30 N at a gap of 5 mm is met. The final design of the AGC is shown in Figure 4.19.
4.5. FINAL CONFIGURATION

Figure 4.15: Force versus gap size for the final magnet combination including the influence of the tolerance of the magnet grade. The tolerance in the magnet grade results in a maximal change in force of \( \pm 3\% \).

Figure 4.16: Stiffness versus gap size for the final magnet combination including the influence of the tolerance of the magnet grade. Low stiffness behaviour is observed for several millimetre displacement from the zero stiffness point. The tolerance in the magnet grade has little effect on the stiffness.
Figure 4.17: Magnetic field of the ring magnet. The coil is placed where the field has the largest radial component because this generates the vertical force. A gap of 5 mm between the ring magnet and the coil is taken. A force of 33 N can be made using the depicted coil under the assumption of a current density of 5 Amm$^{-2}$.

Figure 4.18: (a) Force and (b) stiffness characteristics of the AF. The force specification of 30 N at a gap of 5 mm is met.
4.5.1 3D simulations

Using equation 4.4 one can estimate the radial stiffness. However this does not show whether this stiffness changes rapidly when the magnet displaces radially. Therefore the influence of a radial displacement is investigated using a 3D FEM package and MacMMems [12]. MacMMems produces semi-analytical models which can be used by the CADES framework [15]. MacMMems is designed for the calculation of MEMS but it turns out it can also be used to simulate the AGC. Simulations take just one second per position, compared to a few minutes for 3D FEM. The drawback of the analytic formulations is that the relative permeability of the magnets is assumed unity.

The first analysis is $F_x$ over $x$ at a gap of 16 mm. The results are given in Figure 4.20. According to equation 4.4 there should be a positive stiffness of about 0.15 N/mm because the vertical stiffness is -0.3 N/mm as can be seen in Figure 4.16. This indeed turns out to be true for the simulation in MacMMems but not in 3D FEM because 3D FEM includes a relative permeability of 1.05. The important conclusion however is that the stiffness remains low over a large displacement.

The effect of a small rotation around $y$ is given in figure 4.21. This should not give any problems as the maximum rotation will be below 1 mrad. Furthermore the torque resulting from these small rotations is negligible.

4.6 Construction and realisation

The realization of the AGC is much simpler than that of the IPA because no magnets have to be brought near each other. Therefore also demagnetisation will not happen.
Figure 4.20: Force x versus displacement x at a gap of 16 mm. The force and displacement are defined positive in the same direction therefore the negative slope represents a positive stiffness. The slight difference in results is due to the fact that 3D FEM takes $\mu_r = 1.05$ into consideration.

Figure 4.21: Force x versus a rotation around y at a gap of 16 mm.
4.7 Measurements

For all of the six AGC’s that were fabricated the gravity compensating force and AF have been measured. The measuring is done on the same setup that was used for measuring the IPA. For the measurements the radial displacement was set to zero and the peak was tuned to be at a gap of 6 mm. The gap size was varied between 0 mm and 20 mm in steps of 1 mm. At each position three measurements were done: no current and 5 A and -5 A. The results of the measurements are given in Figure 4.23 to 4.25.

The force is about 5% higher than in simulation. The average peak lies at 47 N compared to about 45 N in simulation. This could be due to the fact that the alignment between the ring and disc magnet was not perfect, which results in a higher force. The AF indeed shows positive stiffness behaviour as expected and the tuning of the zero stiffness peak to 6 mm results in an almost zero total stiffness at a gap size of 5 mm. The measurements are in good agreement with the simulations and all specifications are met. Therefore the AGCs will be implementated in the demonstrator and no major issues are expected.
Figure 4.23: Measured force over gap size for $I = 0$ A. The peak is tuned to be at a gap of 6 mm.

Figure 4.24: Measured force over gap size for $I = 5$ A.
4.8 Summary

A low stiffness AGC system has been made out of a ring magnet, a disc magnet and a coil. The influence of the magnet dimensions was analysed after which a combination of stock magnets was chosen. This configuration meets the specifications and is this is confirmed by measurements. All specifications are met and therefore the AGC will be implemented in the demonstrator.
Chapter 5

Demonstrator

To demonstrate the potential of the TWMS concept a demonstrator has been realized. The goal of this demonstrator is not to reach nanometre precision but to show the concept. In this chapter the design, control strategy and the results are described.

5.1 Mechanical design

The design of the demonstrator consists of the SS, the LS, the wall, the metrology frame and the force frame as depicted in Figure 5.1.

The SS is a 450 mm x 450 mm x 30 mm wafer chuck, see Figure 5.2. The chuck is milled out of a single block of aluminium such that no internal fasteners are required. It contains the magnet arrays of the IPA and the ring magnets of the AGC.

The LS is driven by a spindle stage which allows for displacement in one direction, see Figure 5.3. It can be upgraded to a stacked xy configuration if necessary. The stroke is limited to 100 mm because of size constraints of the demonstrator. The LS is actuated by a Parvex DC servo motor of type RS240B. This motor is equipped with a brake as well as an encoder. Although this motor comes equipped with an encoder a more accurate optical linear encoder is used to measure the LS.

For the demonstrator the wall is 3 mm thick and made out of a transparent plastic to keep the LS visible.

The metrology frame houses all the sensors that measure the SS position. The metrology frame is placed on passive rubbers which results in an eigenfrequency of 13 Hz. The main component of the metrology frame is milled out of a single block of aluminium just like the wafer chuck. The top of the metrology frame is closed using an aluminium plate connected with twenty mechanical fasteners to provide for stiffness.
CHAPTER 5. DEMONSTRATOR

Figure 5.1: The mechanical layout of the demonstrator (a) and an image of the system (b).

Figure 5.2: An image of the bottom side of the wafer chuck. The ring magnets of the AGC’s can be clearly seen. The magnet arrays of the IPA’s are protected by a 0.2 mm thick stainless steel cover.

Figure 5.3: Bottom (a) of the LS and (b) the fully assembled LS.
5.2 Control architecture

The SS is controlled using a decoupled controller around its centre of gravity, see Figure 5.4 Therefore six SISO controllers have to be tuned instead of one MIMO controller which reduces the complexity substantially. The SISO control will be done using tamed PD-controllers, also called lead-lag compensators. Integral action can be added to improve the loop gain at low frequencies, resulting in better steady state behaviour.

The global to local matrix is used to transform the global actuation forces originating from the controller to local actuator forces. This matrix is constant because the position of the actuators relative to the centre of gravity does not change, see Figure 5.5 This is not the case for the decoupling matrix which transforms the sensor information into the displacement of the SS. Therefore an adapting decoupling matrix has been implemented that changes according to displacements in the LS direction.

For tuning the controllers it is important to have good measurements of the dynamics of the plant. Unfortunately the plant is unstable without control and the dynamics
cannot be measured upfront. Therefore the parameters of the PD controllers were tuned based on the SS mass and actuator constants placing the bandwidth at about 20 Hz. These parameters turned out to be good enough to attain stable levitation after which measurements of the plant could be conducted. The frequency response function (FRF) of the plant in y and z direction is given in Figure 5.6 and 5.7; the other directions and can be found in Appendix D. The directions are as given in Figure 5.1.

In y direction the plant shows a long -2 slope which is the mass line of the SS. At about 13 Hz the influence of the metrology frame suspension mode can be seen. The result of this suspension mode is that the decoupling becomes bad especially in the x and y direction as can be seen in Figure D.2. The excitation of the first eigenmodes of the SS is low because the excitation is in plane while the mode is out of plane. Finally at about 900 Hz the bandwidth limiting eigenfrequency is observed.

In z direction the result of the over-actuation becomes apparent in the fact that the first two eigenmodes are excited only mildly. For the first eigenmode this is due to the fact that the contributions of the AGC cancel each other. For the second
5.3. SETPOINT

Figure 5.7: FRF of the SS in z (vertical) direction.

eigenmode the actuators lie in the nodal line and therefore do not excite it. The third eigenmode is excited strongly because the actuators are not placed in its nodal line. The low stiffness behaviour of the AGCs is confirmed by the fact that the eigenfrequency of the suspension mode lies below 1 Hz. It therefore lies well below the specification of 2 Hz, resulting in good decoupling from LS vibrations.

The controllers have been tuned to a 50 Hz bandwidth as can be seen in Figure 5.8. The corresponding phase and gain margins are given in Table 5.1. The LS is also controlled using a PD controller and is currently tuned to a 20 Hz bandwidth.

5.3 Setpoint

A third order setpoint moving the SS and LS together over the full stroke has been made, see Figure 5.9. The servo error of the LS and SS in y direction are given in Figure 5.10a. It is clear that the assumption of 1 mms$^{-2}$ is not valid for the current metrology frame as vibrations in the order of a few micrometre show in the SS error
Figure 5.8: Open loop transfer function of the SS. The controller is tuned to a 50 Hz bandwidth in all directions. The phase and gain margins for all directions are given in Table 5.1.
5.4 SUMMARY

<table>
<thead>
<tr>
<th>Direction</th>
<th>Phase margin (deg)</th>
<th>Gain margin (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>x</td>
<td>40.3</td>
<td>11.9</td>
</tr>
<tr>
<td>y</td>
<td>43.8</td>
<td>12.7</td>
</tr>
<tr>
<td>z</td>
<td>33.5</td>
<td>8.4</td>
</tr>
<tr>
<td>Rx</td>
<td>30.9</td>
<td>7.0</td>
</tr>
<tr>
<td>Ry</td>
<td>33.0</td>
<td>7.8</td>
</tr>
<tr>
<td>Rz</td>
<td>38.9</td>
<td>9.87</td>
</tr>
</tbody>
</table>

Table 5.1: Phase and gain margins for all directions at a 50 Hz bandwidth.

during the setpoint. Currently the maximum acceleration of the stage is $1.5 \text{ m/s}^2$ which is limited by the LS actuator as can be seen in Figure 5.10b.

5.4 Summary

The concept of a through wall maglev stage has been successfully implemented in a demonstrator. Currently the design has only one long stroke direction, but a stacked XY stage can easily be added to the system. The controller is tuned to a bandwidth of 50 Hz in all directions, meeting the specifications. The FRF shows the stiffness between the short and the long results in a suspension mode below 1 Hz, which results in good decoupling from LS vibrations. Currently the acceleration of the stage is limited to $1.5 \text{ m/s}^2$ by the LS motor.
Figure 5.10
Chapter 6

Conclusions and recommendations

The development of the TWMS has led to the development of two new actuators and a demonstrator. For each of these the conclusions are summarized.

6.1 Conclusions

6.1.1 IPA

The current design of the IPA provides a propulsion force of 65 N at a gap of 5 mm and a current of 3 A. The stiffness is zero when the SS is right above the LS and increases to 0.35 N/mm at a displacement of 1 mm. The parasitic force in z is also zero at centre position and becomes ±4.5 N at a displacement of 1 mm. The torques are small in all directions with a maximum of 0.05 Nm, expect for IPA 3 which most likely was a measurement error. The IPA meets all specifications and is therefore implemented in the demonstrator.

6.1.2 AGC

The AGC generates 45 N gravity compensation and 33 N active force with a current of 5 A. The stiffness of the gravity compensation is currently tuned to -0.3 N/mm which results in a net zero stiffness in combination with the active force during steady state. A zero vertical stiffness also implies a zero radial stiffness according to Earnshaw’s theorem. The AGC meets all specifications and is therefore implemented in the demonstrator.

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6.1.3 Demonstrator

The functioning demonstrator shows that the through wall actuation of a wafer chuck is possible. The 20 kg 450 mm wafer chuck can be accelerated with $6.5 \text{ ms}^{-2}$ in plane and with at least $2 \text{ ms}^{-2}$ vertically without active cooling. Currently a bandwidth of 50 Hz has been obtained in all directions which meets the specifications. The implemented one directional LS also works as predicted, and an upgrade to a two directional LS is possible. Currently the acceleration of the system is limited by the maximum acceleration of the LS, which is about $1.5 \text{ ms}^{-2}$. A 30 N mismatch between the gravity of the stage and the gravity compensation results in unnecessary heat dissipation in the AGC. Although this currently does not cause any heat issues better matching would be preferred.

6.2 Recommendations

The demonstrator acceleration limitation of $1.5 \text{ ms}^{-2}$ can be increased to $6.5 \text{ ms}^{-2}$ by taking a stronger LS actuator. For higher accelerations the force of the IPA has to be increased. A 10% increase in force can be obtained by taking the magnet design with custom magnet sizes. For a further increase of the force water cooling can be applied to the IPA coils, increasing the maximum current and therefore increasing the force.

Better matching between the gravity compensation and the mass of the stage will reduce the steady state currents of the AGC. Currently the mass of the SS is about 3 kg higher than the gravity compensation, resulting in unnecessary heat dissipation. An interesting solution would be an AGC with an adjustable force at the zero stiffness peak. A design that can accomplish this is shown in Figure 6.1. In this design the disc magnet is composed of several magnets which can be added or removed depending on the required force. Also a spacer can be implemented in between these magnets to reduce the force.

Improvement of the over-actuation can be made by placing the AGCs on the nodal line of the umbrella mode. This reduces the excitation of the umbrella mode, making higher bandwidths possible. This would however bring the AGC and IPA closer to each other which might introduce crosstalk. It is recommended that this crosstalk is investigated after which the improvement can be implemented.

An interesting safety feature can be implemented which shorts the leads of the IPAs in case of emergency. This generates high damping between the SS and the LS because the magnets will induce currents in the coils, dissipating energy.
Figure 6.1: A new AGC design with adjustable force at the zero stiffness peak.
Bibliography


Appendix A

Halbach array production tool

For the production of the Halbach arrays of the IPA a tool was made because the magnets are too strong to put in by hand. The tool is shown in Figure A.1. It takes about 2 hours to construct the array and another 24 hours is needed for the curing of the epoxy. Finally after removing excess glue a thin protective cover is glued to the array, see Figure A.2.

Figure A.1: (a) The Halbach array production tool. (b) Result after 24 hour curing.

Figure A.2: A completed Halbach array with a protective cover.
Appendix B

Actuator measurement setup

For measuring both the IPA and the AGC a measurement setup was built, see Figure B.1. On this setup the actuator can be displaced in x, y and z direction using three manual positioning stages. The force induced by the actuator is measured using a force/torque sensor which measures all forces and torques simultaneously. The relative position is measured in six DoF using eddy current sensors. A xPC system is used for acquiring the forces, torques and relative position, as well as for setting the output of the current amplifier.

The mechanical design uses 10 mm thick aluminium plates to provide for sufficient stiffness such that the maximum deflection is limited to a few micrometre. Furthermore no magnetic material is placed close to the actuator because this would influence the measurements. This includes the stages, the force sensor and the eddy current sensors which all contain ferromagnetic material.

Figure B.1: The setup used to measure the IPA and the AGC.
Appendix C

Demagnetization curve of N48 magnets

Figure C.1: The demagnetization curve of N48 magnets. For modern magnet material like this one a near linear behaviour of J(H) and B(H) is observed until a certain value of H for which a knee is observed. If the magnets working points are located within this linear area, they can be moved up and down the demagnetization curve without permanent demagnetization. The behaviour of the magnet is then called to be reversible.
Appendix D

Demonstrator FRF

In this appendix the FRF of the Short Stroke are given. The directions are as given in Figure 5.1. In Figure D.1 the full FRF of the plant is given after which it is split up in four 3 by 3 functions. Finally the diagonal terms are shown including the phase.
Figure D.1: Frequency response function of the Short Stroke.
Figure D.2: Top left part FRF of the Short Stroke.
Figure D.3: Top right part FRF of the Short Stroke.
Figure D.4: Bottom left part FRF of the Short Stroke.
Figure D.5: Bottom right part FRF of the Short Stroke.
Figure D.6: FRF of the Short Stroke in x direction.
Figure D.7: FRF of the Short Stroke in y direction.
Figure D.8: FRF of the Short Stroke in z direction.
Figure D.9: FRF of the Short Stroke in Rx direction.
Figure D.10: FRF of the Short Stroke in Ry direction.
Figure D.11: FRF of the Short Stroke in Rz direction.
Abstract

This thesis deals with the through wall actuation of the Through Wall Maglev Stage (TWMS). The TWMS is an alternative design for the inverted planar motor [7]. It consists of a clean/precision vacuum and a dirty/non-precision vacuum which are separated by a wall. The wafer chuck floats inside the clean vacuum just above the wall. The design uses a Short Stroke-Long Stroke (SS-LS) stage configuration where the wafer chuck is the SS. The actuation of the SS is done through the wall using Lorentz actuators with the coils on LS. The LS is placed underneath the wall and can be a conventional xy stage as no critical processes occur in this vacuum.

The through wall actuation is done using separate Lorentz actuators for separate directions. The actuators that generate forces in x/y are called in-plane actuators (IPA) and the vertical actuators active gravity compensators (AGC). Four IPA and four AGC together over-actuate the chuck. The over-actuation is used to improve the performance by reducing the excitation of the torsional mode, which typically limits the performance of large chucks [3]. The air gap of the actuators is 5 mm due to the thickness of the wall, while 0.5 mm is typical.

The IPA consists of a magnet array on the SS and a pair of coils in series on the LS. A magnet design for maximum force has been designed which generates a force of 85.6 N at a current of 3 A. A much simpler design using sixteen identical stock magnets is proposed which generates 67.7 N. Six of these actuators have been manufactured and the characteristics of each of them have been measured. The actuators show about 3% less force than in simulation, which can be explained by the fact that the simulations did not include the relative permeability of the magnets. The IPA meets all specifications is therefore implemented in a functional demonstrator of the TWMS concept.

The AGC consists of a ring magnet on the SS and a disc magnet plus coil on the LS. The magnets are used for zero (low) stiffness gravity compensation and the coil is used to generate the active force. Low stiffness is important because it directly influences the transmission of vibrations from the LS to the SS. The magnet pair has been analysed to find a configuration with a low stiffness over a large displacement. The inner and outer diameter of the ring play an important role in attaining the zero stiffness behaviour while the thickness of the magnets dictates the magnitude of the force. For the fabrication of the AGC stock magnets with a 45 N peak force and a low stiffness over a large displacement are selected. A coil that can generate
a 33 N active force is added. Six AGC are manufactured and the characteristics of each of them have been measured. The zero stiffness point is confirmed in the measurements and the specifications are met.

The four IPA and AGC with the most similar characteristics are implemented in the demonstrator. The working demonstrator confirms the feasibility of the TWMS concept. The LS is implemented as spindle stage which works as predicted and an upgrade to a stacked xy stage is possible. The SS is measured using five eddy current sensors and a laser interferometer in the LS direction. The SS is controlled using a decoupled controller around its centre of gravity. A bandwidth of 50 Hz has been attained in all directions using tamed PD controllers. The potential of the through wall actuators has been demonstrated.