



Design of an active magnetic bearing system for a high-speed permanent magnet synchronous machine

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SUMMARY

Currently, there are two high-speed permanent magnet synchronous machines (HS-PMSMs), nicknamed the TWINS, situated in the McTronX laboratory at the North-West University. However, excessive heat generated by the ball bearings of the motors causes the rotor to overheat. This could cause the permanent magnet on the rotor to demagnetize. The heat is generated by excessive friction caused by high-speed as well as improper alignment of the bearings.

The main objective of this project is to develop an active magnetic bearing (AMB) system to replace the ball bearings of the TWINS with AMBs, in order to reduce the heat generated in the rotor. This is to allow the HS-PMSMs to be operated at a rated speed of 30,000 r/min. The secondary objective is to add knowledge and experience to the McTronX research group on AMBs.

This project will focus on the design of the AMB system; including a feasibility study to replace the ball bearings with AMBs, the design of the electromagnets of the AMBs, as well as the retrofit that was done to the current HS-PMSMs to allow them to be operated on AMBs. Static tests will be done to confirm the design of the AMBs. However, the AMBs will only dynamically be tested in later projects as the rotor still need to be balanced, and the control of the HS-PMSM still need to be designed and implemented before the rotor can be operated at its rated speed.

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LIST OF SYMBOLS

Symbol	Description	Unit
b	Damping	N·s/m
c	Viscous damping coefficient	N·s/m
c_c	Critical viscous damping coefficient	N·s/m
e	Eccentricity	m
e_{per}	Permissible eccentricity	m
E	Young's modulus	Pa
f	Linear frequency	Hz
F	Force	N
F_{ub}	Unbalance force	N
g	Acceleration due to gravity	m/s ²
I	Area moment of inertia	m ⁴
I_b	Biasing current	A
J	Mass moment of inertia	kg·m ²
k	Spring constant	N/m
K	Kinetic energy	J
l, l_i	Length	m
m, m_i	Mass	kg
M	Bending moment	N·m
N	Rotational speed	r/min
s	Laplace variable	
t	Time	s
T	Kinetic energy	J
U_{per}	Permissible unbalance	kg·m
U	Potential energy	J
v	Linear velocity	m/s
w	Distributed weight of mass	N/m
W	Weight of mass	N
x, y, z	Cartesian coordinates, displacements	m
$\dot{x}$	Velocity in x -direction	m/s
$\ddot{x}$	Acceleration in x -direction	m/s ²
θ	Angular displacement	rad
$\dot{\theta}$	Angular velocity	rad/s
ρ	Mass density	kg/m ³
ρ	Resistivity	$\Omega \cdot m$
σ	Stress	N/m ²
τ	Period of oscillation, Time constant	s
α	Thermal expansion coefficient	$\mu m/m \cdot ^\circ C$
ω	Frequency of oscillation, Angular velocity	rad/s
ω_n	Natural frequency	rad/s
ω_d	Frequency of damped vibration	rad/s

LIST OF ABBREVIATIONS

<i>HS-PMSM</i>	-	<i>High-speed permanent magnet synchronous machine</i>
<i>HS</i>	-	<i>High-speed</i>
<i>PMSM</i>	-	<i>Permanent magnet synchronous machine</i>
<i>AMB</i>	-	<i>Active magnetic bearing</i>
<i>PM</i>	-	<i>Permanent magnet</i>
<i>EM</i>	-	<i>Electromagnet</i>
<i>PA</i>	-	<i>Power amplifier</i>
<i>CM</i>	-	<i>Centre of mass</i>
<i>FOS</i>	-	<i>Factor of safety</i>
<i>ADC</i>	-	<i>Analog to digital convertor</i>
<i>DAC</i>	-	<i>Analog to digital convertor</i>
<i>Mmf</i>	-	<i>Magnetic motive force</i>

CHAPTER 1: INTRODUCTION

This project aims to replace the ball bearings of a high-speed (HS) permanent magnet synchronous machine (PMSM) situated in the McTronX research laboratory of the North-West University with active magnetic bearings (AMBs). This chapter will discuss the problem that lead to the decision to replace the ball bearings with AMBs, as well as an introduction on HS-PMSMs and AMBs will be provided. The issues to be addressed, research methodology, as well as the document overview, will also be presented.

1.1 BACKGROUND

The McTronX research group is doing research on HS-PMSMs. Currently, there are two HS-PMSMs nicknamed the TWINS that was intended for this research. Figure 1-1 shows an image of the TWINS.

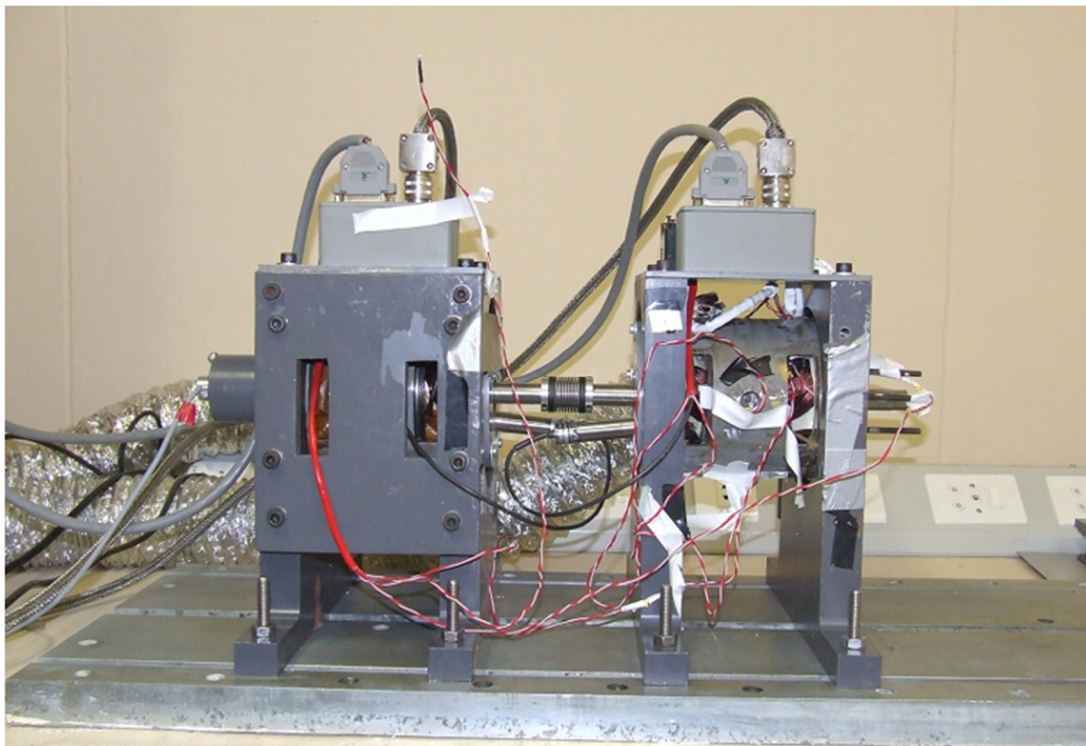


Figure 1-1: TWINS

However, if the machines are operated for a continuous period of time, the rotor overheats, which can cause the permanent magnet (PM) on the rotor to demagnetize. According to previous research, this is mainly due to the heat generated by the TWINS's ball bearings. The losses in the bearings were approximated to be 75 W per bearing at a rated speed of 30'000 r/min [1]. The rotor design has also determined that it should not be operated above 80 °C as this can cause the material shrink-fitted to the rotor's shaft to experience mechanical failures [2]. To solve this problem, it was decided that one of the TWINS will be modified to use active magnetic bearings. This is to allow for relative motion between the rotor and stator, without generating excessive heat. To make this possible, magnetic material needs to be added to the existing rotor of a PMSM. Figure 1-2 shows the existing rotor of the TWINS.

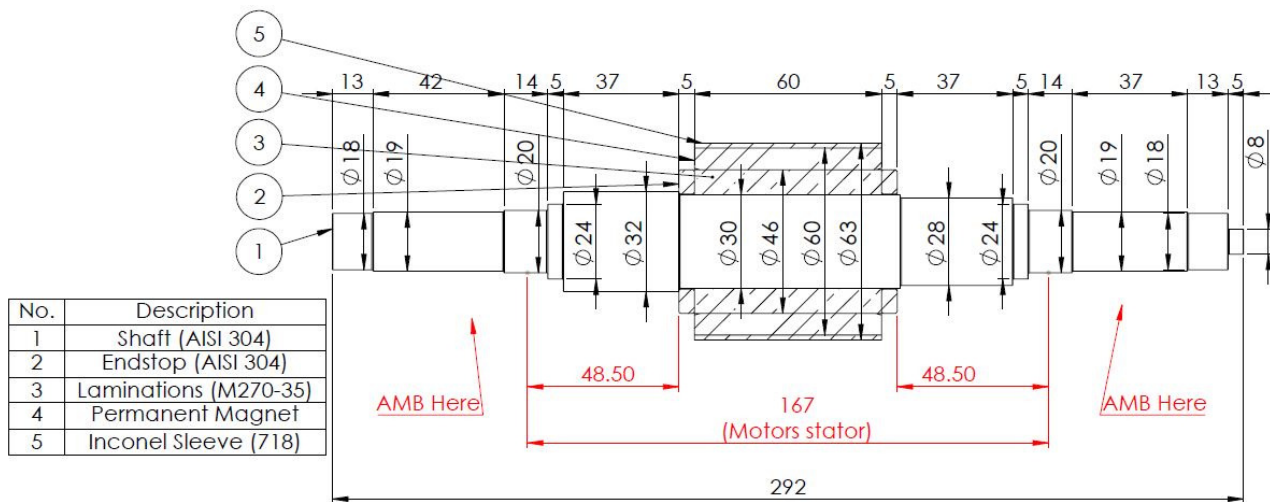


Figure 1-2: TWINS rotor as is

The rotor consists of the shaft, laminations to enhance the magnetic flux of the permanent magnet, two endstops to keep the laminations from buckling, a permanent magnet to enable the motor to rotate at synchronous speed, and an Inconel sleeve to protect the PM from breaking while the rotor rotates at 30,000 r/min. The axial space that is occupied around the rotor by the motor's stator is shown in Figure 1-2 as (*Motor's stator*). This space cannot be used to add magnetic material to the rotor. The only available space is outside the motor's stator indicated by "AMB Here" on the figure.

1.1.1 High-speed permanent magnet synchronous machines

Synchronous machines are widely used in power systems to generate electricity. However, HS-PMSMs have a few advantages over low-speed synchronous machines. Permanent magnet machines have the advantage that they have a smaller size, higher power density, higher efficiency and lower acoustic noise. While an increase in the operating speed of an electrical machine increases its power density. However, a major challenge with PM machines is to retain the permanent magnet at high operation speed. This is because PMs are very brittle and cannot withstand high centripetal forces. Another critical factor is the rotor's critical frequencies. If the rotor's operating speed is close to its critical frequencies, the rotor will start resonating and become unstable. For this reason, the rotor design is a very important aspect to consider when designing a HS-PMSMs. The rotor's diameter is limited by its operating speed, while its length is limited by its first bending critical frequency [3],[4].

In an electrical machine, its power density increases as its operating speed increases. The smaller size and higher power density of an HS-PMSM causes thermal problems, particularly in its rotor. This is because the PM on the rotor can be demagnetized if its temperature rises to excessive levels. To prevent this from happening the losses in the rotor needs to be minimized. One way to do this is to replace ordinary ball bearing with AMBs [3],[5].

Although ceramic ball bearings could have been a solution as it is generally a high-speed low resistance type bearing, it would not work in the TWINS' case. This is because one of the problems that caused the excessive heat generated in the TWINS' bearings was, the perpendicular alignment of the bearing housings to the rotor's shaft. This caused misalignments in the bearings' inner and other races, causing excessive heat to be generated.

1.1.2 Active magnetic bearings

Active magnetic bearings work, by levitating the rotor using electromagnets situated around the rotor. It levitates the rotor with an airgap of typically 0.5 mm between the rotor and stator of the AMBs. Figure 1-3 shows the functional diagram of an active magnetic bearing system. AMBs work by measuring the rotor's position within the AMBs' stator with a high precision proximity sensor. This position information is used by a controller to control the power amplifiers (PA) that drive the electromagnets. Magnetic material is added to the rotor to enable the electromagnets to attract the rotor, as many shafts are made of non-magnetic materials. The power amplifiers provide sufficient power to the control signal given by the controller, as controllers' output power is generally very low. The controller stabilizes the rotor's position by controlling the attracting force on the rotor with the electromagnets. This counteracts any forces that try pulling the rotor away from its desired position, as well as minimizing vibrations caused by, unbalance and external factors. Axial AMBs are used to control the rotor's axial position. However, the TWINS' rotor does not have axial space available to attach an axial AMB, as will be shown in Chapter 3.9. For the time being the axial displacement of the rotor will be controlled by the other TWIN connected to it with a coupling, keeping in mind that the second TWIN still has ball bearings that resist axial motion.

As AMBs are mainly used for high-speed applications, any failure in the system, causing the rotor to delevitate at high-speed, can cause irreparable damage to the machine. For this reason, backup bearings need to be installed, to prohibit the rotor from coming into direct contact with the stator [6], [7]. Axial backup bearings are used with axial AMBs to control the rotor's position in its axial direction

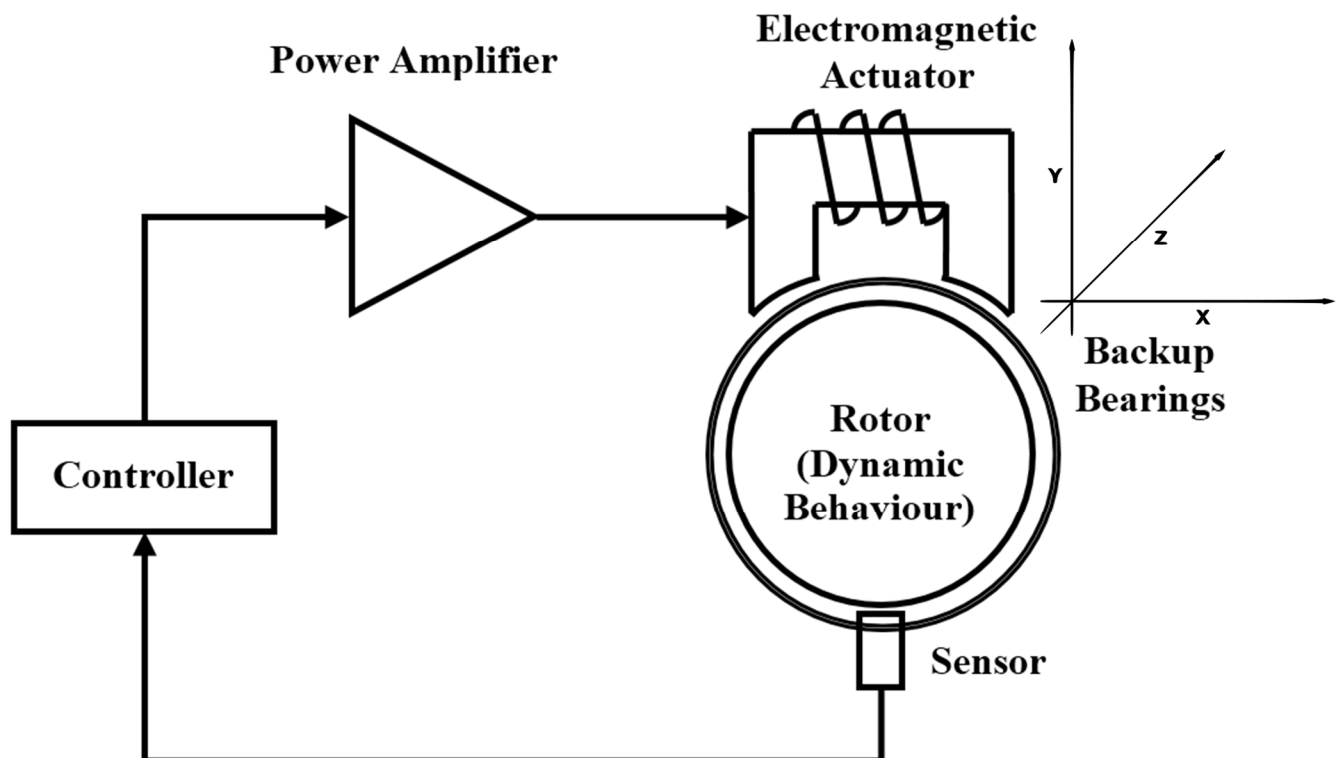


Figure 1-3: AMB functional diagram [8]

1.2 PROBLEM STATEMENT

The TWINS cannot be operated for extended periods, due to excessive heat generated in their rotors. This can cause the permanent magnet on the rotor to demagnetize, or cause mechanical failure of the components shrink-fitted to the shaft. The design, analysis and implementation of an AMB system need to be performed to resolve this issue.

1.3 ISSUES TO BE ADDRESSED

The six aspects that need to be considered when designing the AMB system are: the electromagnetic actuator design, rotor dynamic behaviour, the backup bearings, proximity sensors, the controller and the power amplifiers. Due to the multidisciplinary nature of the components mentioned, not all of them need to be designed and can be sourced instead. The issues that need to be addressed will be discussed in the following subsections.

1.3.1 System specifications

Before the AMB can be designed a set of specifications are needed. These include:

- The rotor should be able to rotate at 30'000 r/min.
- The rotor temperature is not allowed to exceed 80 °C.
- The AMBs' actuators should be able to produce a minimum force, that is determined from a rotor dynamic point of view.
- The rotor and stator of the TWINS shouldn't be redesigned. It should be modified to accommodate the AMBs.

1.3.2 Electromagnet design

The electromagnets need to be designed to be able to meet the system specifications. There exist different topologies of electromagnets that can be used to levitate the rotor. These includes radial, axial and conical magnetic bearings. The one that will be implemented depends on the orientation of the rotor and the amount of axial space available on the rotor. Of the radial magnetic bearing there exist two topologies namely a homopolar and a heteropolar configuration, with each having its own advantages and disadvantages. These topologies will be further explained in section 2.1. After the topology is specified, the number of poles, journal size, coil parameters and stator can be designed.

1.3.3 Rotor dynamic analysis

Although the rotor will not be completely redesigned, a rotor dynamic check needs to be done. This is to ensure that the modification done to the rotor does not change its dynamic behaviour in such a way as to cause the motor to become unstable at its operational speed. The minimum force, stiffness and damping that needs to be produced by the electromagnets is determined by a rotor dynamic analysis. The first critical bending frequency needs to be determined from the rotor dynamics to ensure that, no instabilities are excited by the rotation of the rotor. For this, the first critical bending frequency needs to be sufficiently higher than the maximum operating frequency of the motor.

1.3.4 Fitting the AMB's journal to the rotor

A method needs to be determined to fit the journals of the AMBs to the rotor. This method should enable the rotor to rotate at its designed operation speed of 30'000 r/min. The method that seems the most attractive is to shrink-fit the journal to the rotor, as this was how the existing rotor was

assembled. An analysis will need to be done to ensure that the stresses in the material are low enough and that the journal will not lose contact with the shaft rotating at 30'000 r/min.

1.3.5 Stator modification

Components need to be designed that can be fitted to the existing motor stator of the TWINS, to enable the machine to operate with AMBs.

1.3.6 Power amplifiers

The power amplifiers will not be redesigned. It was decided that power amplifiers available from a previous project will be used if possible. It should however, be determined if the power amplifiers meet the required specifications. This will be determined from the electromagnetic design.

1.3.7 Proximity sensors

There are two Micro-epsilon (DT3701-U1-A-C3) proximity sensors available for use. It should be determined if these sensors meet the required specifications. If they do, a minimum of two other proximity sensors need to be sourced. This is because for a purely radially suspended rotor a minimum of four proximity sensors are needed, 2 for the x and y -position at one end of the rotor and two for the x and y -position at the opposite end.

1.3.8 Controller

To control the AMBs it was decided to use a dSPACE® controller available in the McTronX laboratory. The dSPACE® controller allows for real-time control implementation. The software integrates with Simulink® and MATLAB® to allow for easy implementation and modelling of control systems.

1.3.9 Electrical interface

An electrical interface should be implemented between the sensors and the dSPACE® A/D converter. This is to protect the controller from any overvoltages.

1.4 SYSTEM EVALUATION

After the AMB system have been designed and implemented, it should be verified that the implemented AMBs produces, the desired force, stiffness and damping. The force that the AMBs can produce will be verified by using a spring scale to measure it. The stiffness will be verified by using a spring scale to apply a force on the rotor and measuring the displacement of the rotor from its centre, this can be used to calculate the stiffness. The stiffness and damping will also be measured by subjecting the rotor's reference position to a step input and using its response to calculate the AMBs' stiffness and damping. The test results should be validated to determine how well they correlate with the designed values.

1.5 RESEARCH METHODOLOGY

The research methodology that was followed to solve the presented problem is shown in Figure 1-4, and will be explained in this section.

The first step is to identify the problem that needs to be solved. This will identify the fields that needs to be researched. For this project: a solution must be found to solve the overheating problem that the TWINS are facing.

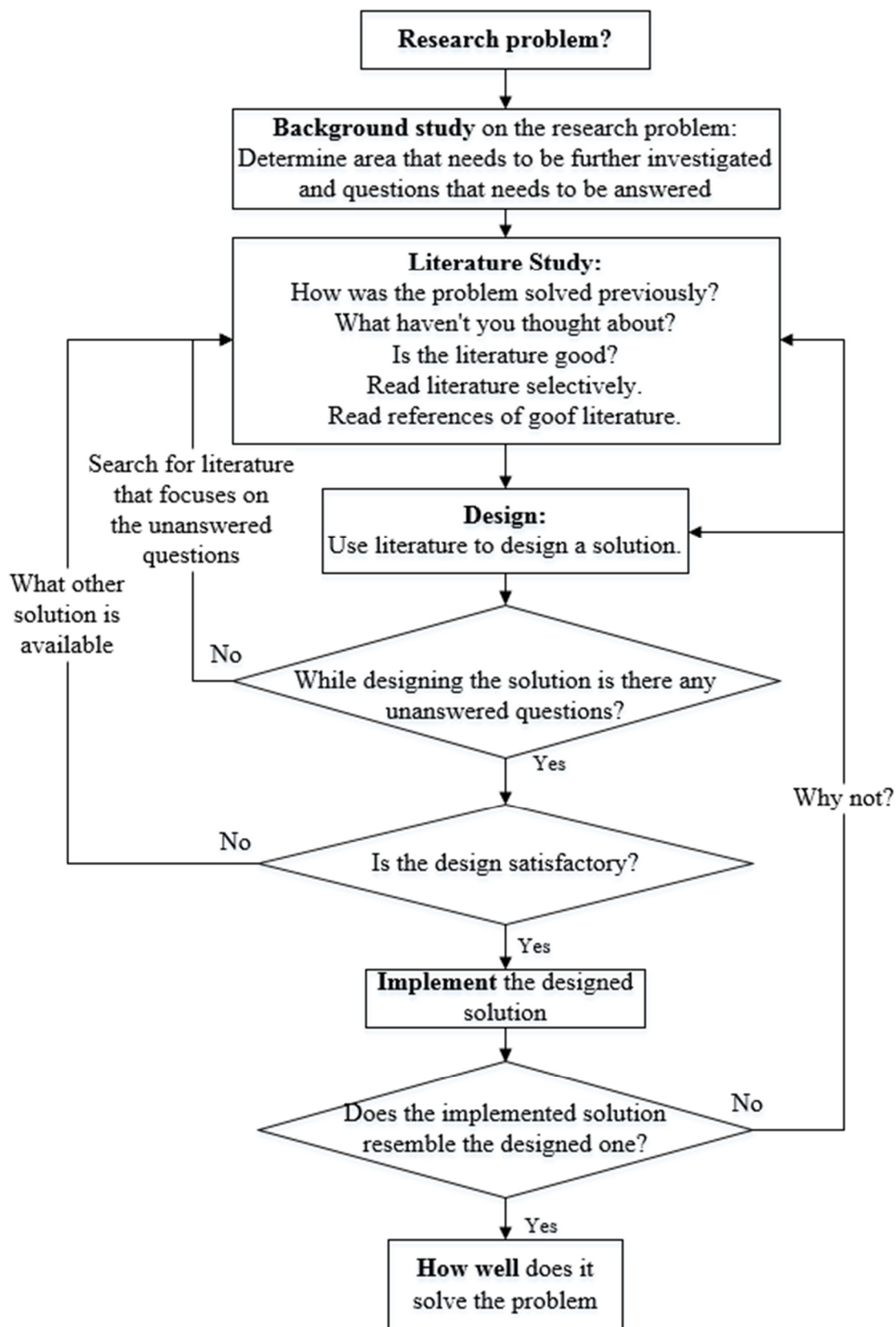


Figure 1-4: Research methodology

After the problem has been identified, a background study was done to determine the direction that need to be followed and which issues need to be addressed. This narrows down the aspects that need

to be researched. This is important as there can be many solutions to a problem and many ways to implement it. It also reduces the amount of literature that need to be studied in detail. This is important to do, as reading irrelevant literature can waste a lot of time, that could have been spent designing or implementing a solution. For this project, it was decided to solve the problem by replacing the bearings of the TWINS with AMBs. The issues that need to be addressed was discussed in section 1.3.

After determining the areas that need to be further investigated and the questions that need to be answered, a literature study was done. When doing the literature study, the following questions were kept in mind: How have other people solved the problem and how well did it work? What haven't you thought about? Determine whether the literature is good and applicable to the specific questions that need to be answered. This was done by reading the abstract, conclusion and index first. If the literature seemed good and applicable, it was studied further. It is also important to read the applicable references of the literature that proofed to be helpful.

After a satisfactory amount of literature have been read to answerer, most of the designer's questions, proceed to design the best solution that came out of the literature. If any new questions arise during the design process, go back to the literature to answer these questions.

After the design is completed (this includes simulating the solution) determine if the designed solution fulfils the required specification of the problem. If not go back to the literature to determine what mistakes were made, and what can be done differently.

After a solution was designed, that meets the desired specifications, implement that design and determine if it resembles the designed solution. If not, go back to the literature and the design process to determine what went wrong and what can be done to rectify the design flaws. If the implemented solution resembles the designed solution, determine how well it works.

1.6 DISSERTATION OVERVIEW

Chapter 2 will contain a detailed literature study about the different aspects of the project. This will include an introduction on the dynamic behaviour of a rotor, the different topologies of electromagnets, different methods that the AMB's journal can be fitted to the shaft of the rotor, backup bearings as well as different proximity sensor that is available. This study will enable the designer to make informed decisions when designing the AMB system.

Chapter 3 contains the detailed design of the AMB's electromagnet. It lists the specifications that the electromagnet was designed to. The electromagnet design is used to determine whether the available amplifiers will be sufficient. A comparison was made between a 4-pole and 8-pole electromagnet configuration generally used within the McTronX research group. The losses expected to be generated within the journal of the electromagnet was calculated to determine whether, it would be feasible to replace the bearings of the TWINS with AMBs. A shrink-fit analysis was done to determine whether it will be possible to fit the designed journal of the AMB to the rotor. Lastly, it describes a rotor dynamic check that was done to determine if the dynamic behaviour of the rotor will influence the operating range of the motor.

Chapter 4 discusses the tests that were performed on the AMBs, namely force, stiffness, damping, inductance and resistance verifications. It provides a method of determining the magnetic centre point of an AMB relative to the sensor's position. This chapter also discusses the Simulink model that was used to levitate the rotor while the measurements were taken. And lastly, it describes the way in which the motor was modified to enable it to use AMBs.

Chapter 5 gives a summary of the important design decisions that were made. It discusses the results that were obtained from testing the physical system as well as the important conclusions that were made from the simulated results. It also provides future work that still needs to be conducted.

1.7 SUMMARY

This chapter describes the problem that this project wishes to solve. The problem at hand is that the rotor of the HS-PMSMs overheats due to excessive losses produced by its ball bearings. This can cause the PM on the rotor to demagnetize or cause it to suffer mechanical failure. This problem will be solved by replacing the ball bearings with AMBs. The main issues that need to be addressed are: the electromagnets need to be designed, a rotor dynamic check needs to be done to ensure that the rotor will not start to resonate during operation, a stress analysis needs to be done to ensure the journals of the AMBs can be fitted to the rotor's shaft, and the stator of the TWINS needs to be modified as to enable the motor to use AMBs. Background knowledge on HS-PMSMs and AMBs was given, and the research methodology that will be used was explained.

CHAPTER 2: LITERATURE STUDY

This chapter will introduce the types of electromagnet configurations that can be used and will present an alternative backup bearing that can be used. It will focus on the dynamic behaviour of the rotor, including how to calculate the 1st three critical frequencies of the rotor. It will also introduce methods that can be used to fit material to a shaft.

2.1 ELECTROMAGNET

As discussed in chapter 1, electromagnets situated around the rotor are used to levitate the rotor with a typical airgap of 0.5 mm around the rotor. The advantages of using AMBs includes [9], [8]:

- The vibrations of the rotor can be controlled to a certain extent.
- The dynamic behaviour of the rotor can be controlled.
- Instabilities in rotor vibrations can be damped.
- No lubrication is required, as there aren't any mechanical contact between the rotor and stator.
- Service intervals are increased.

The biggest disadvantages of AMBs is the high cost that it is subjected to.

There exist different types of magnetic bearings that can support the rotor in its radial and axial direction or both. Radial magnetic bearings, as the name implies are used to suspend the rotor in its radial direction, while a thrust bearing is used, to support it in its axial direction. There also exists conical magnetic bearings that are used to support the rotor in its axial and radial direction simultaneously [8], [10].

There exists two basic types of radial magnetic bearing configurations. These include a heteropolar and a homopolar configuration. They can be distinguished from each other in the direction that the magnetic flux flows inside the rotor. The two configurations are shown in Figure 2-1.

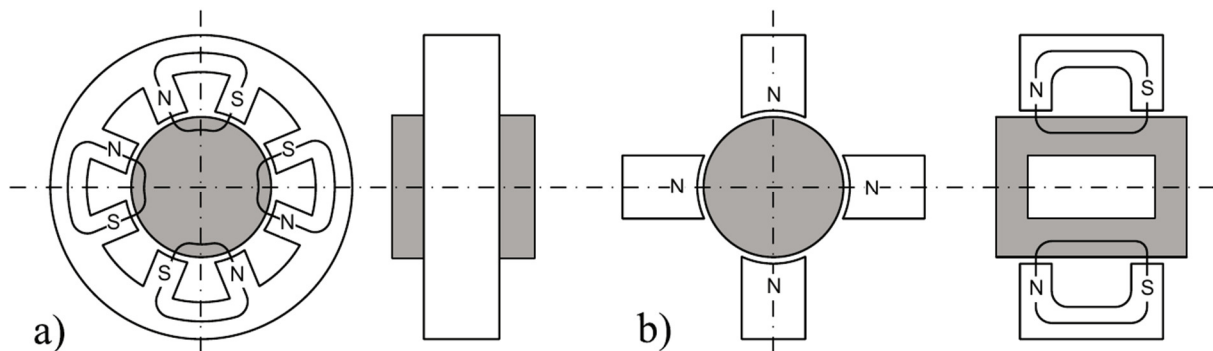
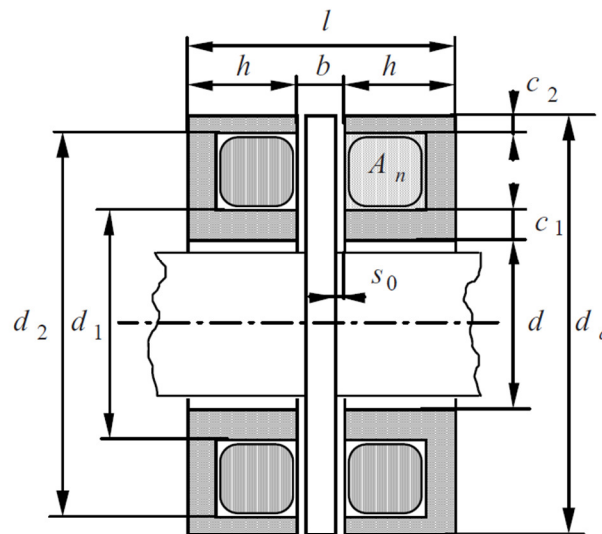


Figure 2-1: Radial magnetic bearing a) Heteropolar b) Homopolar [8]

The magnetic flux in a heteropolar bearing flows perpendicular to the axis of the rotor. This means that the magnetic material on the rotor will continuously experience a magnetic flux with alternating polarity. Whereas the magnetic flux in a homopolar bearing flows mainly in the direction parallel to the rotor's axis. With a homopolar configuration, the rotor's magnetic material does not experience a change in the polarity of the magnetic flux within it. Due to this, the losses caused by eddy currents in the rotor's material is substantially reduced [8]. As explained in chapter 1, the axial space on the

rotor is very limited, due to this it would not be possible to use a homopolar configuration, as they use much more axial space than the heteropolar configuration.

A rotor is suspended in the axial direction using a thrust bearing. The design of the thrust bearing depends on the orientation of the rotor. If the motor is operated in a horizontal orientation, the thrust bearing only needs to counteract the forces produced by the rotor's axial vibration and any axial forces produced by the load. Whereas if the motor is operated in the vertical direction, the thrust bearing will also have to support the weight of the rotor [8]. Figure 2-2 shows the construction of a thrust magnetic bearing.



d Inner diameter (or bearing diameter)	d_a Outer diameter
c_1 Inner leg width	d_1 Inner winding space diameter
d_2 Outer winding space diameter	A_n Slot cross section (winding space)
h Pot magnet height	c_2 Outer leg width
s_0 Nominal air gap	h_n Slot depth
l Bearing length	

Figure 2-2: Thrust magnetic bearing [8]

2.2 BACKUP BEARINGS

Backup bearings need to be installed when AMBs are used. It keeps the rotor from touching the motor or AMB's stator in the event of an AMB failure or overload. The backup bearings should be able to support the rotor for the duration it takes the rotor to come to a stop. Normally ball bearings are used with a diameter between the inner diameter of the electromagnet and the outer diameter of the rotor [8]. This is shown in Figure 2-3. It can be seen that if the AMB fails, the rotor will fall onto the ball bearings. This is to ensure that no damage is done to the inner part of the motor or AMBs' stator.

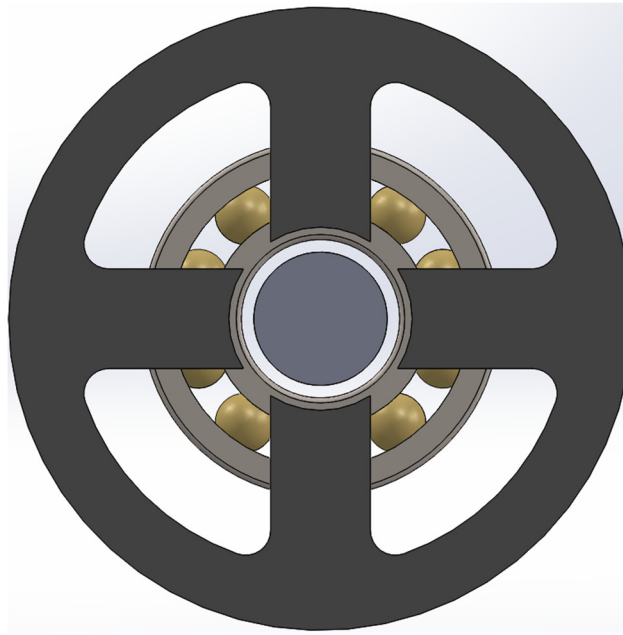


Figure 2-3: Ball bearing used for backup bearings

It was however seen that there is not enough axial space on the rotor for the use of ball bearings as backup bearings. Further research was done to determine another method that can be used as backup bearings, which will save axial space on the rotor. Another solution that was found is to use backup pins. The use of backup pins gives one major advantage over the use of ball bearings. The pins eliminates the chance that the rotor can start backward rolling into the backup bearings [11].

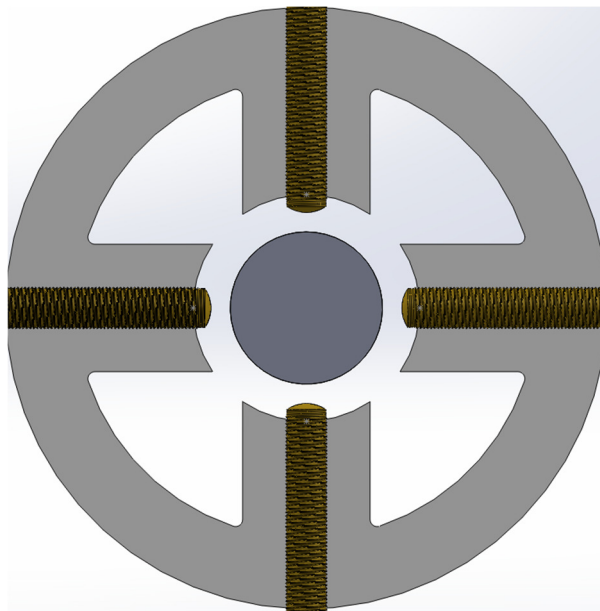


Figure 2-4: Pins used for backup bearings

The backup pins were favourable as they could be placed in line with the sensors as well as with the electromagnets' end coils, in the axial direction.

2.3 ROTOR DYNAMICS

The dynamic behaviour of the rotor is an important component to consider when designing active magnetic bearings, as it determines the force as well as the force slew rate that needs to be produced by the AMBs.

As the rotor rotates, some of its energy is transferred into other forms of energy, than energy to accomplish useful work. Some of this leakage energy is transformed into thermal energy as well as other forms of mechanical energy such as “mechanical side effects”. Several factors are responsible for the energy transfer, from rotational energy to other forms of motion, with the main one being rotor unbalance. These other forms of motion may be various modes of vibrations with varying intensities as shown in Figure 2-5. All three main modes of vibration may be present during rotor operation, but the bending modes are of highest concern, as it is not highly dependent on the bearing stiffness. This means that its position cannot be controlled by the AMBs. These vibrations are also transmitted to the nonrotating parts of the machine and eventually to the machine foundation, building walls, and adjacent equipment [4]. It should, however, be noted that only the rotor’s vibrations will be considered in this document and not the vibrations of the machine’s housing.

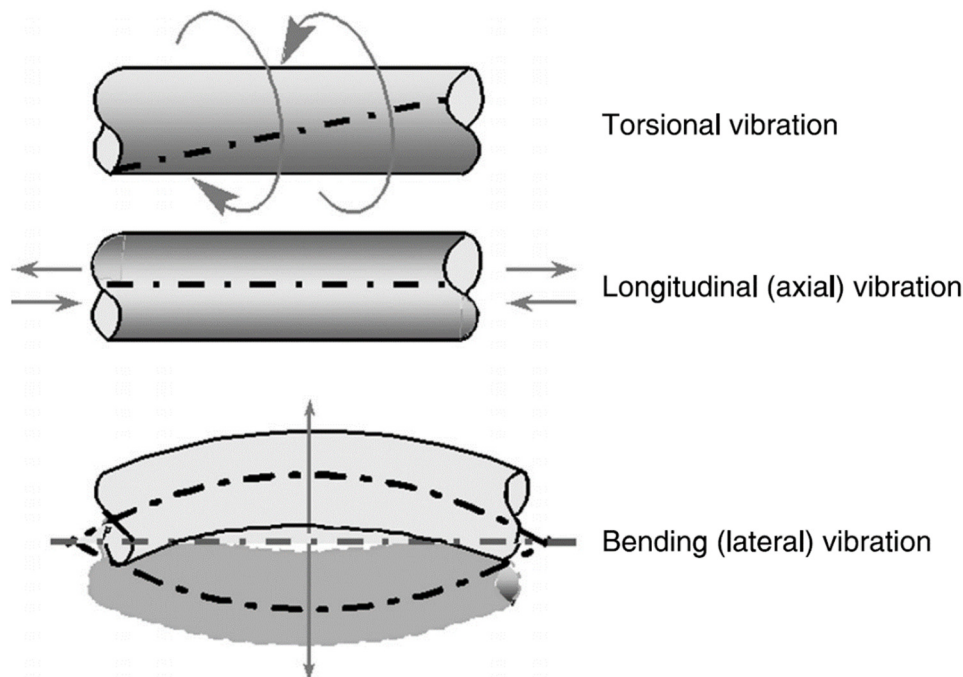


Figure 2-5: Vibration modes of a rotor [4]

The next section will give an introduction to vibrations.

2.3.1 Vibrations

Any repeating motion like a swinging pendulum is called an oscillation or vibration. Generally, a vibrating system includes a method for storing potential energy like a spring or the elasticity of a body, a method for storing kinetic energy like mass or inertia, and a method to dissipate the energy like a viscous damper. Vibrations are caused by the constant interchanging of a system's potential energy to kinetic energy and vice versa [12].

For example, consider the pendulum shown in Figure 2-6. Let the mass be released at position 1 with an angular displacement of θ . At position 1 the mass has maximum potential energy due to its position

above the equilibrium point and zero kinetic energy as its speed is zero. Due to gravitational force, the mass will start to swing to the left, accelerating until it reaches position 2. At position 2 the mass has no potential energy and maximum kinetic energy as this is the point where the mass will have its maximum speed. Due to its kinetic energy, the mass will not stop at position 2, it will keep on moving to position 3 with a deceleration in its speed and increase in its potential energy. At position 3 the mass will again have maximum potential energy and zero kinetic energy, with the same height as at position 1, if all friction is neglected (no energy loss). This motion will continue at the system's natural frequency and is called the free vibration of the system.

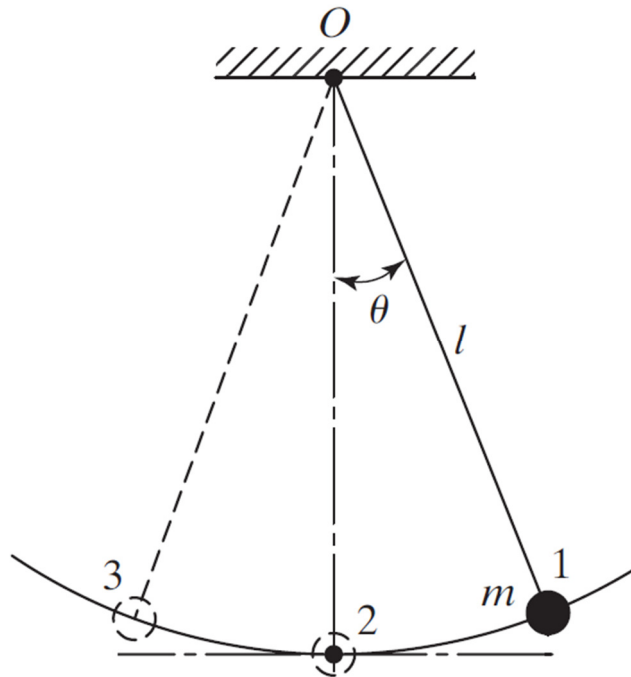


Figure 2-6: Oscillating pendulum system [13]

A free vibration exists if a piece of mass is excited by an initial disturbance and left to vibrate on its own. A forced vibration occurs while the mass is subjected to an external disturbance force while oscillating. If this disturbance force is of the same frequency as the natural frequency, the system will start to resonate, meaning that its amplitude of oscillation will increase [12]. In the swinging pendulum system, if the mass is pushed every time it starts swinging back to position 2, energy will constantly be added to the system. This will cause the mass to start resonating. However, if the mass is energized at a different frequency than its natural frequency, energy will be added to the mass in some instances and removed at others. The closer the excitation frequency is to the system's natural frequency, the more instances energy would be added to the mass than removed from it. This example gives an intuitive feeling for vibrations as everybody is familiar with a swing. The next example will be more relevant to a rotor-bearing system, as the stiffness of a spring can relate to the stiffness of the bearing, and the block of mass can relate to its weight.

Consider the spring-mass system shown in Figure 2-7 with a spring constant k , mass m , elongation x of the spring and zero damping.

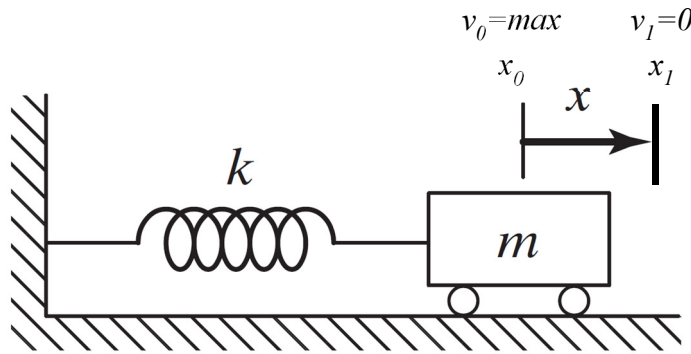


Figure 2-7: Spring-mass system [12]

If the mass m is displaced by an amount x from its equilibrium position and released, it will oscillate around its equilibrium position at the system's natural frequency ω_n (free vibration). As the mass oscillates, the energy of the system is continuously being transferred between the mass (kinetic energy (T)) and the spring (elastic potential energy (U)), if there is no friction. If the displacement of the mass is at a maximum (x_f), its speed will be zero $\therefore T=0$ and the potential energy stored in the spring will be at its maximum $\therefore U=0.5kx^2$ with the force of the spring given by $F=kx$. If the mass is at its equilibrium position, the elastic potential energy stored in the spring is zero, and the speed of the mass is at its maximum $\therefore T=0.5mv^2$ with the maximum velocity being $v=\omega x_{max}$. Rayleigh's energy method can be used to determine the natural frequency of this system [14].

Rayleigh's energy method uses the principle of conservation of energy, by stating that $T_{max}=U_{max}$ as kinetic energy is zero if the elastic potential energy is at its maximum and vice versa [12].

By using Rayleigh's energy method, the natural frequency of oscillation is defined by [14] as:

$$\omega_n^2 = \frac{0.5kx^2}{0.5mx^2}$$

$$\omega_n = \sqrt{\frac{k}{m}} \quad (2.1)$$

Consider the spring-mass damper system shown in Figure 2-8.

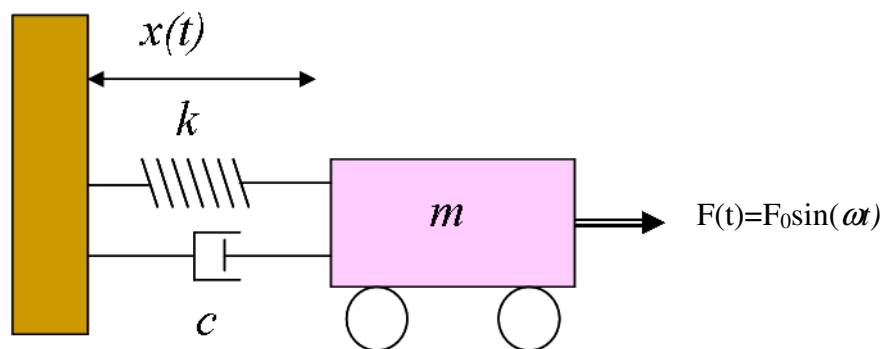


Figure 2-8: Spring-mass damper system

The damper is a viscous damper of which the force is proportional to the velocity of the mass and in the opposite directions $F = -cx$ with c the damping coefficient. It removes energy from the system [12]. The damping of AMBs are modelled in the same way. This is because the force produced by the AMBs changes relative to the velocity of the rotor.

The system will react differently depending on how much damping there is. A system can be underdamped, critically damped, and overdamped. Consider the spring-mass damper system. In an underdamped system, if the mass is displaced away from its equilibrium position and released, it will oscillate around its equilibrium position with decreasing amplitude. If the system is critically damped, the mass will return to its equilibrium position with the fastest time possible without overshooting. While if there is too much damping (overdamped), the mass will return to its equilibrium position in a longer time without oscillating. It is counter-intuitive that there can be too much damping. The easiest way to visualize it is with a swinging pendulum. If the pendulum swings in the air, it will oscillate around its equilibrium position, eventually coming to a standstill (underdamped). If it was swinging in a viscous fluid like syrup, it will not oscillate but reach its equilibrium point much faster than it will if it were in wet concrete.

To distinguish the three degrees of damping the damping ratio is defined as the ratio of the damping of the system to the critical damping of the system:

$$\zeta = \frac{c}{c_c} \quad (2.2)$$

Where the critical damping is defined as:

$$c_c = 2m_{eq}\omega_n \quad (2.3)$$

Note that the frequency of oscillation of the damped system is not the same as that of an undamped system and is defined as:

$$\omega_d = \omega_n \sqrt{1 - \zeta^2} \quad (2.4)$$

Consider a sinusoidal disturbance force on the spring-mass damper system, as shown in Figure 2-8. It is clear that the oscillating frequency of the mass will be the same as that of the excitations force. However, as the frequency of the excitation force moves closer to that of the system's critical frequency, the greater the amplitude of the oscillation would become, as shown in Figure 2-9 [12].

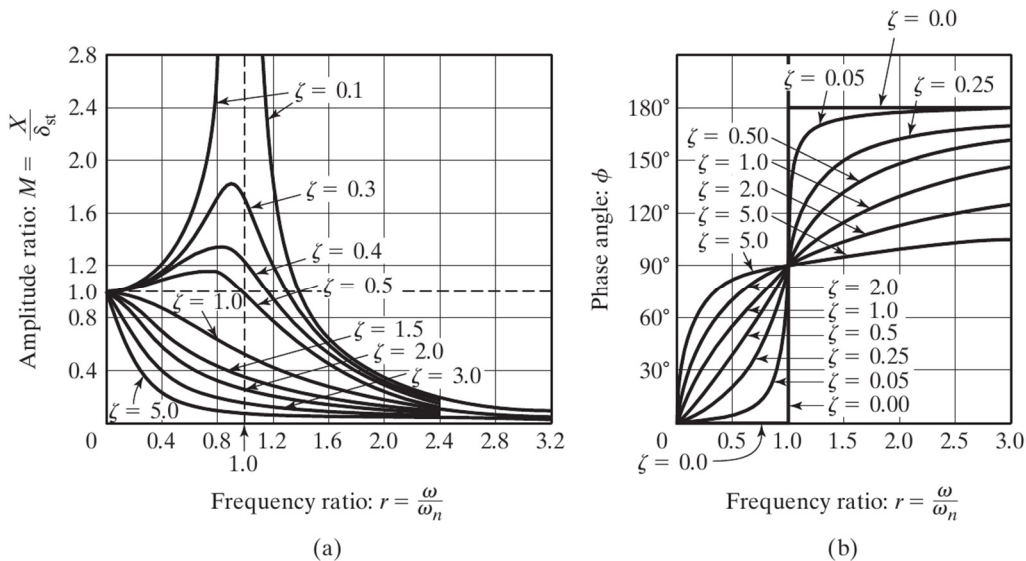


Figure 2-9: Influence of damping [12]

It can be seen that the amplitude of oscillation increases as the excitation frequency approaches that of the damped natural frequency. In Figure 2-9, M represents the ratio of the dynamic amplitude X to the static response given by $\delta_{st}=F_0/k$ (How much more the displacement is due to the oscillation of the excitation force). The phase angle Φ represents the angle between the disturbance force and the dynamic response. These examples offer an intuitive feel on how mechanical systems react to a disturbance force [12].

The different ways that a rotor starts to oscillate will be presented in the next section.

2.3.2 Mode shapes

Consider a rotor suspended by active magnetic bearings. As the rotor's speed increases, it will start to oscillate or resonate at different frequencies. These frequencies are called the critical frequencies of the rotor-bearing system. The shape that the rotor's centreline form while it oscillates is called its mode shape. As discussed earlier, the main thing that excites the different modes is the unbalance in the rotor [4]. Unbalance in a rotor is caused by the inhomogeneity of the material, meaning that its density is not the same everywhere. It is also caused by imperfections in the geometry of the rotor [15]. An illustration of unbalance is shown in Figure 2-10.

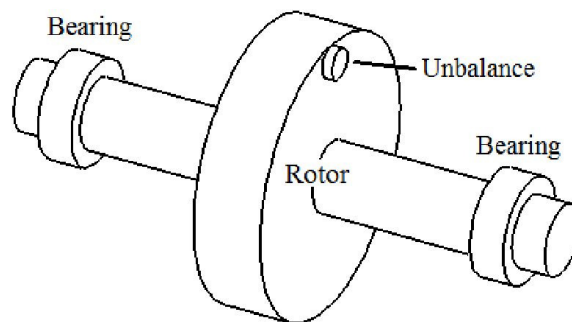


Figure 2-10: Rotor with unbalance [16]

Unbalance is essentially a small piece of mass that is added to one side of a homogeneous rotor. While the rotor rotates centripetal forces act on this piece of mass. As the rotor rotates the force generated by the unbalance will cause the rotor to start oscillating in different mode shapes. This unbalance force can be counteracted by adding a piece of mass on the opposing side of the rotor that counteracts the first unbalance force, or a piece of mass can be removed on the side of the unbalance. The different mode shapes that can be excited in a rotor with different bearing stiffness' are shown in Figure 2-11. The mode shapes are a function of the rotor's and bearings' stiffness as well as the distribution of mass along the rotor [17].

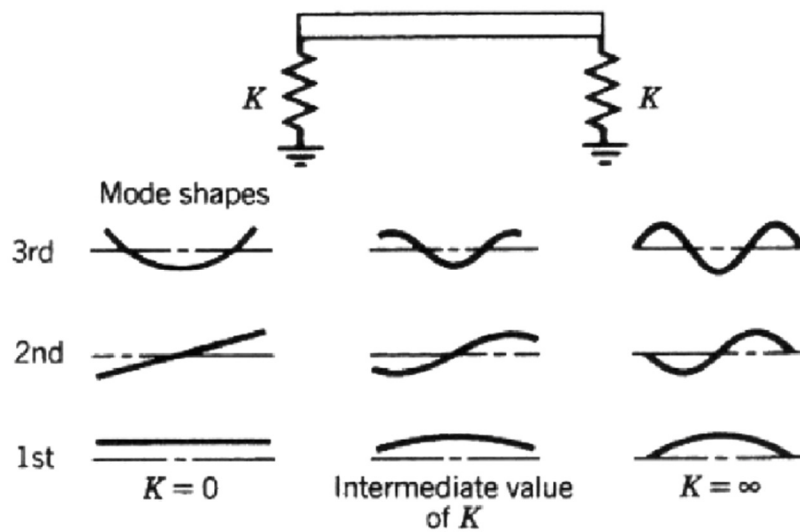


Figure 2-11: Mode shapes [18]

Only the first three critical frequencies are shown, but in reality, there exists an infinite number of mode shapes. However, only the first three or four fall within the operating speed of high-speed machines [18]. If the bearing stiffness is very low ($k \approx 0$) the rotor will experience very low bending at the first two critical frequencies. These two modes are called the rigid modes of the rotor as it stays essentially straight at these frequencies. The rigid modes of the rotor are shown in Figure 2-12. Consider a rotor that is completely balanced in its axial direction, meaning that both disks are the same weight as well as the same distance from the rotor's centre. With both bearings having the same stiffness. This will cause the rotor's centreline to form a cylinder-shape while the rotor resonates at its first rigid critical frequency. While the rotor's centre line will form a double cone with an apex at the middle of the rotor, at the second rigid critical frequency of the rotor-bearing system. Any changes in the rotor's symmetry or an unbalance in the bearing's stiffness will modify the mode shapes.

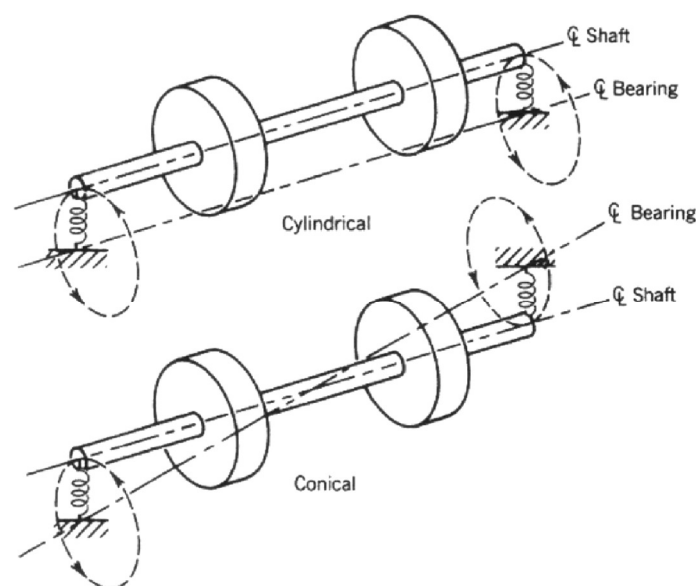


Figure 2-12: Mode shapes of a symmetrical rigid rotor.

An analytical method to calculate the first three critical frequencies with a low bearing stiffness will be provided in the next few subsections.

2.3.2.1 1st and 2nd rigid modes

The analytical calculation to determine the critical frequencies of the rotor's first two rigid mode shapes will be discussed in this sub subsection. This method can be found in [16], [12].

The critical frequencies of the first two rigid mode shapes of a rotor-bearing system can be determined by approximating the rotor as a 2 degrees of freedom system [16]. Meaning that the rotor's response can be fully described by two independent variables. This can be done by assuming that, the rotor will remain rigid for its first two critical frequencies, or by assuming that its Young's modulus is infinite (meaning that it cannot bend). In this case the rotor's motion will be described by its translation (up and down movement) in the x -direction and its rotation around its centre of mass (CM) as shown in Figure 2-13. The system consists of a rotor with mass m_{eq} and moment of inertia J_{eq} around an axis perpendicular to the page through the centre of mass, two bearings with stiffness k_1 and k_2 situated at a distance l_1 and l_2 respectively away from the centre of mass [12]. Note that the rotational speed of the rotor around its rotational axis is zero.

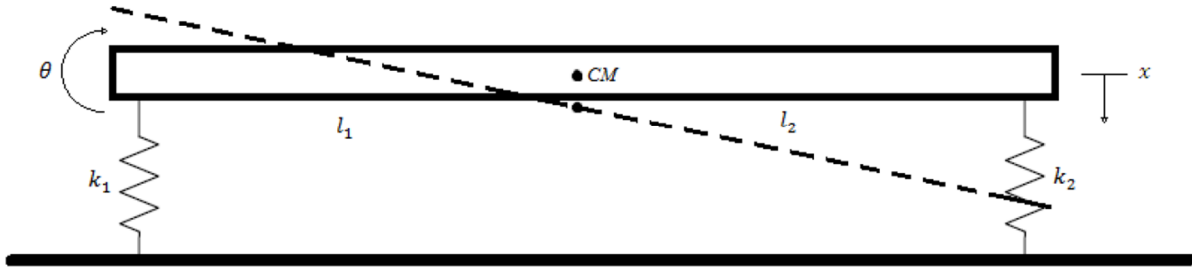


Figure 2-13: Two degree of freedom approximation [16]

The translation and rotational motion of a rotor that was subjected to an initial translation disturbance can be described by [16]:

$$m_{eq}\ddot{x} + k_1(x - \theta l_1) + k_2(x + \theta l_2) = 0 \quad (2.5)$$

$$J_{eq}\ddot{\theta} - k_1(x - \theta l_1)l_1 + k_2(x + \theta l_2)l_2 = 0 \quad (2.6)$$

With x representing the translation of the rotor and θ representing the rotational motion of the rotor around CM. Where the double derivative describes the translational and rotational acceleration. If the motion of the rotor's translation is assumed to be harmonic, then its translation response can be defined as [16].

$$x(t) = X\cos(\omega t) \quad (2.7)$$

With the amplitude of oscillation being X . This motion of the rotor can be produced by pulling the whole rotor down and releasing it.

While if it is assumed that the rotational motion around CM is harmonic, its rotational response can be written as [16]:

$$\theta(t) = \Theta\cos(\omega t) \quad (2.8)$$

With the amplitude of the oscillation being Θ . This motion can be reproduced by pulling one side of the rotor down while pushing the other side of the rotor up and releasing it.

By differentiating (2.7) and (2.8) the rotor's translational and rotational acceleration can be defined as [16]:

$$\ddot{x}(t) = \frac{d^2}{dt^2} x(t) = -\omega^2 (X \cos(\omega t)) \quad (2.9)$$

$$\ddot{\theta} = \frac{d^2}{dt^2} \theta(t) = -\omega^2 (\Theta \cos(\omega t)) \quad (2.10)$$

By using the definition in (2.9) and (2.10), (2.5) and (2.6) can be rearranged as (2.11) [16]

$$\begin{bmatrix} (-m_{eq}\omega^2 + k_1 + k_2) & -k_1 l_1 + k_2 l_2 \\ (-k_1 l_1 + k_2 l_2) & (-J_{eq}\omega^2 + k_1 l_1^2 + k_2 l_2^2) \end{bmatrix} \begin{bmatrix} X \\ \Theta \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (2.11)$$

The natural frequencies of the rigid body can be determined by setting the determinant of (2.11) equal to zero [16].

$$\det \begin{bmatrix} -m_{eq}\omega^2 + k_1 + k_2 & -k_1 l_1 + k_2 l_2 \\ -k_1 l_1 + k_2 l_2 & -J_{eq}\omega^2 + k_1 l_1^2 + k_2 l_2^2 \end{bmatrix} = 0 \quad (2.12)$$

This results in a 4th-degree polynomial, as shown in (2.13).

$$J_{eq} m_{eq} (\omega^2)^2 - [m_{eq} (k_1 l_1^2 + k_2 l_2^2) + J_{eq} (k_1 + k_2)] \omega^2 + [(k_1 + k_2) (k_1 l_1^2 + k_2 l_2^2) + (k_1 l_1^2 + k_2 l_2^2)^2] = 0 \quad (2.13)$$

The roots of this equation can be determined by using the quadratic equation:

$$\omega^2 = \frac{m_{eq} (k_1 l_1^2 + k_2 l_2^2) + J_{eq} (k_1 + k_2)}{2 J_{eq} m_{eq}} \pm \frac{\sqrt{[m_{eq} (k_1 l_1^2 + k_2 l_2^2) - J_{eq} (k_1 + k_2)]^2 - 4 J_{eq} m_{eq} (k_1 l_1^2 + k_2 l_2^2)}}{2 J_{eq} m_{eq}} \quad (2.14)$$

If $k_1 l_1 \neq k_2 l_2$ the mode shapes will be cylindrical as well as conical. With the first mode shape being mostly cylindrical and partially conical and the second mode shape being mostly conical and partially cylindrical. Meaning that the double cone centre point will not coincide with the CM of the rotor. However, if $k_1 l_1 = k_2 l_2$, the two mode-shapes of the rotor can be uncoupled from each other [12]. The critical frequencies of the two rigid modes can then be reduced with the cylindrical mode at:

$$\omega_{cylindrical} = \sqrt{\frac{k_1 + k_2}{m_{eq}}} \quad (2.15)$$

And the conical mode at:

$$\omega_{conical} = \sqrt{\frac{k_1 l_1^2 + k_2 l_2^2}{J_{eq}}} \quad (2.16)$$

It can be seen that the conical mode can occur before the cylindrical mode if the length of the bearing positions away from the rotor's centre of mass is reduced.

2.3.2.2 1st Bending frequency

Rayleigh's energy method can be used to estimate the first bending frequency of the rotor. By approximating the rotor-bearing system as a pin-supported beam, as shown in Figure 2-14. Rayleigh's energy method is based on the constant transfer of energy between kinetic and potential energy [16].

The natural frequency ω_n of a beam can be determined by using its deflection due to its own weight [19].

The calculations will be illustrated by using a simple beam as shown in Figure 2-14. The density of the beam's material is 8000 kg/m^3 with a Young's modulus of 200 GPa . Equations of typical beam deflections are shown in Appendix A as found in [20].

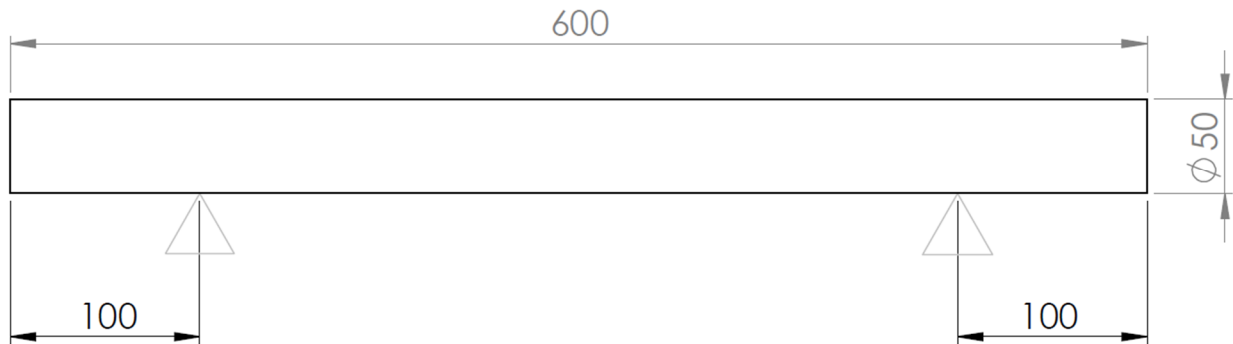


Figure 2-14: Pin supported beam

Figure 2-15 shows the free body diagram of the rotor at its first bending frequency.

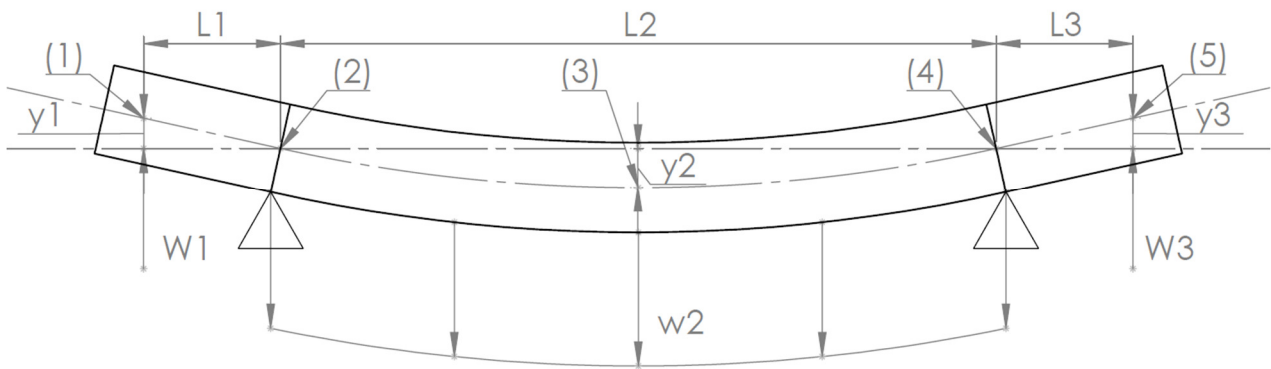


Figure 2-15: Bending of pin supported beam [16]

Stations (2) and (4) represents the locations of the supports, where stations (1), (2), (3) is the locations that the bending of the shaft will be determined. The weight of the shaft to the left of station (2) will be lumped into force W_1 a distance L_1 away. The weight of the shaft to the right of station (4) will be lumped into force W_3 a distance L_3 away. While the weight of the shaft between station (2) and (4) will be represented as a uniformly distributed force w_2 [16].

Stations (2) and (4) will be the positions of the bearings if the stiffness of the bearings is infinite. However, if the stiffness is relatively small, these node points will move inward. The 1st bending frequency will only be verified for a bearing stiffness of infinity, as it is not known how much the

node points will shift. This calculated value of the 1st bending frequency will also be lower than if the bearing stiffness is relatively small.

The length of L_1 and L_3 will be determined by calculating the length that the lumped weight W_1 and W_3 should be away from locations (2) and (4) to produce the same deflection of the beam, that would be produced by the uniformly distributed weight of the beam. To determine this, case 8 and 9 in Appendix A was used to determine (2.17):

$$0 = L_1^3 - 3L_T \cdot L_1^2 + 0.75L_T^3 \quad (2.17)$$

With L_T the total length of the beam to the left of station (2) and L_1 the length away from the node position that the weight will be concentrated at, in the example $L_T=100$ mm and L_1 can be calculated as 29.12 mm. The values of the numerical model are summarized in Table 2-1.

Table 2-1: Properties of the numerical model

Section (1)-(2)	Section (2)-(4)	Section (4)-(5)
$L_1=29.12$ mm	$L_2=400$ mm	$L_3=29.12$ mm
$\phi_1=50$ mm	$\phi_2=50$ mm	$\phi_3=50$ mm
$W_1=9.8 \cdot 0.5\pi$ N	$w_2=9.8 \cdot 2\pi/L_2$ N/m	$W_1=9.8 \cdot 0.5\pi$ N

The deflections of the beam at positions (1), (3), (5) will be calculated in the next section.

2.3.3 Deflection calculations

The deflection of the beam can be calculated by using conventional beam theory and using the superposition principle to determine the total deflection [16]. An upward deflection of the beam will be indicated by a positive sign, while a downward deflection will be indicated by a negative sign.

Consider the free body diagram of the beam, as shown in Figure 2-15. The weight of the overhung piece needs to be reversed, as shown in Figure 2-15. The total deflection of the beam is determined by determining the deflection at stations (1), (3), and (5) due to weight W_1 , w_2 and W_3 separately, and using superposition to determine the total deflection.

The deflection of the beam due to force W_1 is as follow:

The deflection y_1 is calculated with the sum of two partial deflections. The first is the partial deflection due to the cantilever beam approach, and can be calculated as [16]:

$$\begin{aligned}
 y_1' &= \frac{W_1 \cdot L_1^3}{3 \cdot (EI)_{(1)-(2)}} \\
 y_1' &= \frac{9.8 \cdot 0.5\pi \cdot (0.02912)^3}{3 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)} \\
 y_1' &= 2.065 \text{ nm}
 \end{aligned} \quad (2.18)$$

With E the Young's modulus (stiffness due to material) and I the moment of inertia of the shaft (stiffness due to shape of beam) between sections (1) and (2).

The partial deflection of y_1 calculated from a pin-supported beam approach can be calculated as [16]:

$$y_1'' = \frac{(M_1) \cdot (L_2) \cdot (L_1)}{3 \cdot (EI)_{(2)-(4)}} \quad (2.19)$$

$$y_1'' = \frac{(448.27e^{-3}) \cdot (0.4) \cdot (0.02912)}{3 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)}$$

$$y_1'' = 28.368 \text{ nm}$$

Where E and I is the Young's modulus and moment of inertia of the shaft between stations (2) and (4), and M_I the moment that is produced by W_I around station (2).

The total deflection at station (1) due to force W_I can be calculated as [16].

$$y_1 = y_1' + y_1''$$

$$y_1 = (2.065e^{-9}) + (28.368e^{-9}) \quad (2.20)$$

$$y_1 = 30.43 \text{ nm}$$

Bending moment M_I also causes a deflection station (3) and can be calculated by using case 6 in Appendix A as in [20]:

$$y_2 = \frac{M_1 \cdot L_2^2}{16(EI)_{(2)-(4)}}$$

$$y_2 = \frac{(448.27e^{-3}) \cdot (0.4)^2}{16 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)} \quad (2.21)$$

$$y_2 = -73.056 \text{ nm}$$

The deflection of station (5) due to bending moment M_I can be calculated by using case 6 in Appendix A:

$$y_3 = \theta_4 L_3$$

$$y_3 = \frac{M_1 \cdot L_2}{6 \cdot (EI)_{(2)-(4)}} \cdot L_3$$

$$y_3 = \frac{(448.27e^{-3}) \cdot (0.4)}{6 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)} \cdot (0.02912) \quad (2.22)$$

$$y_3 = 14.183 \text{ nm}$$

where θ_4 is the angle of deflection at station (4)

The deflection of the beam due to force w_2 is as follow:

The deflection at station (1) can be calculated by using case 4 in Appendix A:

$$y_1 = \theta_2 L_1$$

$$y_1 = \frac{w_2 \cdot L_2^3}{24 \cdot (EI)_{(2)-(4)}} \cdot L_1$$

$$y_1 = \frac{49\pi \cdot (0.4)^3}{24 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)} \cdot 0.02912 \quad (2.23)$$

$$y_1 = 194.82 \text{ nm}$$

The deflection at station (3) can be calculated by using case 4 in Appendix A:

$$y_2 = \frac{5 \cdot w_2 \cdot L_2^4}{384 \cdot (EI)_{(2)-(4)}} \quad (2.24)$$

$$y_2 = \frac{5 \cdot 49\pi \cdot (0.4)^4}{384 \cdot \left(\frac{\pi}{64} (0.05)^4\right)}$$

$$y_2 = -836.27 \text{ nm}$$

And the deflection at station (5) can be calculated by using case 4 in Appendix A:

$$y_3 = \theta_4 L_3$$

$$y_3 = \frac{w_2 \cdot L_2^3}{24 \cdot (EI)_{(2)-(4)}} \cdot L_3$$

$$y_3 = \frac{49\pi \cdot (0.4)^3}{24 \cdot (2e11) \cdot \left(\frac{\pi}{64} (0.05)^4\right)} \cdot 0.02912 \quad (2.25)$$

$$y_1 = 194.82 \text{ nm}$$

The deflection of station (1), (3), and (5) due to W_3 can be calculated by using the same equations than those used to calculate the deflections due to W_1 and switching it around.

A summary of the rotor's deflection at stations (1), (3), and (5) due to weights, W_1 , w_2 , and W_3 are listed in Table 2-2.

Table 2-2: Summary of deflections in the rotor as in Figure 2-15

Deflection at	W_1	w_2	W_3
y_1	30.43 nm	194.82 nm	14.183 nm
y_2	-73.056 nm	-836.27 nm	-73.056 nm
y_3	14.183 nm	194.82 nm	30.43 nm

The total deflection at station (1), (3) and (5) can be determined by using the principle of superposition. By adding the deflections in Table 2-2 the total deflection can be determined as:

$$y_1 = (30.43 \text{ nm}) + (194.82 \text{ nm}) + (14.183 \text{ nm})$$

$$y_1 = 239.43 \text{ nm}$$

$$y_2 = (-73.056 \text{ nm}) + (-836.27 \text{ nm}) + (-73.056 \text{ nm})$$

$$y_2 = -982.38 \text{ nm} \quad (2.26)$$

$$y_3 = (14.183 \text{ nm}) + (194.82 \text{ nm}) + (30.43 \text{ nm})$$

$$y_3 = 239.43 \text{ nm}$$

Before the natural frequency of the rotor can be determined, the evenly distributed load w_2 between stations (2) and (4) (measured in N/m) needs to be converted to a point load P_2 situated at station (3). This point load should produce the same deflection y_2 at station (3) as the distributed load w_2 . This is done by comparing case 1 and case 4 in Appendix A. Point load P_2 can be calculated as follow:

$$P_2 = \frac{5}{8} \cdot w_2 \cdot L_2$$

$$P_2 = 38.4845 \text{ Kg} \quad (2.27)$$

The natural frequency of the rotor rigidly suspended at stations (2) and (4) can now be determined by using (2.28) [19].

$$\omega_n = \sqrt{g \frac{\sum Wy}{\sum Wy^2}}$$

$$\omega_n = \sqrt{9.8 \cdot \frac{W_1 \cdot y_1 + P_2 \cdot y_2 + W_3 \cdot y_3}{W_1 \cdot y_1^2 + P_2 \cdot y_2^2 + W_3 \cdot y_3^2}} \quad (2.28)$$

$$\omega_n = 32214 \text{ rad/s}$$

This produces a natural frequency of 536.89 Hz. This compares to a simulated value of 543 Hz when using RotFE a MATLAB® toolbox written by Izhak Bucher. It shows that this is an accurate analytical method to calculate the first bending critical frequency of a rotor.

2.3.4 Unbalance specification standard

As stated earlier in section 2.3, unbalance is the largest cause of mechanical vibrations in an electrical motor. The amount of unbalance permissible on a rotor is specified by the ISO 1940/1 balancing standard for rigid rotors. The standard uses the letter “G” to specify the extent of rotor balancing. The “G” number is specified to be the product of the rotor’s maximum operating speed (ω) in radians per second and the eccentricity (e) of the rotor as illustrated in (2.29). The rotor’s eccentricity is the distance between the rotor’s centre of mass (C_M) to its geometric centre (C_G), as shown in Figure 2-16 [21].

$$G = e \cdot \omega \quad (2.29)$$

For a specific “G” balancing the rotor’s eccentricity will be inversely proportional to its maximum operating speed.

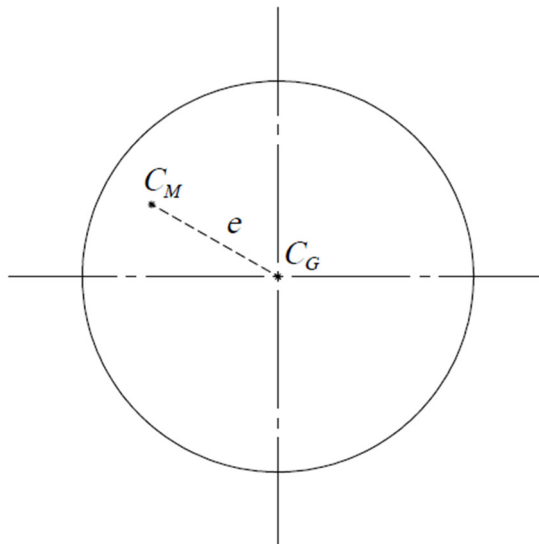


Figure 2-16: Illustration of eccentricity

The rotor’s permissible unbalance (U_{per}) is specified as the product of the rotors eccentricity and its mass:

$$U_{per}(kg \cdot mm) = e_{per} \cdot m \quad (2.30)$$

Where e_{per} is the permissible eccentricity of the rotor’s centre of mass in *millimetres* and m the rotor’s mass in *kilograms*.

By using (2.29) with the eccentricity the subject of the equation and substitution it into (2.30), the permissible unbalance can be written as a function of the balancing standard number “ G ” as, shown in (2.31). Note the eccentricity’s unit of measure was converted to *meters*.

$$U_{per}(kg \cdot m) = 9.549e^{-3} \cdot G \cdot \frac{m}{N} \quad (2.31)$$

with N the maximum speed of the rotor in r/min.

The permissible unbalance can be used to calculate the unbalance force that will be produced by the rotor’s unbalance.

The unbalance force produced by the eccentricity of the rotor can be defined by [16] as:

$$F_{ub} = m \cdot e \cdot \omega^2 \quad (2.32)$$

Equation (2.31) can be substituted into (2.32) to define the unbalance force created by the rotor as a function of the balancing standard “ G ” number:

$$F_{ub} = 9.549e^{-3} \cdot G \cdot \frac{m}{N} \cdot \omega^2 \quad (2.33)$$

This equation is very relevant as it determines the unbalance force that the AMBs will need to counteract.

2.4 SHAFT LAMINATION CONNECTION

As stated in Chapter 1, the existing rotor of one of the PMSMs need to be modified to enable the rotor to be levitated with active magnetic bearings. This means that the rotor needs to be fitted with magnetic material as well as endstops to keep the thin laminations that are used as the magnetic material from buckling.

There are a few methods that can be used to connect the lamination stacks to the rotor’s shaft, in order to avoid loss of contact between the two at high rotational speeds. The different methods are shown in Figure 2-17 and will be briefly discussed in this section [22].

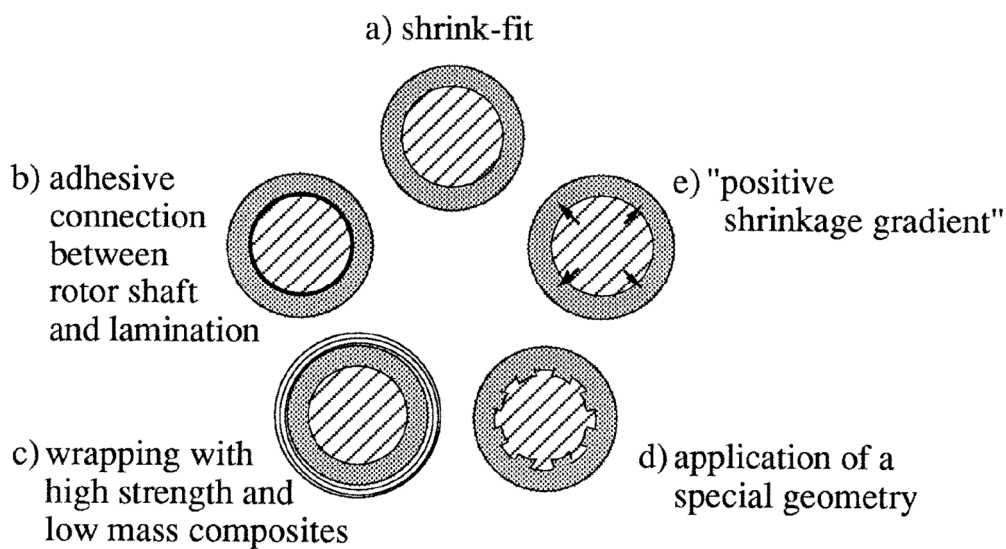


Figure 2-17: Rotor-stator connection methods [22]

a) According to literature, the shrink-fit method is the most classic way to connect the lamination stack to the rotor. It works by manufacturing the inner diameter of the lamination stack smaller than

the outer diameter of the shaft. This means in order to fit the two together, the lamination stack will have to be heated to increase its inner diameter. While cooling the rotor to reduce its outer diameter. The amount of allowable interference between the inner and outer ring is limited by the yield strength of the material as well as the temperature that the laminations can be heated to before shrink-fitting it. For this method, the stress and strain on the material can be derived analytically.

b) The second mounting method is to use an adhesive to stick the two parts together. This method is rather simple but has a few drawbacks. The drawbacks include; the properties of most adhesives change uncontrollably, especially in unfavourable environmental conditions. For instance, when operating at high temperatures. Adhesives also behave optimally under pure shear stress, wherein the case with a rotating rotor, tensile stress needs to be transferred in the radial direction. Thus for this application, an adhesive connection is not ideally suited.

c) The third connection method is to wrap the lamination stack with a pre-stressed carbon fibre or Kevlar. This is not a favourable method as this will increase the effective air-gap by several millimetres. Something that is not desirable as this would increase the amount of mmf needed to provide a certain force on the rotor by the AMBs. This method is generally not appropriate for an internal rotor type but rather used for external rotor machines.

d) The fourth way is to design the shaft and lamination stack in such a way that loss of contact is geometrically prevented, as shown in Figure 2-17d. A disadvantage of the method is that the geometric discontinuities will experience very high-stress peaks that can result in a high risk of cracks forming or resulting in plastic deformations of the material. It will also be difficult to manufacture the components in these shapes.

e) The fifth and final method that will be discussed is called the “positive shrinkage gradient method”. It works by choosing the materials and geometry in such a way that the shaft’s radial displacement increases at a higher rate than the radial displacement of the lamination stack. Meaning as the rotor spins up, the two rings will essentially “shrink” into each other preventing it from losing contact. The biggest drawback of this method is that it is only realizable for well-matched materials and for a very limited geometry. According to literature, the most favourable method to connect the lamination sheets to the rotor’s shaft is to shrink-fit them together [22].

It was decided that the shrink-fit method will be used to connect the laminations to the rotor. This method will be further discussed in the next section.

2.4.1 Shrink-fit

Each infinitesimally small piece of mass of a rotating rotor is subjected to centripetal forces that are equal to $F = m r \omega^2$. With r the radius the mass is located at, and m the mass that is rotating at a speed of ω . This means that in high-speed machines the centrifugal forces will be high; as a result, the stresses in the material will also be high [22]. Due to this, care must be taken to ensure that the stress experienced by the material during the motor's entire operating range does not exceed its yield strength. The yield strength of a material describes the stress that will start deforming the material permanently. Where stress is defined as the amount of force a certain area is experiencing $\sigma = F/A$.

As discussed above, the shrink-fit method will be used to connect the lamination stack to the existing rotor. A shrink-fit is a non-permanent method of connecting a hub to a shaft. It works by manufacturing the inner diameter of the hub smaller than the outer diameter of the shaft that it is connected to. The overlap of the two rings’ dimensions is called the interference between them. The

two parts are fitted together by heating up the outer ring to enlarge its diameter and cooling the inner ring to reduce its diameter. These temperatures depend on the thermal expansion coefficients of the two materials and the amount of interference that was specified for the fit [23].

When different materials are used the interference between the two parts should be measured at the same temperature as a change in temperature will influence the interference. The temperature range that the rotor will operate at should be taken into account, as different materials will have different thermal expansion coefficients. Thus if the rotor expands at a greater rate than the laminations, the interference will essentially increase and will increase the stresses on the material. However, if the laminations expand at a higher rate than the shaft, the interference will essentially decrease, this can cause the rotor to lose contact at high speed [23].

Figure 2-18 shows a ring shrink-fitted onto a shaft, as well as the radial, tangential, and reference stress that is experienced by the two rings. Where the reference stress is the total stress that is experienced by the material.

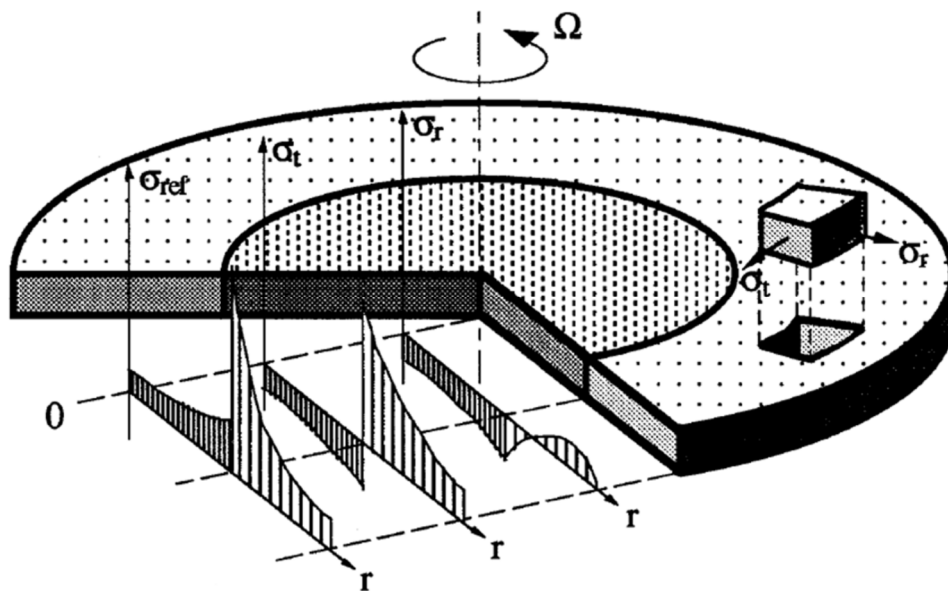


Figure 2-18: Two ring shrink-fit Stress distribution [22]

According to literature, the radial stress at the interface region should remain negative for the entire operating speed of the rotor. If the rotor speeds up too much, the radial stress will start to become positive. At this point, the outer ring will start to lose contact with the inner ring. To increase the speed at which the outer ring starts losing contact with inner ring, one should increase the interference between the two rings. This will, however, increase the stress on the material, which can cause the materials to plastically deform. This will happen if the mechanical stress experienced by the material supersedes that of its yield strength. To ensure that the material does not plastically deform, the stress on the material should be at least half of the yield strength of the material. This is to account for imperfections in calculations such as 3D effects that were not taken into account, or inaccuracies in calculations.

The interference between the two rings should be chosen carefully. It should be low enough to ensure that the stress on the material does not exceed its yield strength. However, it should be high enough

to ensure that the outer ring does not lose contact with the inner ring at the rotor's maximum speed [23],[22].

There exists two types of shrink-fits, an elastic shrink-fit, and a plastic-elastic shrink-fit. In an elastic shrink-fit, the stress on the material never exceeds that of its yield strength. However, if this is not possible due to small dimensional tolerances, a plastic-elastic shrink-fit can be used. With a plastic-elastic shrink-fit, the material experiences stress above its yield strength, which permanently deforms it. However, it should be noted that the effective interference should still be enough to ensure that the outer ring does not lose contact with the inner, while the rotor rotates at its full speed [23].

2.5 SUMMARY

This chapter provided background knowledge that was used when designing the AMBs, that will be retrofitted onto the TWINS. It introduced different topologies of electromagnets that can be used in the construction of an AMB. An alternative method of providing mechanical support to the rotor in case of delevitation, of the rotor was given, namely backup pins. This alternative was necessary as conventional backup bearings uses too much axial space. Background was given on rotor dynamic behavior. This is useful information to understand how the rotor will react while in operation. The last thing that this chapter discussed is different methods that can be used when connecting a lamination stack to a rotor.

CHAPTER 3: DESIGN

This chapter will explain the detail design of AMB's electromagnets. It will start by explaining the specifications of the electromagnets obtained from a rotor dynamic point of view, followed by the detail design of the electromagnets. Lastly, the feasibility of replacing the ball bearing of the TWINS with AMBs will be considered. This will depend on; the reduction of losses produced by the AMBs, if the AMB's journal can be shrink-fitted to the existing rotor of the TWINS and if the dynamic behaviour of the rotor allows for stable suspension of the rotor.

3.1 RADIAL AMB DESIGN

The radial AMB was designed using the design steps of a heteropolar configuration, as outlined in [24]. These design steps were used in a MATLAB® program to allow for easy manipulation of variables. It enables one to determine the effect a change in a certain variable will have on the design output. This enables one to perform trade-offs between things like; the outer diameter of the stator, the number of coil turns and the axial length of the electromagnet.

3.2 PERFORMANCE SPECIFICATIONS AND DESIGN CHOICES

As discussed earlier, the AMBs needs to be designed to levitate the rotor of the TWINS at 30'000 r/min. As the whole purpose of this design is to lower the losses of the ball bearings currently installed in the TWINS. The AMB will need to be designed to ensure that the losses on the rotor's laminations are a lot less than the current losses in the ball bearings. To verify this, the losses will be calculated by using a FEMM analysis program. As discussed earlier, the homopolar configuration produces fewer losses than the heteropolar configuration. Although the purpose of this design is to lower the losses produced, the heteropolar configuration will be used, as it was seen that there is not a lot of axial space available on the rotor to do the retrofit. It was concluded that the homopolar configuration would use a lot more axial space on the rotor than the heteropolar configuration.

The maximum force that the electromagnet needs to produce was determined from the maximum unbalance that will be present on the rotor. An unbalance of G6 was used for the calculation. It should, however, be noted that the rotor needs to be balanced to a G1 standard; this is to guarantee a good safety margin.

A preliminary mass of 4.7 kg was estimated for the rotor. The maximum predicted unbalance force can then be calculated by using (3.1): with m the mass of the rotor, N the speed in r/min and "G" the balancing standard as explained in section 2.3.4.

$$\begin{aligned}
 F_{ub} &= U_{per} \cdot \omega^2 \left[kg \cdot m \cdot \frac{rad}{s^2} \right] \\
 F_{ub} &= 9.549e^{-3} \cdot G \cdot \frac{m}{N} \\
 F_{ub} &= 88.6 \text{ N}
 \end{aligned} \tag{3.1}$$

This force is the total unbalance force that needs to be counteracted by both AMBs. There is however, also a static force that needs to be counteracted namely gravity. The gravitational force produced by the rotor is 46 N, which brings the total force needed by the one AMB to 68 N. With the designing of the AMB, the force the electromagnet needs to produce is specified as, the force that the electromagnet can attract the rotor at the nominal air-gap i.e. if the rotor is situated at the centre of the AMBs. It is known that the force applied to the rotor will decrease quadratically with an increase

in the air-gap. As the rotor will not always be situated at the centre, the force needs to be increased to ensure that the AMB can produce the required force at the maximum air-gap length. It was specified that the rotor's maximum displacement from its centre should not be more than 120 % of the nominal air-gap. Thus if the air-gap is at 120 % of nominal the AMB still needs to be able to produce 68 N.

This means that the force needed to be increased by 1.2^2 , which increases the maximum force (F_{max}) that the AMB's needs to produce in all direction to:

$$\begin{aligned} F_{max} &= 68 \cdot 1.2^2 \\ F_{max} &= 98 \text{ N} \end{aligned} \tag{3.2}$$

After discussing it with experts on the field at the North-West University, it was decided that the AMB will be designed to produce 150 N. This is to leave room for extra rotor dynamic forces that may be percent, an increase in the effective air-gap due to the cutting errors in the laminations and variations in the air-gap due to manufacturing tolerances. The 150 N force was found to be the maximum force that the AMBs can produce with the available power amplifiers while still being able to produce the desired force slew rate.

Other design choices that were made include:

- A 4-pole electromagnet will be used as the diameter of the rotor is quite small. If the rotor has a small diameter, the possible thickness of the coils decreases as the number of poles increase. This will make the assembly harder as there is not a lot of space between the pole tips. Fewer poles also reduces the number of coils that need to be wound. This is an advantage from a manufacturing point of view. The 4-pole configuration will be compared with the 8-pole configuration later in this chapter.
- Pre-wound coils will be used; this will greatly increase the ease of assembly as the coils can be wound outside the stator with a motor. This will also most likely increase the packing factor that can be achieved as the coils can be wound more neatly outside the stator.
- Bias linearization will be used meaning that all the coils will be biased with a biasing current of I_b . This linearizes the force exerted onto the rotor with respect to the control current.
- The rotor laminations inner diameter will be 18.5 mm. This dimension was determined by the current dimensions of the TWINS' rotor. A larger inner diameter would have improved the stresses on the laminations and endstops due to the shrink-fit.
- The outer diameter of the laminations and endstops was chosen to be 63 mm. This is the same maximum outer diameter of the current rotor. The laminations's outside diameter was chosen like that to enable one to slide the rotor out without having to disassemble the motor. It should, however, be noted that the sensors on the side of the rotor need to be removed, before removing the rotor. This is to ensure that the sensors aren't damaged in the process.
- The maximum operating speed of the motor is 30'000 r/min.
- A bearing stiffness of $k_{eq}=500 \text{ N/mm}$ will be used; this parameter can, however, be adjusted during implementation. This value is a conservative value used for the bearing stiffness in AMBs, as found in previous projects [25],[18].
- A bearing damping of $b_{eq}=2.5 \text{ N}\cdot\text{s/mm}$ will be used; this parameter can, however, be adjusted during implementation. This value is a conservative value used for the bearing damping in AMBs, as found in previous projects [25],[18].
- A nominal air-gap of $g_0=0.5 \text{ mm}$ will be used.

3.3 ELECTROMAGNET GEOMETRY CALCULATIONS

Figure 3-1 shows the geometry of a 4-pole radial electromagnet. This section will explain the detail design of the electromagnet. The steps that were used for the design is outlined in [24], it also explains the design of the 8-pole electromagnet, which the 4-pole will be compared to later in this chapter.

r_s stator outer radius

r_c coil radius

r_p pole radius

r_j journal radius

r_r rotor shaft radius

g_0 air-gap

w pole width

n number of poles

l axial length

A_g air-gap area

N number of coil turns

σ Scaling constant

f_s 0.5 for flux splitting / 1 for unsplit flux

β Biasing ratio

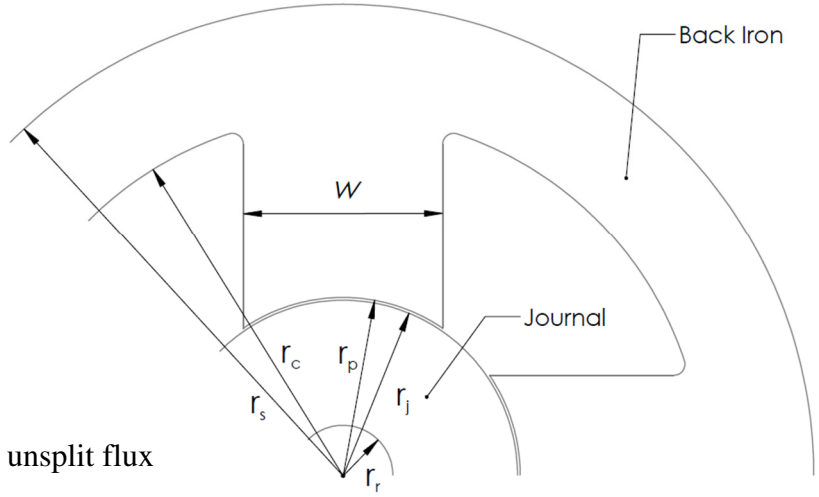


Figure 3-1: Electromagnet geometry

Firstly the air-gap area was calculated by using a chosen pole width (w). The final pole width that was chosen is 36 mm. The pole width was chosen by considering the thickness of the coils that can be slid past the adjacent pole tips as well as the axial length that is taken up by the coils. By increasing the pole width, the axial length of the AMBs can be reduced. Since a certain air-gap area is needed for a desired operating magnetic flux density. However if the pole width becomes too large, the possible coil thickness will reduce, increasing the coil length needed as will be illustrated in Figure 3-4. If the pole width is too short the axial length of the electromagnet will increase. In other words, it's a trade-off between the thickness of the coils, the axial length of the electromagnet, the outer diameter of the electromagnet and the operating magnetic flux density versus available axial space. From this, the air-gap area (A_g) can be calculated as:

$$A_g(l) = w \cdot l \quad (3.3)$$

A lot of different combinations of pole widths and axial lengths (l) can be chosen to produce different results. The axial length of the electromagnet was varied from 15 mm to 25 mm while using the chosen pole width. This was used to calculate the magnetic flux density that is needed for a specific air-gap area to produce the desired force using (3.4).

$$B(A_g) = \sqrt{\frac{2 \cdot F_{max} \cdot \mu_0}{\sigma \cdot n \cdot A_g}} \quad (3.4)$$

The scaling constant σ is 0.25 for a 4-pole configuration and 0.231 for an 8-pole configuration with quadrant coil control. Where n is the number of poles. Figure 3-2 shows how the needed flux density changes as the journal's axial length change. This is for a pole width of 36 mm.

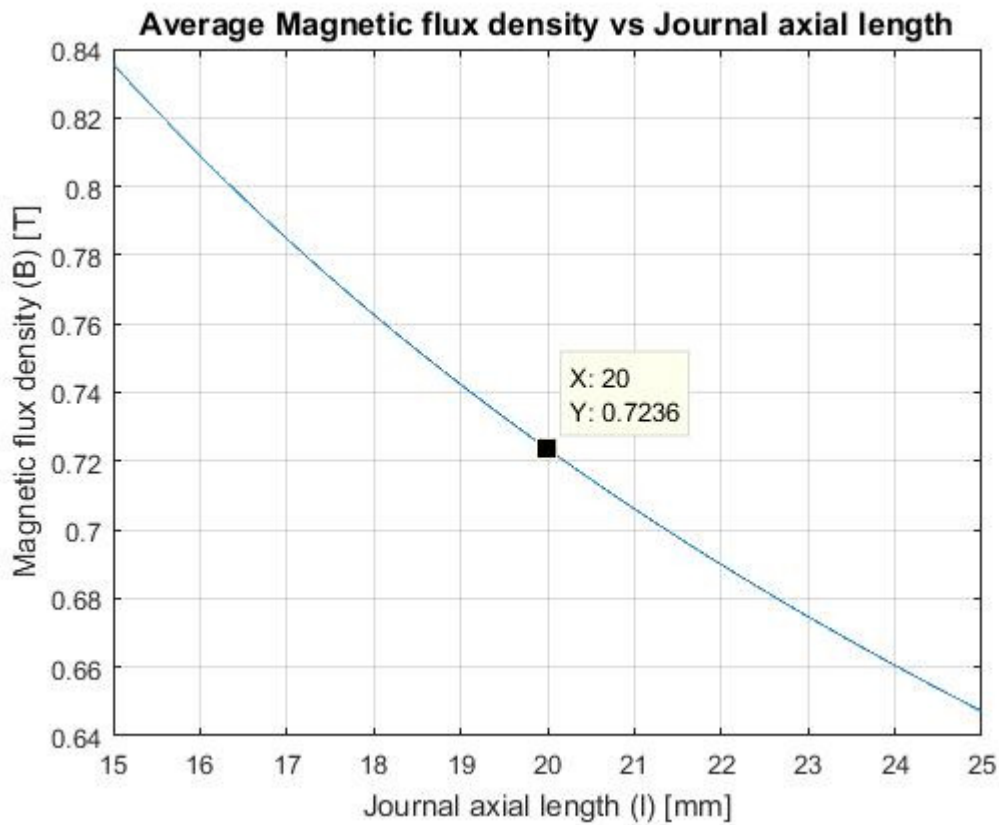


Figure 3-2: Magnetic flux density vs journal axial length

A journal axial length of 20 mm was chosen. This produced a flux density below the saturation point of the material while keeping the axial length short enough.

Next, the mmf needed to produce the desired force can be used to calculate the number of turns that are needed per coil for a specified current. A maximum current ($I_{\max}$) of 5 A was chosen from the specifications of the power amplifiers. The number of turns (N) can be calculated by using (3.5).

$$N(B) = \frac{B \cdot g_0}{\mu_0 \cdot I_{\max}} \quad (3.5)$$

Figure 3-3 shows the number of coil turns that is necessary to produce the desired force. It can be seen that for an axial length of 20 mm, 58 turns is necessary for each coil.

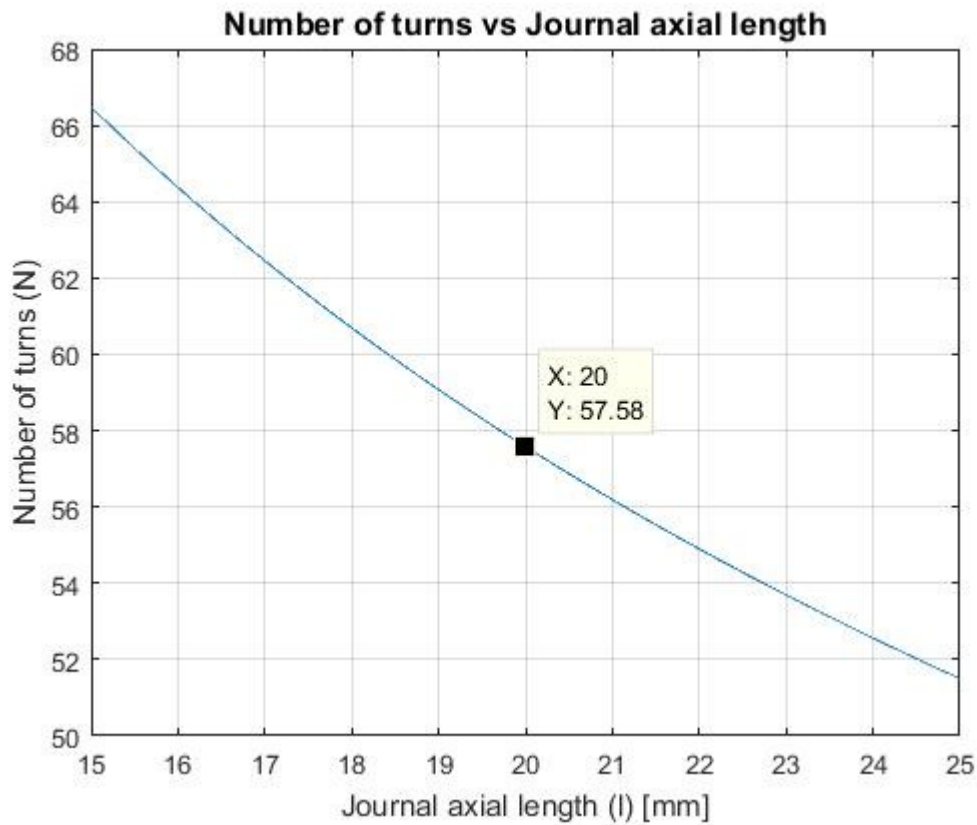


Figure 3-3: Number of turn's vs journal axial length

As discussed earlier, removable coils will be used to simplify the assembly. As shown in Figure 3-4 the coils can be described by the coil length (l_c) and the coil thickness (t_c). The area of the coil is described by $A_v = t_c l_c$.

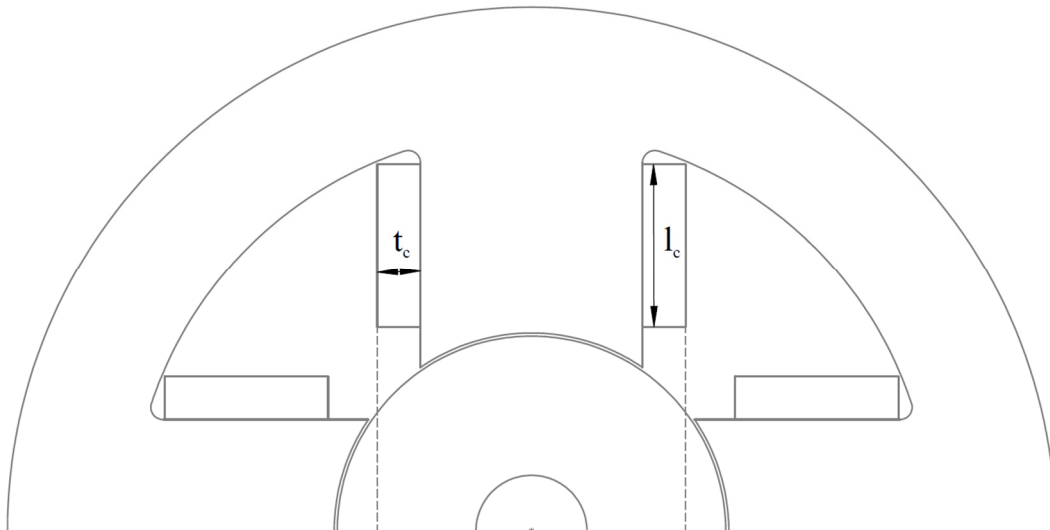


Figure 3-4: Removable coil arrangement

The coil length needs to be calculated to determine the coil radius r_c of the electromagnet. The coil length can be determined by using the area of the wire that is used and the coil thickness that was specified to be 7 mm. This decision was made by considering the chosen pole width as well as the

axial length available for the coil. By using a current density of 3 A/mm² the copper area needed can be calculated by using (3.6). This value was obtained from previous projects. It was found to be a good choice from a thermal point of view.

$$\begin{aligned} A_{wire} &= \frac{I_{max}}{J_{rms}} \\ A_{wire} &= 1.67 \text{ mm}^2 \end{aligned} \quad (3.6)$$

From this, the required wire thickness can be calculated by:

$$\begin{aligned} d_{wire} &= 2 \cdot \sqrt{\frac{A_{wire}}{\pi}} \\ d_{wire} &= 1.46 \text{ mm} \end{aligned} \quad (3.7)$$

However, a wire thickness of 1.46 mm is not a readily available size. It was decided that a 1.4 mm enamel coated wire will be used, since it takes up less space. The current density is only increased to 3.25 A/mm², which not considered to be a major increase.

The wire length that will be needed to turn one coil can be approximate by using the mean length of one turn as:

$$\begin{aligned} l_{coil} &= N \cdot [2 \cdot (t_c + w) + 2 \cdot (t_c + l)] \\ l_{coil} &= 8.12 \text{ m} \end{aligned} \quad (3.8)$$

The length of a coil can be used to determine the resistance of a coil by using a resistivity (ρ) of $1.912 \times 10^{-8} \Omega \cdot \text{m}$ for copper. The resistance is determent as:

$$\begin{aligned} R_{coil} &= \rho \cdot \frac{l_{coil}}{A_{wire}} \\ R_{coil} &= 101 \text{ m}\Omega \end{aligned} \quad (3.9)$$

The nominal inductance of a coil pair in the case of an 8-pole configuration and a coil in case of a 4-pole configuration can be calculated as [24]:

$$\begin{aligned} L_{coil} &= \frac{a \cdot \mu_0 \cdot A_g \cdot N^2}{g_0} \\ L_{coil} &= 6 \text{ mH} \end{aligned} \quad (3.10)$$

With a equal to 1 from a 4-pole configuration and 2 for a 8-pole configuration.

The length and thickness of the coil were used to determine the coil radius r_c of the electromagnet by using SolidWorks®, producing a coil radius of $r_c=65$ mm. The required back iron is used to calculate the stator outer radius r_s . As the magnetic flux from a pole split up to the two adjacent poles (as shown in Figure 3-5) in a 4-pole configuration the back iron should be at least half of the thickness of the pole width. This ensures that the flux in the poles does not reduce due to saturation of the back iron. This produces a stator outer radius of $r_s=85$ mm.

3.4 AMPLIFIER SIZING

The power amplifiers will not be designed for this project, power amplifiers available from another project will rather be used. It should, however, be determined if the available power amplifiers are adequate to be used in this project. The needed force slew rate determines the DC bus voltage that must be used to control the current in the coils.

The maximum force slew rate that the AMBs need to produce is the maximum rate at which the force applied on the rotor needs to be changed. This can be determined by considering the maximum rotational speed of the rotor as well as the maximum force that is expected to be produced by the rotor in the radial direction. If the maximum force is being produced by the rotor at the maximum rotational speed a force vector with an amplitude of 150 N will rotate with the rotor at a frequency of 500 Hz. Meaning that the force in the x -direction can be described by a sine wave:

$$F_x(t) = 150\sin(2 \cdot \pi \cdot f \cdot t) \quad (3.11)$$

The required force slew rate can be determined by differentiating $F_x(t)$ with respect to time. This produces an amplitude of the derived equation as:

$$\begin{aligned} \left| \frac{dF}{dt} \right|_{max} &= F_{max} \cdot 2 \cdot \pi \cdot f_{max} \\ &= 150 \cdot 2 \cdot \pi \cdot 500 \text{ N/s} \\ &= 471.2 \times 10^3 \text{ N/s} \end{aligned} \quad (3.12)$$

The maximum force slew rate is used to determine the maximum VA rating needed by the power amplifiers using (3.13).

$$VA_{max} = \left| \frac{dF}{dt} \right|_{max} \frac{2g_0}{\beta n} \quad (3.13)$$

The symbol β represents the biasing ratio at which the coil current is biased at. This value should be between 0 and 0.5 to ensure that the electromagnet is operated in the linear range. It is given by:

$$\beta = 0.5 \cdot \sqrt{\frac{F_{lin}}{F_{max}}} \quad (3.14)$$

It was chosen that the linear force range (F_{lin}) will be set to its maximum, which is equal to the F_{max} . This produces a biasing ratio of 0.5 and a biasing current of 2.5 A. However, this is not a critical design parameter as it can be changed during implementation. By lowering the biasing ratio, the losses produced in the rotor will also be lowered but will also reduce the force slew rate as well as the linear range. A trade-off will need to be made between the force slew rate and the losses if it is seen that the losses are too high. It was calculated that the maximum voltage V_{max} needed by the power amplifiers is 47.12 V. This means that the available power amplifiers will be sufficient as it can handle a voltage up to 80 V.

3.5 STIFFNESS AND DAMPING

Both a change in the air-gap and current changes the force within the electromagnet. A linearized equation for the instantaneous force in terms of the displacement and current are given by:

$$F(x, i) = k_s x + k_i i \quad (3.15)$$

The change in force due to a change in the position is described by the open-loop stiffness (k_s) measured in N/m. While the change in force due to a change in current is described by the current gain (k_i) and is measured in N/A.

The current gain of a single actuator is used to calculate to control parameters and is defined as:

$$k_i = \frac{\mu_0 \cdot A_g \cdot N^2 \cdot I_b}{g_0^2} \quad (3.16)$$

$$k_i = 30.4 \text{ N/A}$$

This describes the ratio of change in force due to a change in the current in one coil, while the other coils are biased with the biasing current.

The open-loop stiffness of a single actuator for a 4-pole electromagnet is given by:

$$k_s = -\frac{\mu_0 \cdot A_g \cdot N^2 \cdot I_b^2}{g_0^3} \quad (3.17)$$

$$k_s = -152.2e^3 \text{ N/m}$$

This is half the ratio of the change in force to a change in position. Half is taken as the open-loop stiffness is required for a single actuator. It is evaluated with the rotor at its centre and all the coils biased with the biasing current. This is due to the way the required PD constants are calculated to produce an equivalent stiffness and damping.

As discussed in chapter 2 the desired bearing stiffness and damping can be achieved by using a PD controller. The PD control constants can be determined by using:

$$b_{eq} = 2K_D k_i \quad (3.18)$$

$$k_{eq} = 2K_P k_i - 2k_s \quad (3.19)$$

From this, the differentiator gain can be calculated as:

$$K_D = \frac{b_{eq}}{2k_i} \quad (3.20)$$

And the proportional constant can be determined by using:

$$K_P = \frac{\frac{k_{eq}}{2} + k_s}{k_i} \quad (3.21)$$

The PD control parameters can be varied to change the stiffness and damping that the rotor is suspended with.

3.6 FEMM ANALYSIS

A FEMM analysis was used to verify the design of the electromagnet. This included the verification of the: force, open-loop stiffness, current gain as well as the inductance of the coils. The designed parameters of the electromagnet are shown in Table 3-1.

Table 3-1: Electromagnet parameters

Parameter	Value
Shaft radius (r_r)	9 mm
Journal radius (r_j)	31.5 mm
Pole radius (r_p)	32 mm
Coil radius (r_c)	65 mm
Stator outer radius (r_s)	85 mm
Axial length (l)	20 mm
Pole width (w)	36 mm
Number of turns (N)	58
Maximum current (I_{max})	5 A
DC bus voltage (V_{max})	47.12 V

The force was verified by injecting the maximum current in the top coil which is 5 A, switching the bottom coil off and biasing the left and right coil with the biasing current of 2.5 A. This will produce a force in the upward vertical direction. The FEMM simulation of the designed 4-pole electromagnet is shown in Figure 3-5.

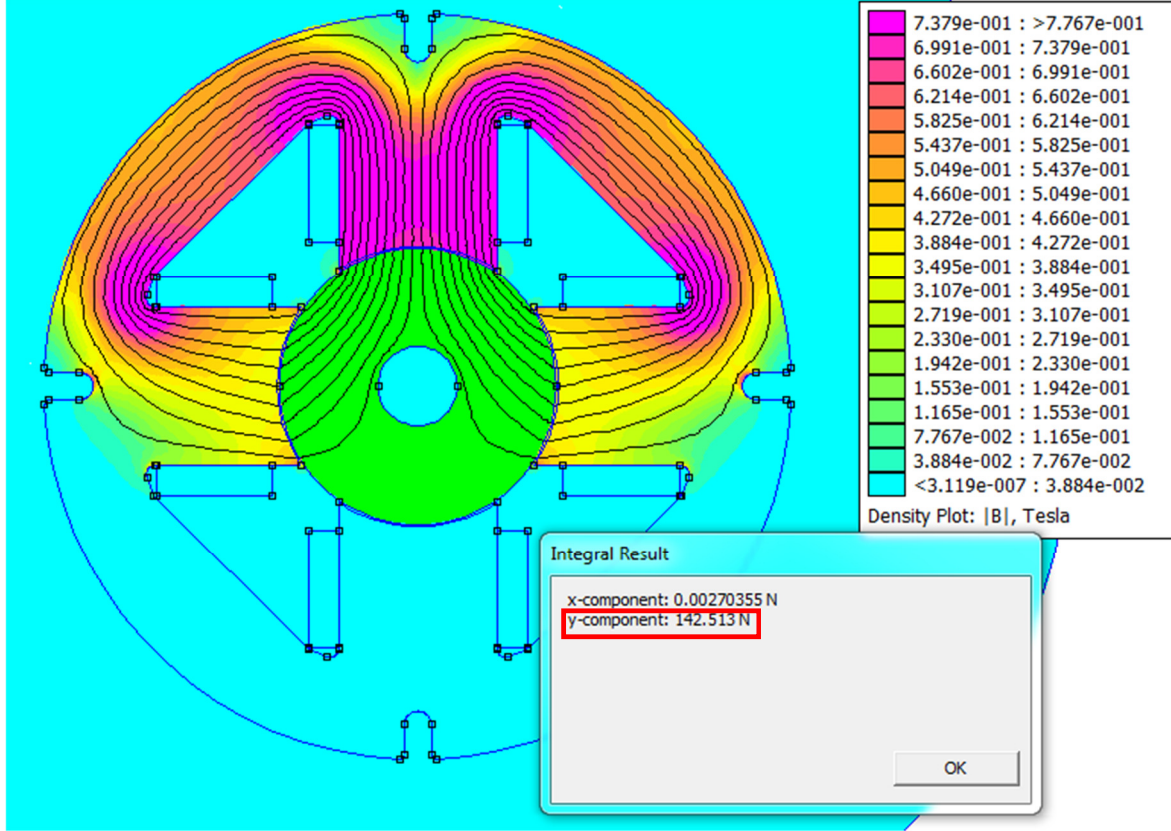


Figure 3-5: Force simulation 4-pole magnetic bearing

It can be seen that the electromagnet produced a force of 142.5 N in the upward direction. This is very close to the maximum force that the AMB was designed for. The drop in force is due to the fact that only the reluctance in the air-gap was considered in the calculations. If the relative permeability of the magnetic material is made very high, the simulated force increases to 152 N.

Next, the open-loop stiffness was verified by biasing all the coils with the biasing current and shifting the rotor 50 μm to the right from its centre position. This changed the force in the x -direction from 0 N to 14.93N. The open-loop stiffness at the centre can be calculated from this as follow:

$$k_s = \frac{\Delta F}{\Delta x} \quad (3.22)$$

$$k_s = 299 \text{ kN/m}$$

Half of this value should, however, be compared to the value obtained in (3.17). This is because (3.17) calculates the open-loop stiffness of one pole and in the simulation, two poles are responsible for the change in force on the rotor. The simulated open-loop stiffness of one pole is 150 kN/m compared to 152 kN/m that was calculated. This change is due to the fact that fringing effects changes the force applied to the rotor; in the calculations the fringing was not taken into account.

The change in the open-loop stiffness with a change in the position of the rotor was investigated. This was done by varying the x -position of the rotor with increments of 20 μm and taking the ratio of the

change in force in x -direction to the change in position. This was done for different displacements away from the centre positions on the y -direction and is plotted in Figure 3-6. Positive and negative displacement in the y -direction will produce the same results.

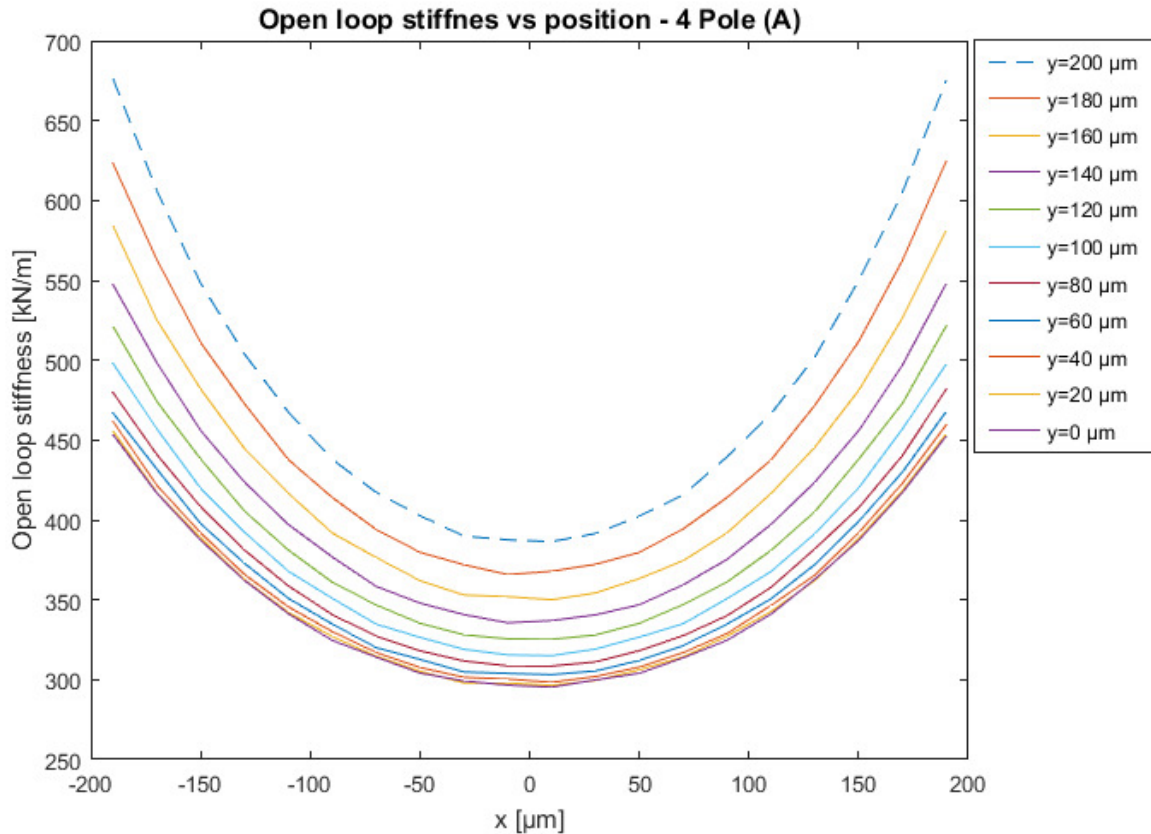


Figure 3-6: Open-loop stiffness (x-direction) vs position

It can be seen that the open-loop stiffness changes as the rotor moves away from its centre position, this will change the equivalent stiffness and damping that is achieved by a specific set of PD controller parameters.

The current gain was verified by placing the rotor in the centre and increasing the current in the top coil by 1 A. This caused an increase of 31.8 N, resulting in a current gain of:

$$k_i = 31.8 \text{ N/A} \quad (3.23)$$

This simulated value of 31.8 N/A correlates closely to the calculated value of 30.4 N/A in (3.16). This change can be attributed to fringing effects.

The inductance value was simulated in FEMM by biasing all the coils with 2.5 A. A screenshot of the FEMM simulation is shown in Figure 3-7.

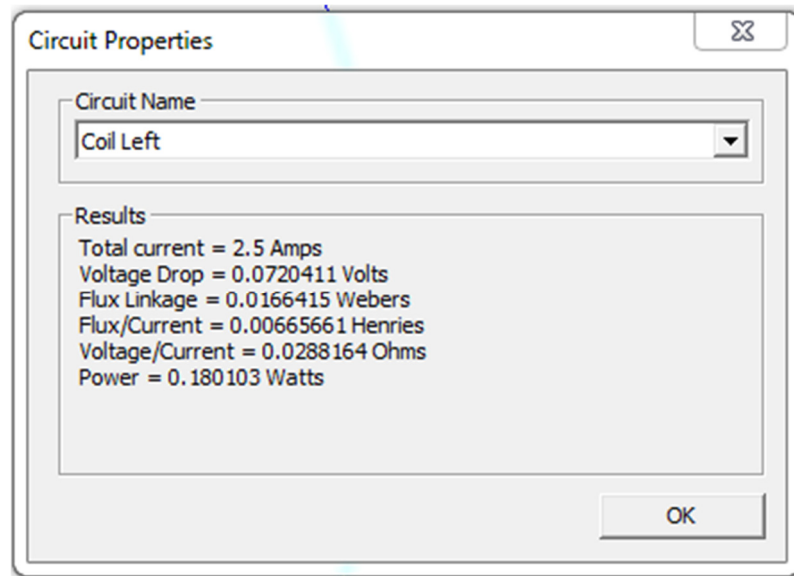


Figure 3-7: FEMM simulation of coil's inductance

It can be seen that the simulated inductance of the left coil is 6.6 mH with the rotor at its centre position and all the coils biased at 2.5A. This compares closely to the calculated value of 6 mH in (3.10). This variation can also be attributed to the reluctance of the magnetic material that was not considered in the analytical calculations, as well as the effects of fringing. The fringing effects change the air-gap area, which changes the magnetic flux density in the pole. This changes the inductance of the coil and the force on the rotor by the pole.

3.7 COMPARISON BETWEEN A 4-POLE AND 8-POLE ELECTROMAGNET CONFIGURATION

It was previously believed by the author that the force in the x and y -direction of the 4-pole electromagnet configuration will be coupled together. This stems from the fact that the coils applying a force in the x -direction share a magnetic flux path with the coils applying a force in the y -direction as can be seen in Figure 3-8. This can result in the applied forces in the x and y -direction being coupled together. Meaning changing the current in the top and bottom coils to vary the force in the y -direction, may also vary the force in the x -direction. This is because the top coil will also control the magnetic flux in the right pole.

However, an 8-pole configuration can be wound to produce a NSSNSSN magnetic polarity sequence for the poles tips. This causes the coils that control the force in the x -direction not to share a flux path with the coils that control the force in the y -direction as can be seen in Figure 3-8. Meaning that the top coil pair will not considerably influence the flux in the left or right coil pair.

According to new literature [26] the forces in the 4-pole electromagnet can be decoupled from each other by controlling the opposing coils differentially. This is done by applying a biasing current to the left and right coils which will result in forces that cancel each other out. To increase the force to the left coil current will be increased in it, while the right coil current will be decreased. The same applies to the top and bottom coils.

The amount of cross-coupling, if any, was investigated by comparing a 4 and 8-pole electromagnet with the same outer diameter and axial length to each. Also, a second 4-pole with the same pole width as that of the 8-pole configuration was compared with the first two. This investigates the influence

the pole shape have on the cross-coupling between the forces in the x and y -direction, as the pole tips of the 4-pole configuration have a larger curvature as that of the 8-pole configuration, as can be seen in Figure 3-8.

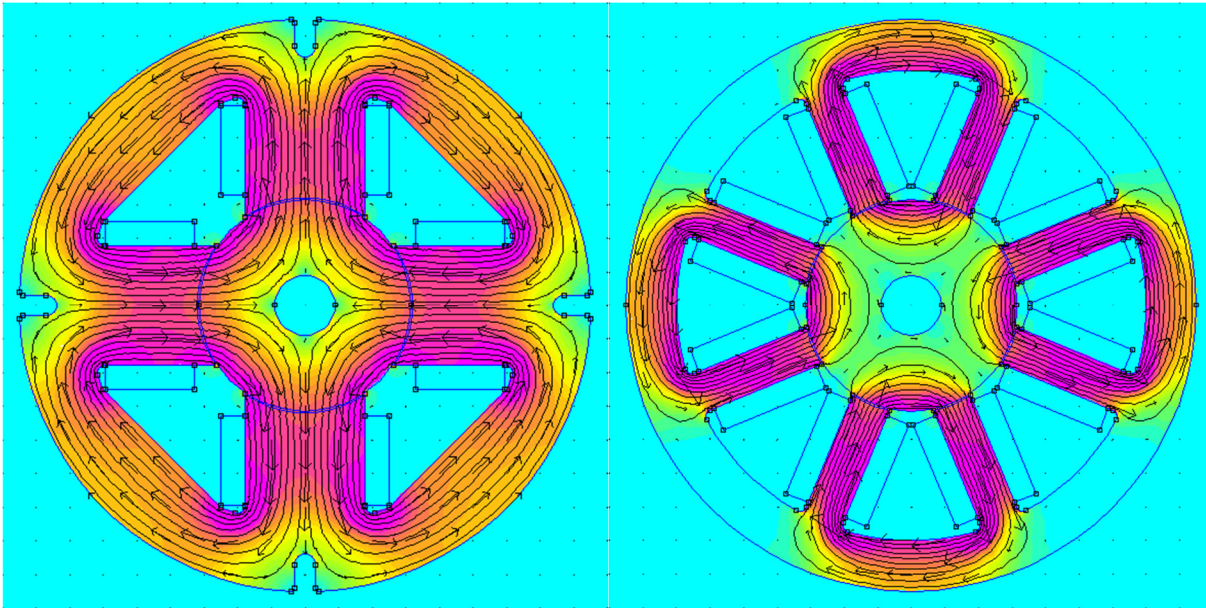


Figure 3-8: Magnetic flux paths 4 & 8-pole configuration

A FEMM simulation was used to compare the 3 configurations. The parameters of the 3 electromagnet configurations are shown in Table 3-2 with the differences in red.

Table 3-2: Electromagnet parameters for compared electromagnets

Parameters	4-pole (A)	4-pole (B)	8-pole (C)
Shaft radius (r_r)	9	9	9
Journal radius (r_j)	31.5	31.5	31.5
Pole radius (r_p)	32	32	32
Coil radius (r_c)	65	65	70
Stator outer radius (r_s)	85	85	85
Axial length (l)	20	60	20
Pole with (w)	36	12	12
Airgap Flux density (B)	0.7236	0.7236	0.922
Number of turns (N)	58	58	74
Coil length (l_c)	26	26	33
Coil thickness (t_c)	7	7	7
Designed force (F_{max})	150	150	150

The cross-coupling was investigated by biasing all the coils with the biasing current and varying the position of the rotor in the electromagnet. If the rotor is moved away from its centre position, it would experience a force to the sides where the air-gap is the smallest. However, if the x and y -axis are truly decoupled from each other, a movement in the y -direction will not affect the force in the x -direction and vice versa. The rotor's x and y -positions were varied from $-200\ \mu\text{m}$ to $200\ \mu\text{m}$ in increments of $20\ \mu\text{m}$. Figure 3-9 to Figure 3-11 shows the force on the rotor in x -directions as the position of the

rotor varies. The x -axis shows the y -position of the rotor, whereas the different x -positions are shown by the different plots on the figure.

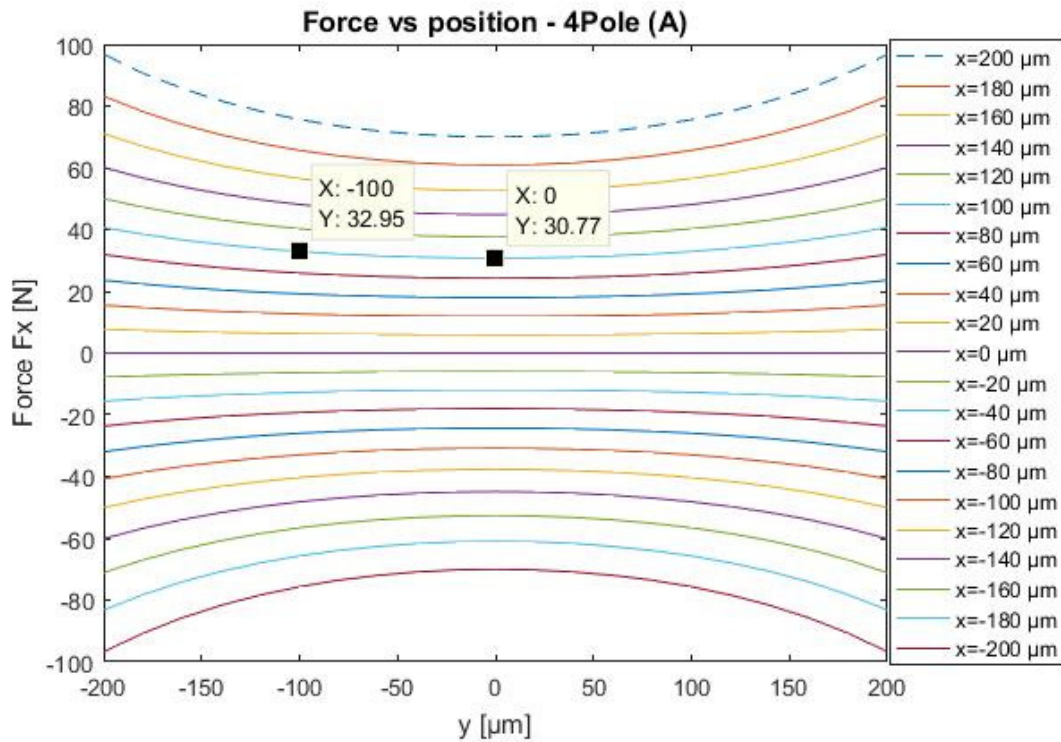


Figure 3-9: Force x-direction vs rotor position (EM A)

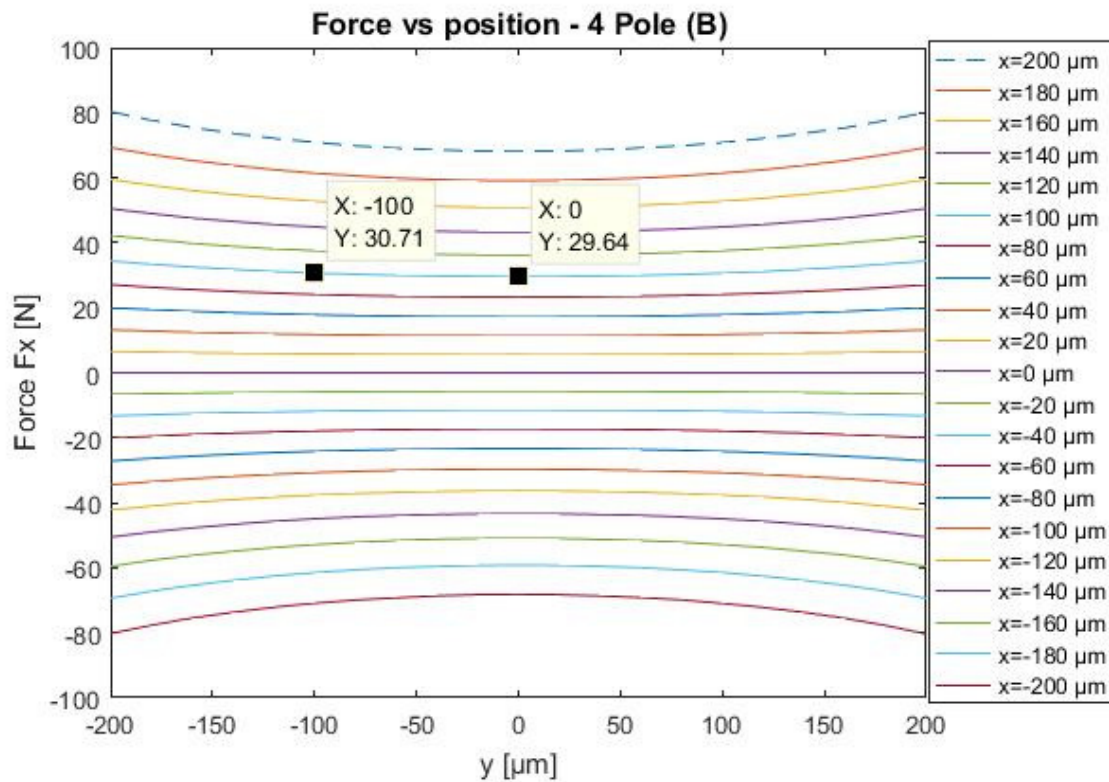


Figure 3-10: Force x-direction vs rotor position (EM B)

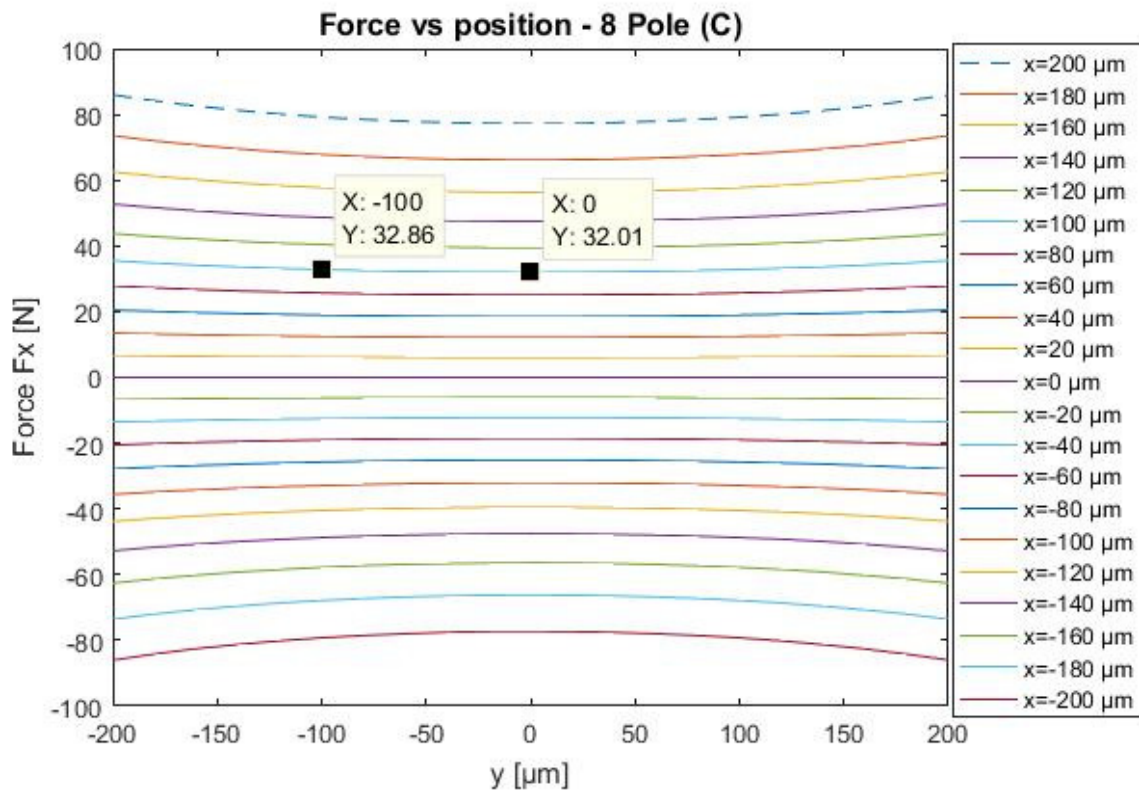


Figure 3-11: Force x-direction vs rotor position (EM C)

It can be seen that a change in the y -position of the rotor also causes a change in the force on the rotor in the x -direction. This change is small if the x -position of the rotor is close to its centre and is increased if the rotor's x -position moves further away from the centre. This happens in both the 4-pole electromagnets and the 8-pole electromagnet. However, the change is smallest in the 8-pole configuration and largest in the 4-pole(A) configuration with a large pole width. It can be seen that the 4-pole(B) configuration with the smaller pole width has less cross-coupling than the 4-pole(A) configuration with the larger pole width. This suggests that the pole tip geometry also have an effect on the cross-coupling; this was also suggested by literature [27]. The percentage change of the force in the x -direction as the y -position changes from 0 to -100 μm with the x -position constant at 100 μm is shown in Table 3-3 for all three electromagnet designs.

Table 3-3: Change in force with rotor position

Electromagnet configuration	x -position (μm)	y -position (μm)	Force F_x (N)
4-pole (A)	100	0	30.77
	100	-100	32.95
	Percentage increase in Force:		7.1%
4-pole (B)	100	0	29.64
	100	-100	30.71
	Percentage increase in Force:		3.6%
8-pole (C)	100	0	32.01
	100	-100	32.86
	Percentage increase in Force:		2.7%

It was also simulated what the change in the force in the x -direction will be, if the current in the top coil is increased with 1 A and the current in the bottom coil is decreased by 1 A i.e. a control current of 1 A. This was done with the x and y -position of the rotor both at 50 μm . The results are shown in Table 3-4. This produced the same results as the previous test, that indicates that the cross-coupling in the 8-pole is least, and most in the 4-pole(A) with the larger pole width.

Table 3-4: Change in force with a change in control current.

Electromagnet configuration	Control current (A)	F _x (N)	F _y (N)
4-pole (A)	0	16.4	16.4
	1	17.85	79.8
Percentage increase in Force		8.8%	386.6%
4-pole (B)	0	14.4	14.4
	1	15	76.68
Percentage increase in Force		4.2%	432.5%
8-pole (C)	0	15.4	15.4
	1	15.8	78.5
Percentage increase in Force		2.6%	409.7%

It can be seen that x and y -axis are not completely uncoupled from each other for all 3 of the electromagnet configurations. It was, however, decided to use the 4-pole(A) electromagnet. Type B EM cannot be used as there is not enough axial space on the rotor to use it. The 4-pole (A) EM was chosen over the 8-pole EM for the following reasons: 1) the construction of the 4-pole is simpler and uses fewer coils, meaning there are fewer coils that need to be wound, 2) all the previous research at the NWU was done on 8 and 12-pole electromagnets, thus investigating the 4-pole EM will add to the expertise 3) the cross-coupling is not expected to have a great effect on the performance of the AMBs. It is expected that the PD controller that controls the y -direction will counteract the force that the coils in the x -direction produce on the in the y -direction.

3.8 LOSSES IN THE ROTOR'S MAGNETIC MATERIAL

As stated in chapter 1 the whole propose of this project is to reduce the losses that is being produced by the bearings of the TWINS. This is to prevent the permanent magnet from demagnetising, if the motor is operated at full speed for continuous periods of time. According to previous research done on the TWINS the losses in its ball bearings are 75 W per bearing. For this reason, it is important to calculate what the losses will be in the magnetic material of the AMBs situated on the rotor. As these losses will be responsible for heating the rotor. There will also be windage losses present on the rotor, which will be caused by air resistance on the journal part of the AMB. This is, however, not of big concern as it will not heat up the AMBs but rater act as an additional load on the motor.

The type of losses that will be present on the journals of the AMBs includes, hysteresis and eddy-current losses. These two losses can be grouped as Iron loss as both will heat up the magnetic material. Eddy-current losses are produced by eddy currents induced in the magnetic material due to the changing magnetic field experienced by the journals of the AMBs. The eddy current loss will be reduced by constructing the AMB's journals with thin laminated magnetic material. The material that was chosen is 35H270 electrical steel that has a thickness of 0.35 mm. This was the thinnest electrical steel that could be found within the time constraints. Hysteresis loss is produced when the flux in the

magnetic material is made to travel on its BH curve, which means that the magnetic flux changes in magnitude or in direction. This will be the case as the journals of the AMBs will continually see a change in the direction of the magnetic flux in it, while the rotor rotates. As well as the fact that the current in the coils will be controlled to levitated the rotor at the desired position. This will cause the magnetic flux in the rotor to change continuously.

The losses in the journal material were determined by using a MATLAB® program written by Dr. David Meeker, it can be found in Appendix B or on his website "<http://www.femm.info>". The program uses FEMM to calculate the hysteresis, eddy current and windage losses experienced by the journals of the AMBs. The results of the loss calculations are shown in Table 3-5. For the sake of completeness, the losses in all the AMB topologies, as discussed in the previous section was calculated.

Table 3-5: Losses in the journals of the different AMB topologies

	4-pole (A)	4-pole (B)	8-pole (C)
Iron loss	8.05 W	11.46 W	12.75 W
Windage loss	16.14 W	48.42 W	16.14 W

As can be seen in the table above the simulated iron losses in the AMBs is 8.05 W, this is considerably lower than the current losses in the TWINS's ball bearing which is 75 W. This implies that it will be worth modifying the TWINS to used AMBs, to solve the problem of its overheating rotor. Comparing the iron losses of the 3 different AMBs, it can be seen that there is not a considerable change in it. However, the iron loss in the chosen design is the lowest, which makes it is the best option. The windage loss is not of great concern, as it does not heat up the journals of the AMBs. The reason design **B** has such a high windage loss is because it has a much longer axial length with the same diameter.

3.9 AXIAL AMB DESIGN

Although it was decided to mount the motor horizontally, an axial AMB was designed to dampen the axial vibration. However, it was discovered that the rotor would have had to be extended. The extension that would have been added to the rotor is shown in Figure 3-12.

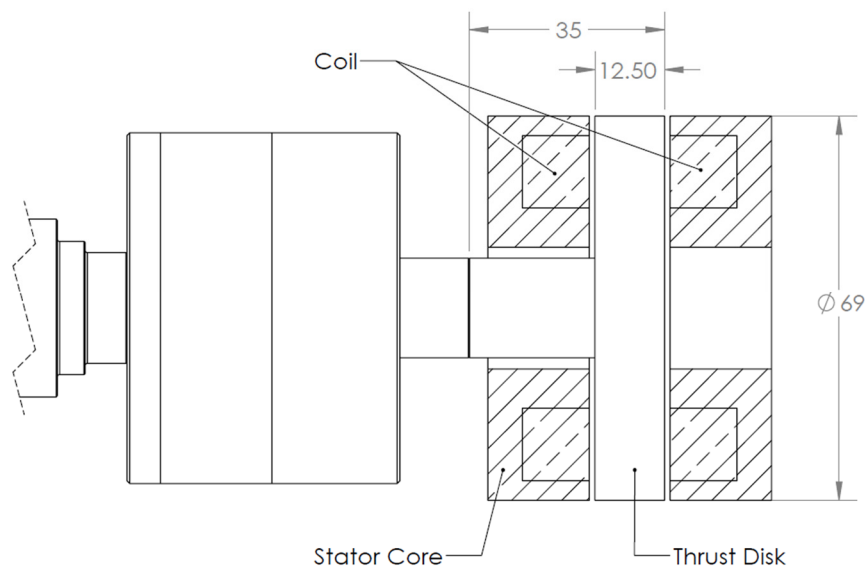


Figure 3-12: Axial magnetic bearing

By adding the axial AMB it was seen that the 1st bending critical frequency was reduced from 1200 Hz to 800 Hz, which is not desirable. The addition of the thrust disk will also make the balancing holes that was inserted into the endstops inaccessible. A better solution would be to just add a permanent magnet with the diameter of the shaft to the rotor. This will not lower the 1st bending critical frequency as the weight of the permanent magnet will be a lot less, and the distance it will add to the rotor will be a lot less. A thorough investigation should, however, be done to ensure that the permanent magnet will be able to withstand the centrifugal forces that it will have to withstand at 30'000 r/min. It should also be checked if the force will be high enough and if there will be enough space to insert a passive axial magnetic bearing. Due to time constraints, it was decided not to add an axial magnetic bearing at this moment, as the TWINS will be operated as a pair. The one TWIN will still be equipped with ball bearings until the retrofit was deemed successful. For the time being the TWIN with the ball bearing will support the rotor in its axial direction.

3.10 SHRINK-FIT

The journal of the AMB consists of magnetic material that needs to be added to the rotor, to enable the rotor to be attracted by the electromagnet. As previously stated, a shrink-fit will be used to connect the journal to the shaft. Endstops will also be added on the sides of the laminations to keep it from buckling as the laminations are only 0.35 mm thick. This section will discuss the results of the stress analysis that was done on the lamination/endstop shrink-fit connection with the shaft of the rotor. This is to ensure that the laminations/endstop assembly, does not lose contact with the shaft while the rotor rotates at 30'000 r/min. It also needs to be ensured that the mechanical stress on the material is lower than the yield strength of the material, to ensure that the material does not break or deform when the shrink-fit is done or when the motor is operating at full speed.

The material that was chosen for the endstops was made on the basis of speed of availability, strength, magnetic properties and cost. The material had to be non-ferromagnetic for two reasons. One; it will be situated next to the laminations, and it is desirable that all the magnetic flux flows in the laminations. As the endstops is a thick piece of metal, any magnetic flux flowing in the material will cause high eddy current losses. Two; the proximity sensors that are currently available are made to

sense on a non-ferromagnetic material. This is important as the position of the rotor will be sensed on the endstops. Stainless steel 304 was chosen for a few reasons, it was easily available, it is non-ferromagnetic and there weren't any stronger non-ferromagnetic stainless steel available. It is also a good thing that it is the same material the rotor shaft is made of. This is because there wouldn't be any difference in the thermal expansion coefficients (α) of the shaft and the endstops. This is important as any variation in α can cause additional mechanical stress on the material as will be explained later in this chapter.

The type of laminations was chosen on the basis of availability, thickness and grain orientation. The thickness of the material was the most important factor as the losses in the rotor were desired to be as low as possible. The thinnest non-grain orientated material that was available in South Africa was 35H270 electrical steel. Non-grain orientated electrical steel was chosen as the rotor and stator will experience changes in magnetic flux in all direction. Grain-oriented electrical steel is more suited for the manufacturing of transformers.

The material specifications of the shaft, endstops and laminations are shown in Table 3-6. These material specifications were used to calculate the total stress on the material, radial stress as well as the displacement of the material after a shrink-fit, by using the MATLAB® program “*Shrinkfit_Cal.m*” included in Appendix B.2. The method for calculating the amount of interference required between the outer diameter of the shaft and the inner diameter of the laminations/endstop assembly used in “*Shrinkfit_Cal.m*”, can be found in [22].

Table 3-6: Summary of material properties

Material	S_y Yield strength (MPa)	E Modulus of elasticity (GPa)	ρ Density (kg/m ³)	ν Poisons ratio	α Thermal expansion ($\mu\text{m}/\text{m}\cdot^\circ\text{C}$)
Rotor Shaft/ Endstops (Stainless steel AISI 304)	215	195	8000	0.27	17.3
Laminations (35H270)	410	185	7650	0.3	12

A summary of the dimensions and tolerances that the shaft and lamination/endstop assembly will be manufactured to is shown in Table 3-7. It was seen from the stress analysis that the interference needs to be kept to a minimum to ensure that the material does not permanently deform, when the machine is in operation. The results of the stress analysis will be shown later in this section.

Table 3-7: Component dimensions

Component	d_i (mm)	d_o (mm)	Interference (μm)
Shaft		18.5 / 18.5+0.01	
Laminations/Endstops	18.49 / 18.49-0.01	63	10 / 30

Due to the manufacturing techniques available, the smallest increment these parts can be manufactured to is 10 μm . Because there are two parts that need to be machined, the minimum possible change in the interference due to tolerances is 20 μm , as both parts can only be made to within 10 μm of its specified value. This means that the minimum interference possible between the two components is between 0 and 20 μm . This is, however, a problem as it is possible that the interference is 0 μm which will result in a loss of contact between the shaft and lamination/endstop

assembly. The next possible range to define the interference is between 10 and 30 μm . This was found to be an acceptable interference range by conducting a stress analysis. The results of the stress analysis for this range will be explained next.

The results of the analysis can be seen in Figure 3-13 to Figure 3-26. While doing the analysis, one should ensure that the radial stress on the material at the interface between the two rings should be negative at maximum rotational speed, indicating that there is no loss of contact between the two rings. A negative value means that the radial stresses direction is towards the centre point. Loss of contact can also be seen from the displacement of the materials at the interface between the two rings. If the displacement of the material at the outer radius of the inner ring is negative, the outer ring is still forcing pressure onto the inner ring. Also, one should consider the mechanical stress that is experienced by the material. The stress should be lower than the yield strength of the material. The yield strength of the material defines the stress where the material will start to experience permanent deformation.

Firstly it was checked that the endstops would not lose contact with the shaft at 30'000 r/min with a minimum interference (Int) of 10 μm . The results of the radial stress and displacement under these conditions is shown in Figure 3-13 and Figure 3-14

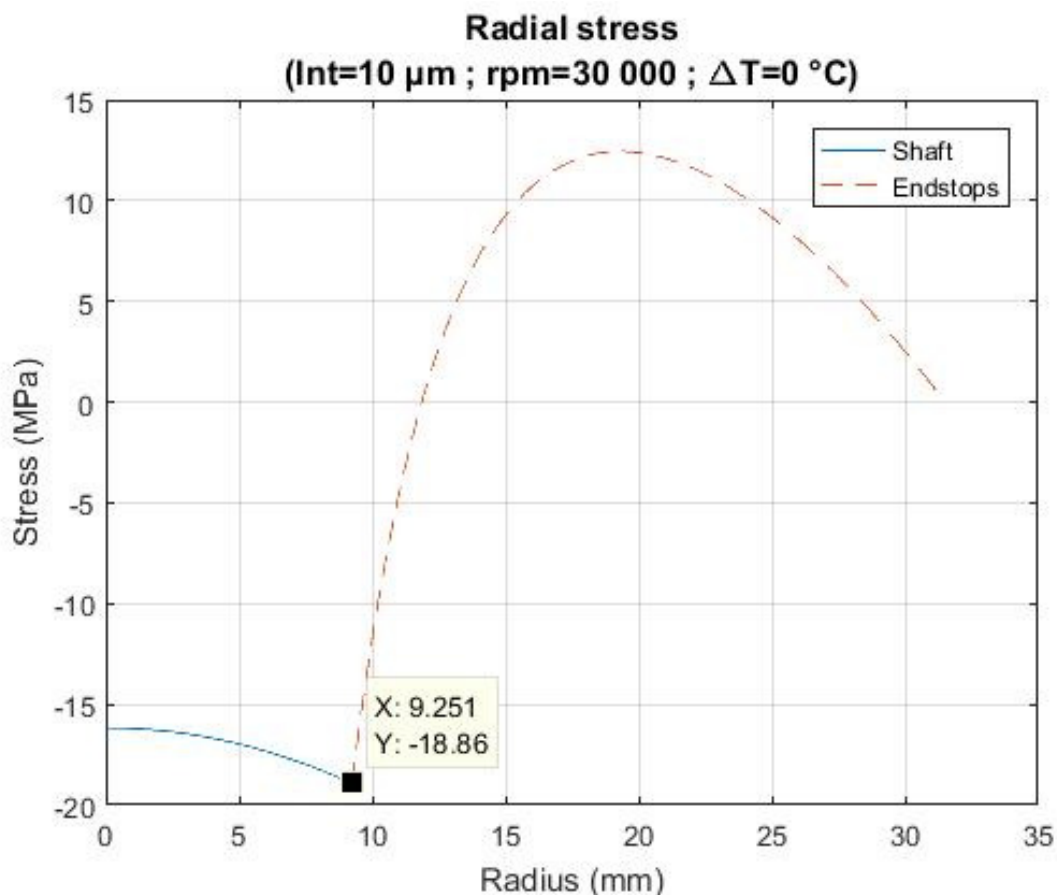


Figure 3-13: Endstops: Radial stress at Int=10 μm , speed= 30'000 r/min

It can be seen that the radial stress is negative meaning that there is a force to the centre point at the interface. This means that the endstops will not lose contact while the rotor rotates at 30'000 r/min.

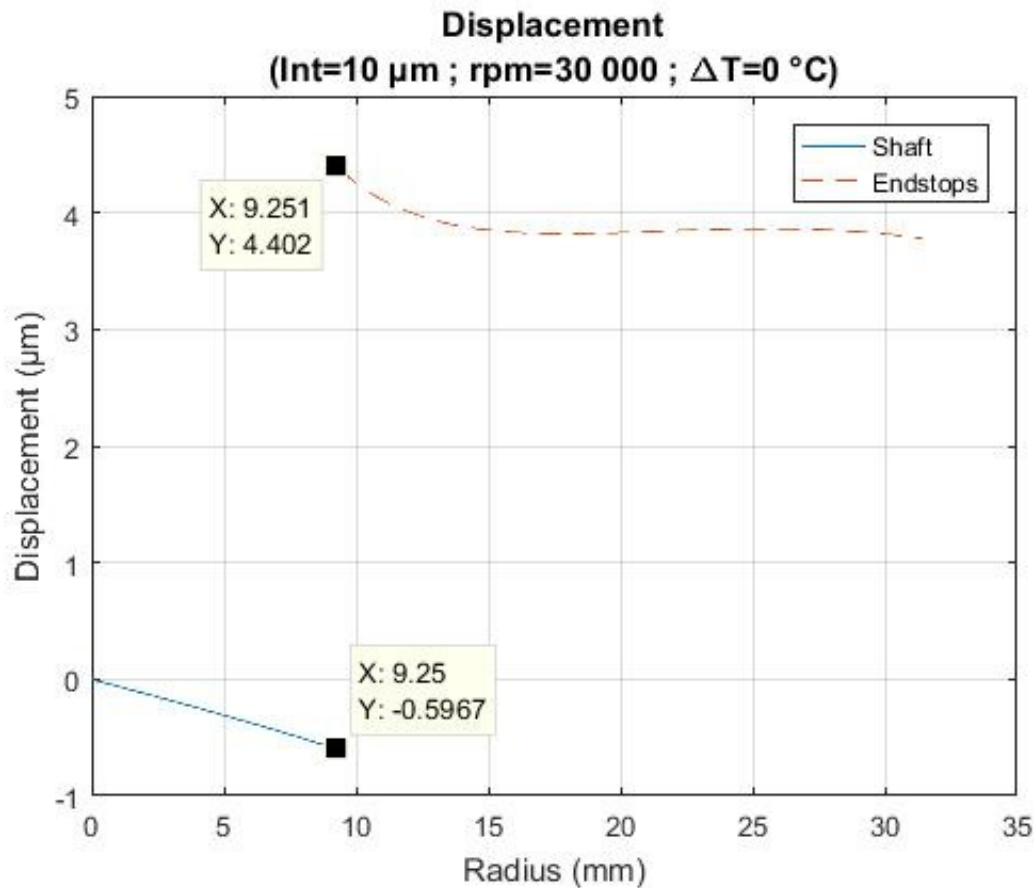
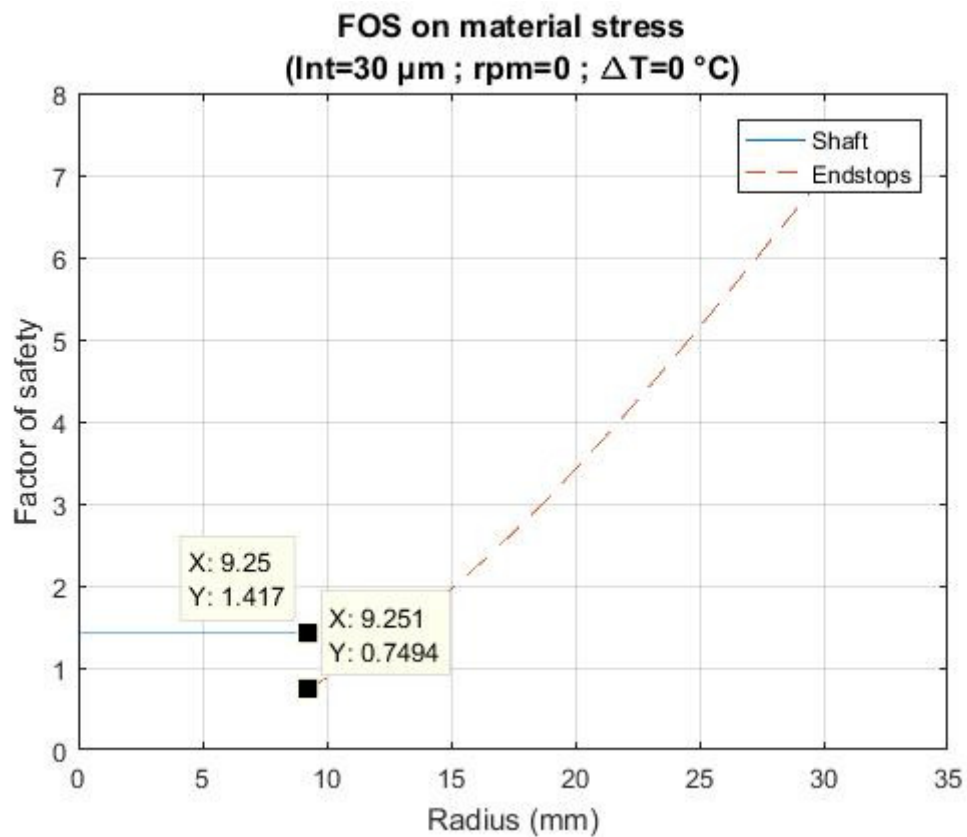
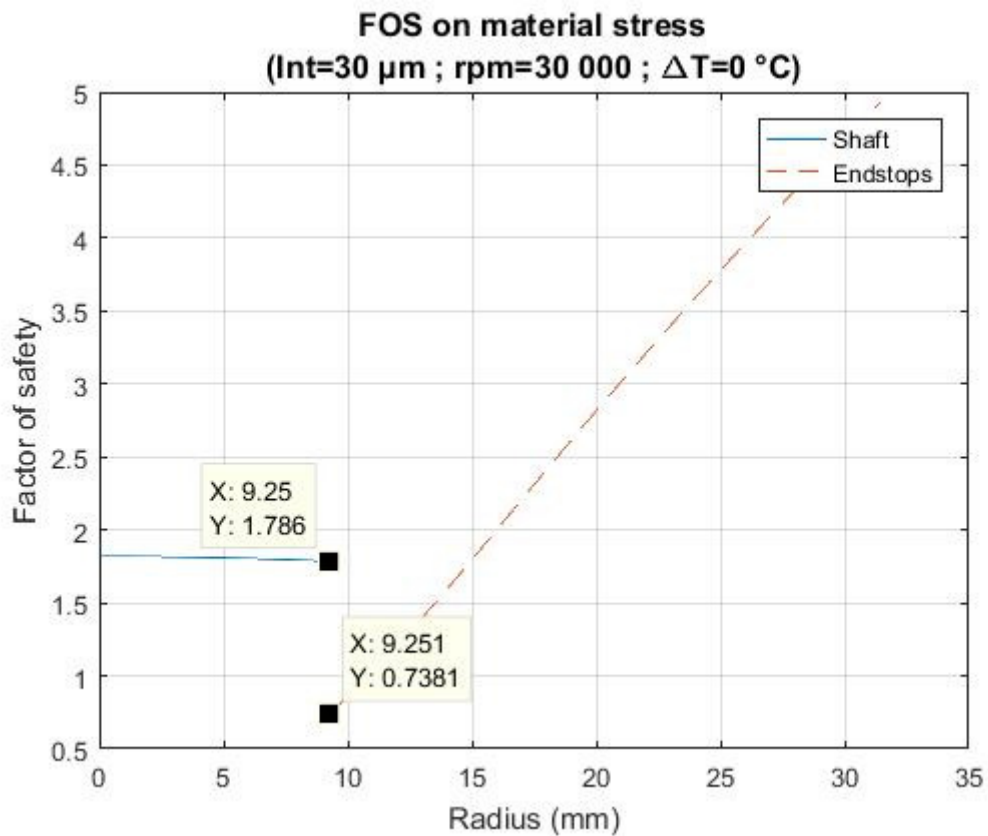


Figure 3-14: Endstops: Displacement at Int=10 μm , speed= 30'000 r/min

It can also be seen from Figure 3-14 that the displacement of the shaft at its outer diameter is negative this means that the endstops are still pressing on the shaft due to the interference fit. This is because the shaft is smaller than it was before the fit was done. The shaft will thus not lose contact with the endstops at 30'000 r/min according to the results shown in Figure 3-14. It is not necessary to calculate the radial stress and displacement of the material with a maximum interference as this will result in higher inward pressure for the endstops.

Next, the stress that the material will encounter due to the shrink-fit and centrifugal forces was calculated and is shown in Figure 3-15 and Figure 3-16, for a rotational speed of 0 and 30'000 r/min respectively at a maximum interference of 30 μm . The stress was expressed as a ratio to the material's yield strength. This is called the factor of safety (*FOS*) as it shows, how much the stress on the material is relative to its yield strength. For example, a *FOS* of 2 will indicate that the stress experienced by the material is half that of its yield strength.

Figure 3-15: Endstops: Factor of safety at Int=30 μm , speed= 0 r/minFigure 3-16: Endstops: Factor of safety at Int=30 μm , speed= 30'000 r/min

It can be seen that the stress experienced by the endstops is more than its yield strength, meaning that plastic deformation will occur. This is not a good thing as this will decrease the effective interference between the shaft and the endstops, which will decrease the radial force that is holding the two rings together. However, it was shown in Figure 3-13 and Figure 3-14 that the two rings would stay in contact at 30'000 r/min with a minimum interference of 10 μm . This means that the endstops and shaft just should not deform as much as to give an effective interference of less than 10 μm .

The stress on the shaft and endstops material at a minimum interference of 10 μm are shown in Figure 3-17 and Figure 3-18.

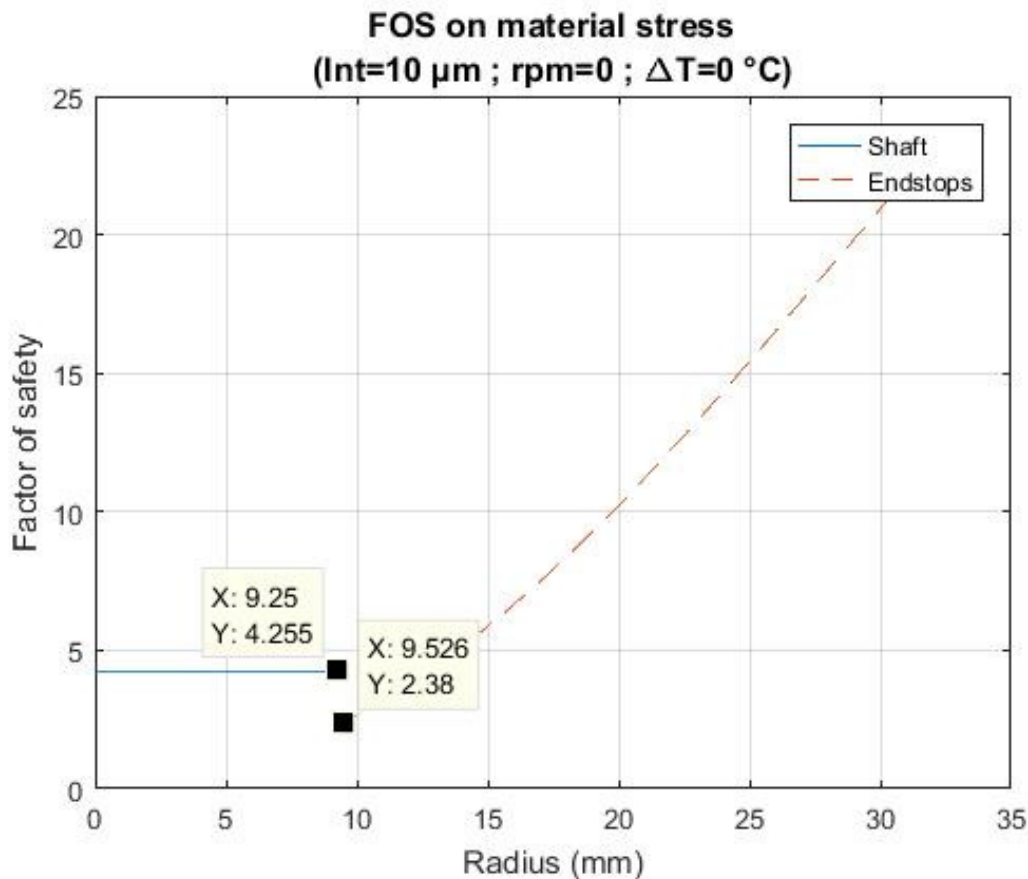


Figure 3-17: Endstops: Factor of safety at Int=10 μm , speed= 0 r/min

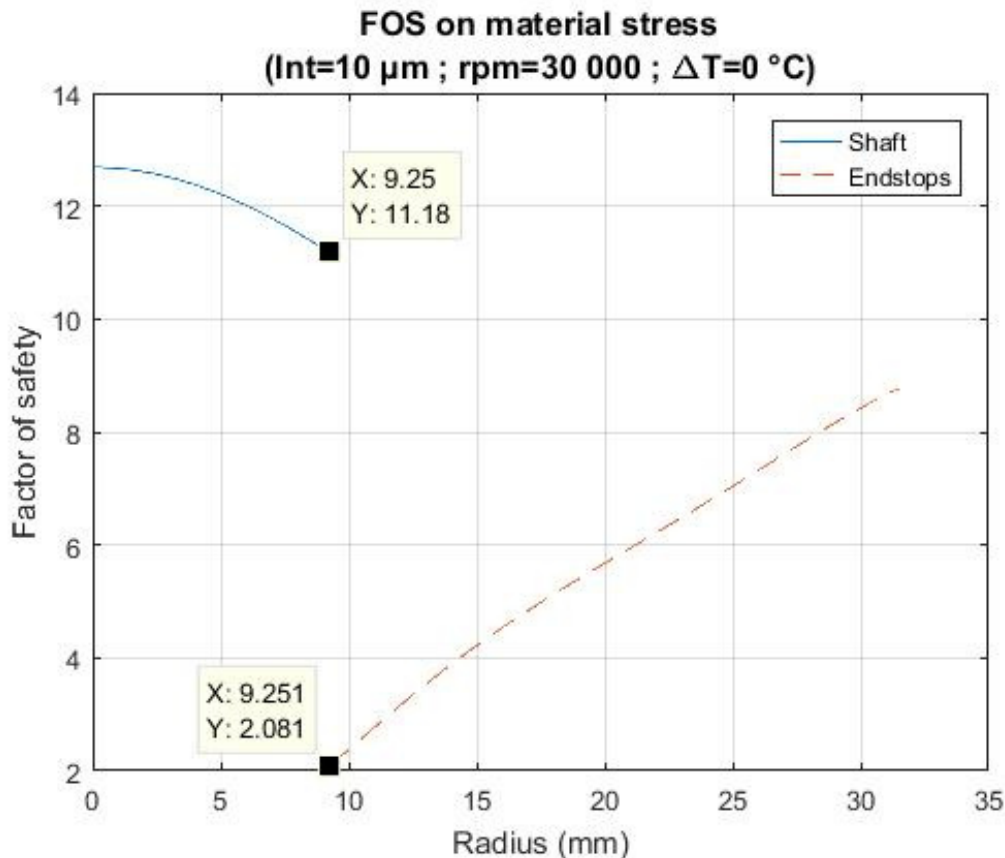


Figure 3-18: Endstops: Factor of safety at Int=10 μm , speed= 30'000 r/min

It can be seen in both cases, with the speed of the rotor at 0 and at 30'000 r/min, that a *FOS* of at least 2 is being maintained. From this, it can be concluded that the endstops and shaft will not deform to create an interference of less than 10 μm , if the two parts were manufactured to have an interference of 30 μm . This is because as the material plastically deforms the stress on it also decreases. This type of fit is called a plastic-elastic fit, as explained in section 2.4.1.

As the endstops and shaft are made of the same material, it will expand at the same rate with an increase in temperature. Due to the fact that the two rings are so tightly bound together, the two parts will experience the same increase in temperature at the interface point. This means that the material at that point will expand at the same rate, effectively keeping the interference between the two points constant. For this reason, an increase in the temperature doesn't have to be taken into account. The same calculations were done to ensure that the laminations will be able to be shrink-fitted to shaft. The results will be explained next.

The radial stress and material displacement of the laminations shrink-fitted to the shaft is shown in Figure 3-19 and Figure 3-20, respectively.

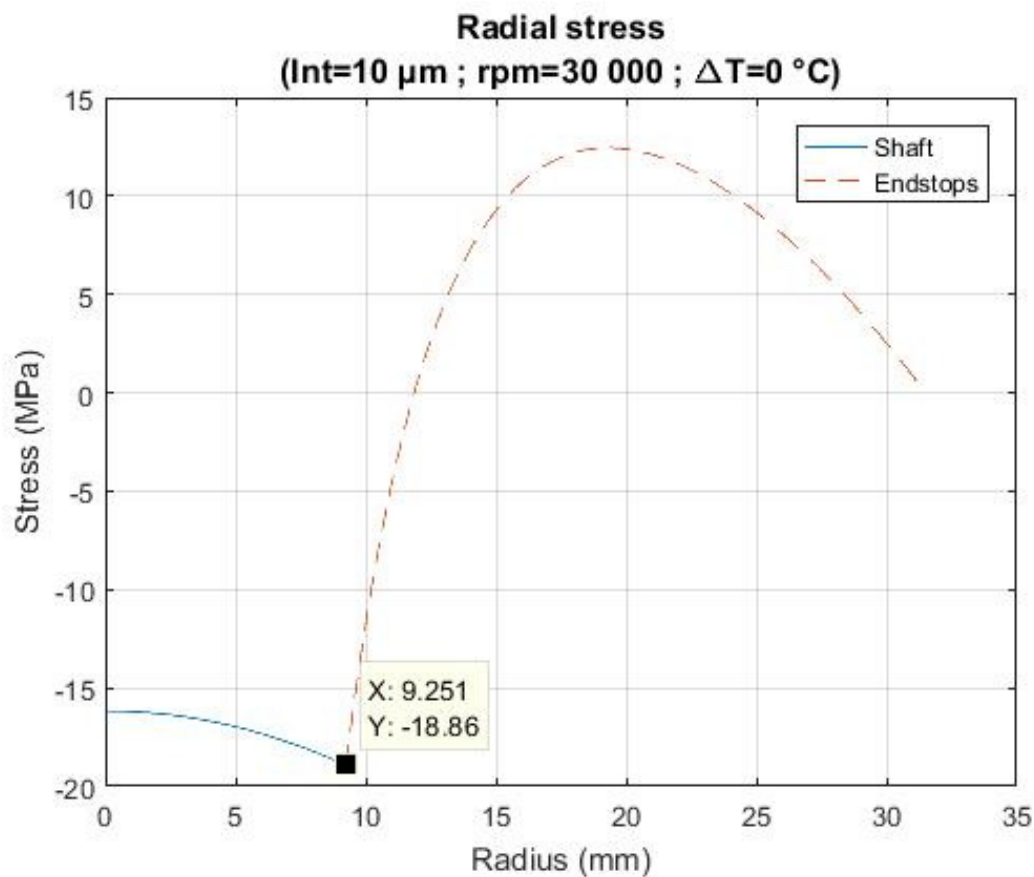


Figure 3-19: Laminations: Radial stress at Int=10 μm , speed= 30'000 r/min

It can be seen that the radial stress at the interference point, still remains negative while the rotor rotates at 30'000 r/min. This means that for a minimum interference of 10 μm between the two rings, that they will still stay in contact if the rotor rotates at 30'000 r/min. As explained in the results of the shaft-endstops stress calculation results.

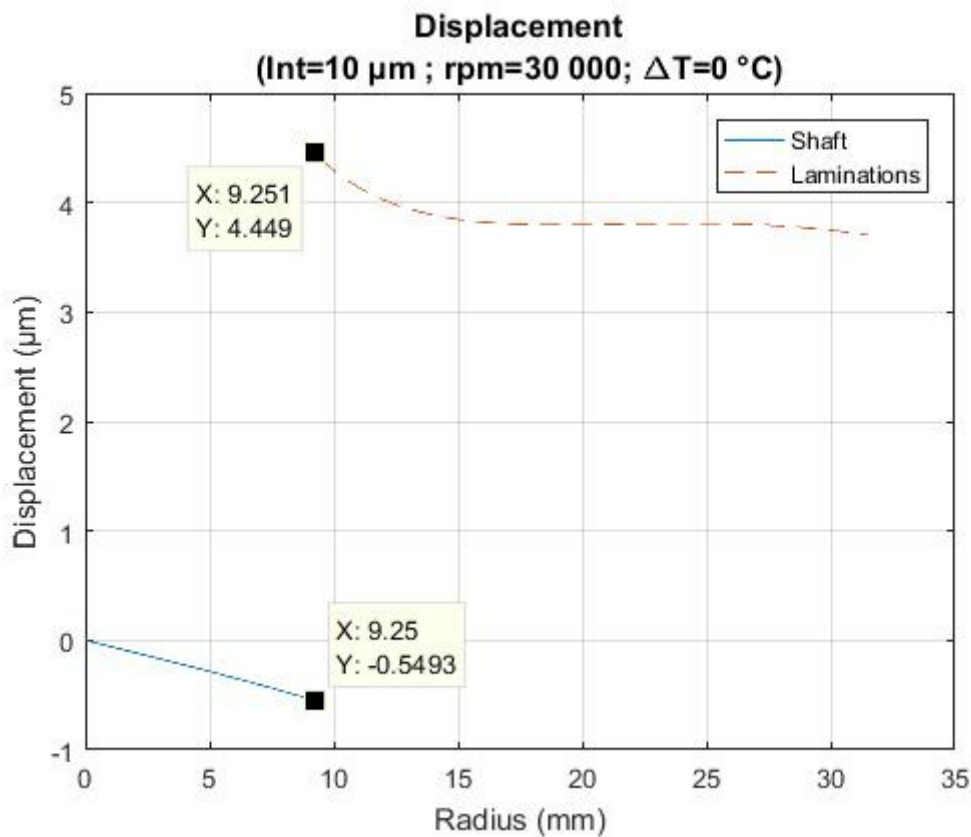


Figure 3-20: Laminations: Displacement at Int=10 μm , speed= 30'000 r/min

The same thing can be seen from the displacement of the material at the interface region. The displacement of the shaft at its outer diameter is negative at a rotational speed of 30'000 r/min, with an interference of 10 μm . This means that the laminations will still remain in contact with the shaft at that speed.

As can be seen in Figure 3-21 and Figure 3-22 below the stress experienced at both rings is less than the yield strength of the material. However, according to the literature, the stress experienced by the material should be at least half of the yield strength of that material [22]. This is to ensure that the material will not experience plastic deformation. However, it can still work as a plastic-elastic shrink-fit.

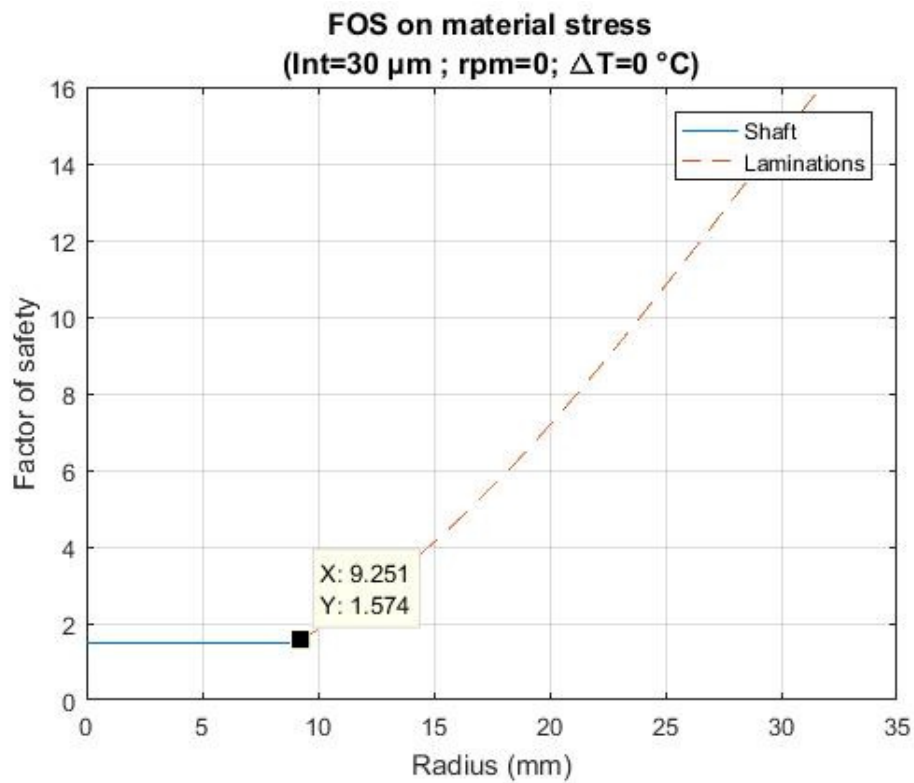


Figure 3-21: Laminations: Factor of safety at Int=30 μm , speed= 0 r/min

Figure 3-21 shows that the material might experience plastic deformation if a interference of 30 μm exists between the two rings, while the rotor is at a standstill.

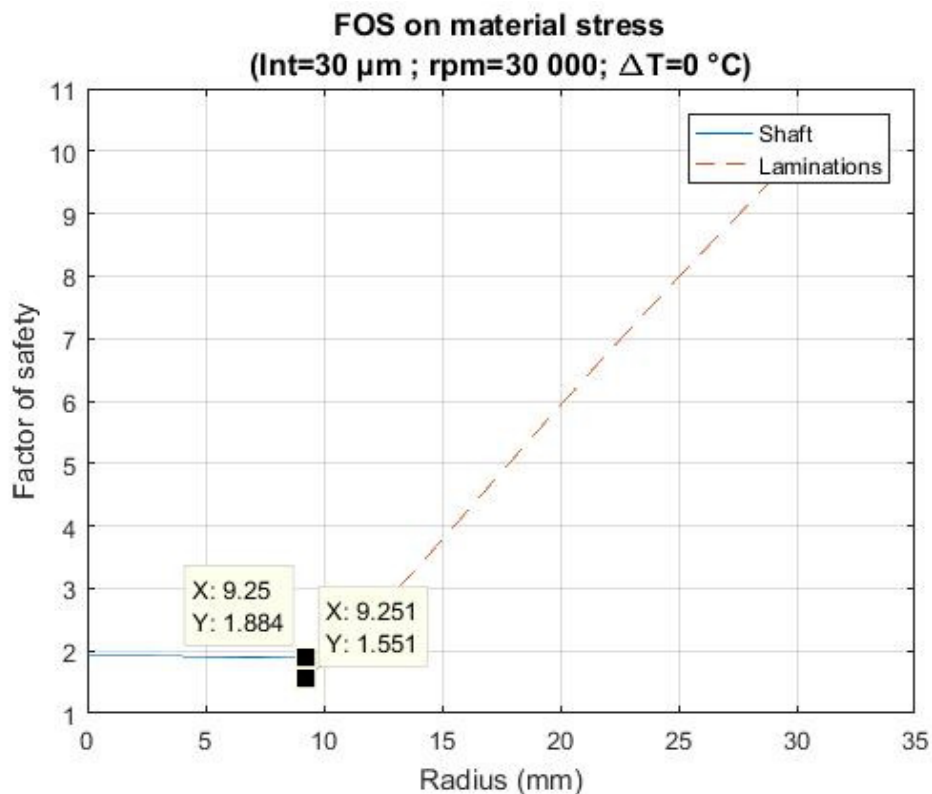


Figure 3-22: Laminations: Factor of safety at Int=30 μm , speed= 30'000 r/min

Figure 3-22 shows that the material might experience plastic deformation if an interference of $30\text{ }\mu\text{m}$ exists between the two rings, while the rotor rotates at $30'000\text{ r/min}$. However, it was shown in Figure 3-19 and Figure 3-20 that the laminations would not lose contact with the shaft while the rotor rotates at $30'000\text{ r/min}$, if there was a minimum interference of $10\text{ }\mu\text{m}$ between them.

Figure 3-23 and Figure 3-24 shows that with an interference of $10\text{ }\mu\text{m}$, the materials of the two rings will not experience plastic deformation. This is true for a rotational speed of 0 and $30'000\text{ r/min}$.

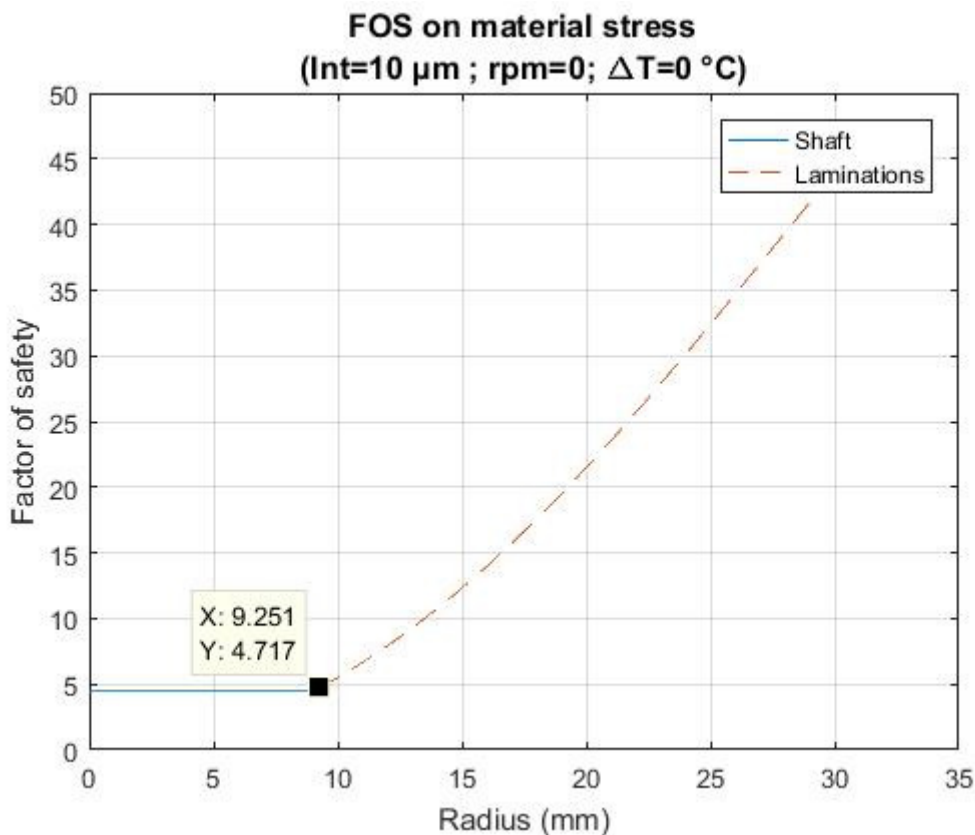


Figure 3-23: Laminations: Factor of safety at Int= $10\text{ }\mu\text{m}$, speed= 0 r/min

As can be seen in Figure 3-23 the stress experienced by the two rings is less than half of their respective yield strengths. This means that if the material deforms at 0 r/min, it will not deform as much as to produce an effective interference of less than $10\text{ }\mu\text{m}$.

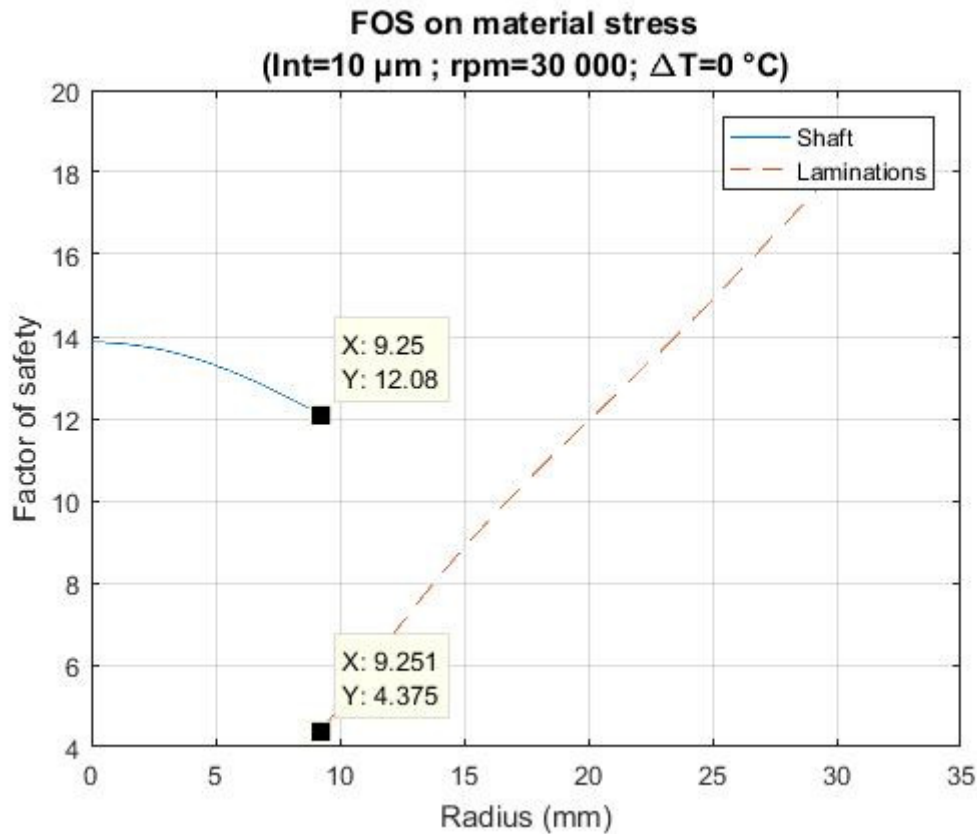


Figure 3-24: Laminations: Factor of safety at Int=10 μm , speed= 30'000 r/min

As can be seen in Figure 3-24 the stress experienced by the two rings are less than half of their respective yield strengths. This means that if the material deforms at 30'000 r/min, it will not deform as much as to produce an effective interference of less than 10 μm .

As stated in Table 3-6 the thermal expansion coefficient of the shaft's material is 17.3 $\mu\text{m}/\text{m}\cdot^\circ\text{C}$, whereas the thermal expansion coefficient of the lamination's material is 12 $\mu\text{m}/\text{m}\cdot^\circ\text{C}$. This means that the shaft will expand more than the laminations. Due to this the effective interference between the 2 rings will effectively increase as the temperature rises. Meaning that the radial stress, holding the two rings together, will increase. In other words, an increase in the temperature will not result in a loss of contact between the two rings. However, it will increase the stress experienced by the material.

The stress on the material was calculated for a change in temperature of 55 $^\circ\text{C}$. This is because the machine was not designed to operate at a temperature above 80 $^\circ\text{C}$ as stated in Chapter 1. The previous calculations assumed that the parts would be machined to produce the specified dimensions at 25 $^\circ\text{C}$.

The results of the stress calculations with the shaft and laminations at 80 $^\circ\text{C}$ and an interference of 10 μm is shown in Figure 3-25 and Figure 3-26. These results are shown for the rotational speed of the rotor at 0 and 30'000 r/min respectively.

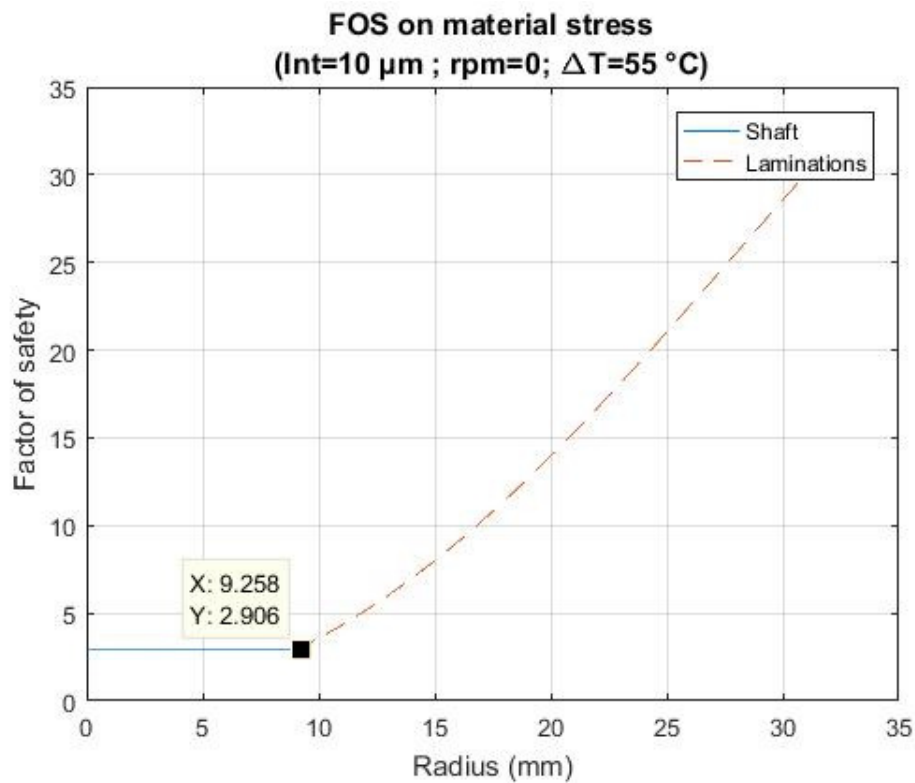


Figure 3-25: Laminations: Factor of safety at Int=10 μm , speed= 0 r/min $\Delta T=55^\circ\text{C}$

From Figure 3-25 it can be seen that the FOS is still above 2. This means that the two rings will not experience permanent deformation if the rotor heats up to 80°C .

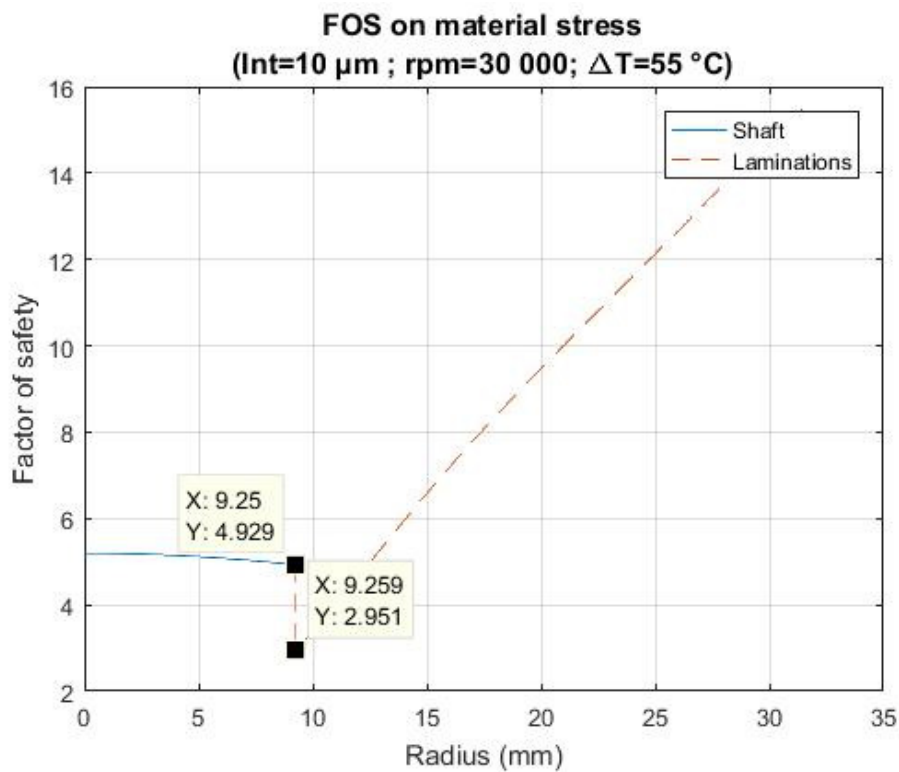


Figure 3-26: Laminations: Factor of safety at Int=10 μm , speed= 30'000 r/min $\Delta T=55^\circ\text{C}$

From Figure 3-26 it can be seen that the FOS is still above 2. This means that the two rings will not experience permanent deformation if the rotor heats up to 80°C.

From the above-discussed analysis, it was concluded that it would be possible to shrink-fit the endstops and laminations to the rotor's shaft, with the defined tolerances and dimensions.

3.11 ROTOR DYNAMICS

A rotor dynamic analysis was done to determine if the critical frequencies of the rotor falls within the operational speed of the motor. The critical frequencies of the rotor were determined analytically and by using simulation. The simulation package that was used is called RotFE. It is a MATLAB® toolbox written by Izhak Bucher. The mass and mass moment of inertia was simulated by using SolidWorks®. The analytical calculations used were explained in section 2.3.2.1 and 2.3.2.2. For the rigid mode calculations, the simplified equation can be used, this is (2.15) and (2.16). This is due to the fact that the rotor centre of mass is located in the centre of the bearing locations. The results are shown in Table 3-8.

The 1st and 2nd rigid modes were calculated for a bearing stiffness of $k=500$ N/mm, were the 1st bending mode was calculate for a bearing stiffness of infinity i.e. a rigid bearing. The calculation was not done as this required a more in-depth analysis into rotor dynamics, which was not necessary. This will be further explained in the next paragraph. The locations of the mode shapes are the frequencies at which the rotor will start to resonate, i.e. the critical frequencies of the rotor. These can be excited by the unbalance in the rotor. For example, if the rotor rotates at 74 Hz the 1st rigid mode shape will be excited, and the rotor's axis will start to form a cylindrical shape as explained with Figure 2-12 in section 2.3.2.

Table 3-8: Mode shape results

Mode shape	Analytical calculations	Simulation
Mass (m)	4.48 kg	4.5 kg
Mass moment of inertia (J)	0.0337 kg·m ²	0.0324 kg·m ²
1 st rigid mode ($k=500$ N/mm)	73.72 Hz	74.67 Hz
2 nd rigid mode ($k=500$ N/mm)	87.18 Hz	87.79 Hz
1 st bending mode ($k=\infty$ N/mm)	849.21 Hz	1028.79 Hz
1 st bending mode ($k=500$ N/mm)	Not calculated	1126.55 Hz

The simulated and calculated values of the 1st and 2nd rigid modes proved to be close to each other. A bearing stiffness of infinity is not achievable by AMBs. It is more the stiffness of mechanical bearings. However, the calculations were done at $k=\infty$ N/mm for simplicity as a limited stiffness moves the effective position where the rotor is effectively rigidly supported to the inside of the AMBs with an unknown amount. This is called the node points that the rotor form as it vibrated and will be further explained with Figure 3-28 later in this section. It was, however, seen that a limited bearing stiffness will always increase the 1st bending mode. This is not a problem since this means that the maximum operating frequency (500 Hz) will be further away from the 1st bending mode than expected.

The difference between the calculated and simulated values for the first bending frequency at a stiffness of 500 N/mm can be attributed to approximations made during the calculations. The calculated frequency of 849.21 Hz is considered to be the lowest value that the 1st bending frequency

can be located. This is still considerably higher than the operating frequency of the rotor. According to the literature, the rotor's operating frequency should be at least half of the 1st bending mode shape's frequency, for it to be considered a rigid rotor [28]. This will ensure that the 1st bending mode shape is not excited by the rotor's unbalance.

A critical speed map was drawn by using simulation. This is to show how the critical frequencies, at which the mode shapes occur change due to a change in the stiffness of the bearings, with the rotor rotation at 0 r/min. This is shown in Figure 3-27. If the rotor starts to rotate the gyroscopic effects of the rotor will start to change the critical frequencies, this is because it effectively changes the stiffness of the bearings. This is where the dynamic stiffness of AMBs comes from.

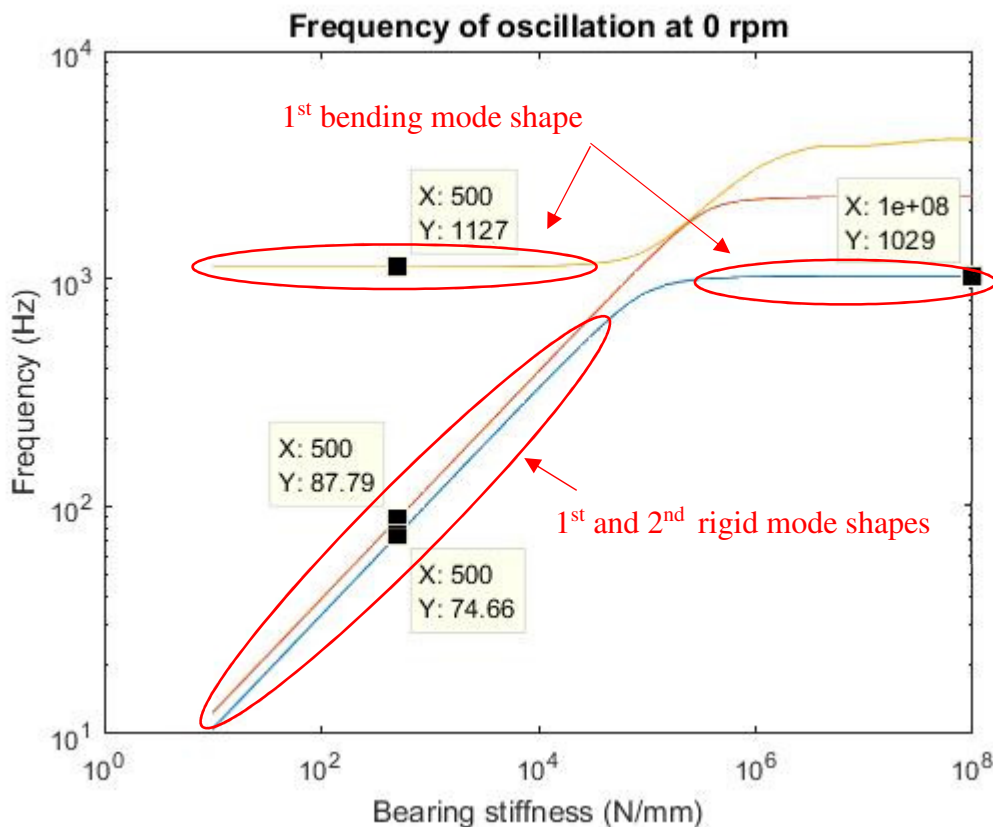


Figure 3-27: Critical speed map of the rotor's critical frequencies

Figure 3-27 shows the change of the frequency where the critical frequencies are located at, due to a change in bearing stiffness. It can be seen that the 1st and 2nd rigid mode shape's frequencies can be changed by changing the stiffness of the AMBs, within limits, of course. While the 1st bending mode's location can't be changed. While doing the simulation, new insight was gained into the mode shapes, as explained in Figure 2-11 of section 2.3.2. Figure 2-11 is shown again below in Figure 3-28.

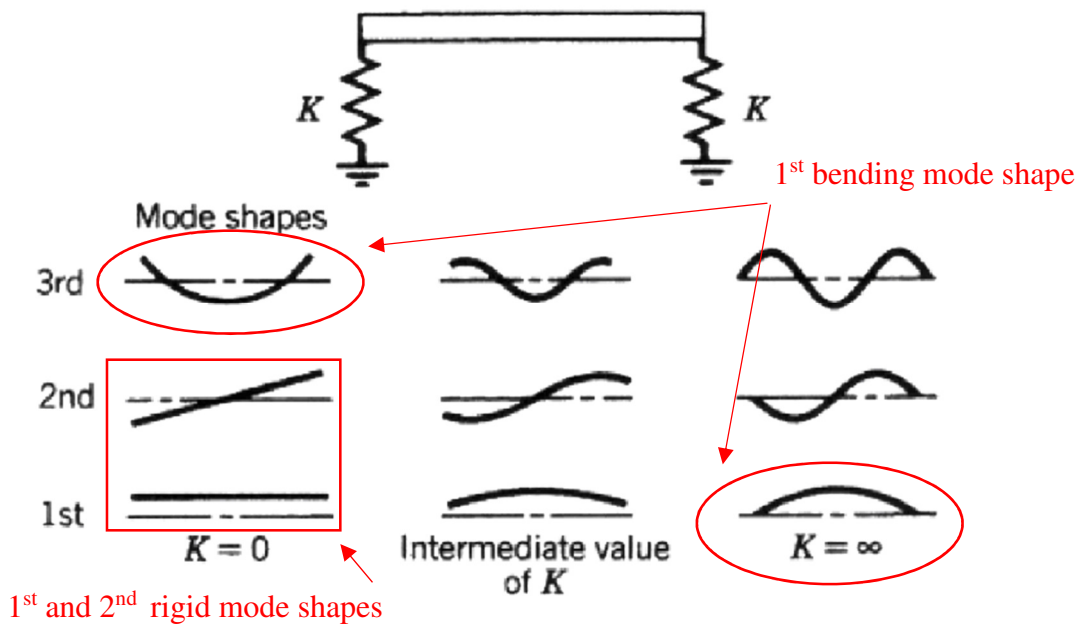


Figure 3-28: Mode shapes

The locations of the different mode shapes in Figure 3-28 is shown in Figure 3-27. It can be seen that, if the bearing stiffness increases from a low bearing stiffness $k \approx 0$ to a very high bearing stiffness $k = \infty$, the 1st rigid mode shape becomes the 1st bending mode shape. This is because if the bearing stiffness is low, the rotor is allowed to move up and down at the bearing locations allowing it to move rigidly. However, if the bearing stiffness is made very high, the rotor cannot move up and down at the bearing locations, meaning it cannot form a rigid mode shape.

Consider the nodes of the mode shape. It is the point where the mode shape crosses the centreline in Figure 3-28. It can be seen that for a low bearing stiffness the nodes of the 1st bending mode will be located inside the bearing locations not at the bearing locations, as it is for a very high bearing stiffness. These node positions are the positions where the rotor is effectively suspended with a rigid bearing ($k \approx \infty$) when calculating the 1st bending frequency as explained in section 2.3.2.2. From the calculations, it was seen that the critical frequency would be lower for a rotor which bends more under its own weight. From this, it can be concluded that if the location of the AMB is moved outward on the rotor, the 1st bending frequency will be lowered. It also explains why the 1st bending frequency of a rigidly suspended rotor is lower than if it is suspended by bearing with a low stiffness.

From the rotor dynamic analysis, it was seen that the motors maximum operating speed is lower than half of the 1st bending critical frequency of the rotor. The 1st bending critical frequency of the rotor is also its natural frequency. This is important as its location cannot be changed. The 1st and 2nd rigid critical frequencies can, however, be changed to ensure that they are not at the operating speed of the rotor.

3.12 SUMMARY

The electromagnets of the AMB were designed to be able to produce a force of 150 N. This is the maximum force that is expected to be produced by the unbalance of the rotor after it has been balanced to a balancing standard of G1. It was seen that the cross-coupling of the force in the x and y -direction of a 4-pole electromagnet would be a little higher than the cross-coupling in an 8-pole electromagnet. It was however decided that a 4-pole electromagnet will still be used as it has a much simpler construction and a lot less coils need to be wound. It was also seen that the 4-pole EM has lower iron loss than the 8-pole EM, this can be attributed to the fact that the 4-pole can be designed to operate at a much lower magnetic flux density than an 8-pole EM can, for the same outer diameter and axial length.

It was seen that the losses in the rotor produce by the bearing of the TWINS can be reduced from 75W to 8W by replacing its ball bearings with AMBs. This made it feasible to do the retrofit.

The retrofit will be possible from a mechanical point of view. It was seen that the journal of the AMB will be able to be shrink-fitted to the current rotor of the TWINS. It was also seen from a rotor dynamic point of view that the AMBs will be able to suspend the modified rotor. This is because the rotor's natural frequency is double than the maximum operating frequency of the motor. Meaning that it will not be excited, causing the rotor to become unstable.

CHAPTER 4: IMPLEMENTATION AND TESTING

The designed AMBs needs to replace the ball bearings that are currently installed in the TWINS. This chapter will explain the modifications that were made to one of the PMSMs to equip it with AMBs. It will explain the Simulink model that was used to perform tests on the designed AMBs. The parameters that were tested include the force that can be produced by the electromagnets, the stiffness that is produced by the AMBs, the coil's resistance, and its inductance. It will also explain a method that can be used to determine the magnetic centre point of the AMBs.

The next section will explain the modifications that were made to the TWINS.

4.1 SYSTEM ASSEMBLY

Figure 4-1 shows a section view of the current assembly of one of the TWINS. The complete set of detailed drawings of the TWINS before the modification can be found in Appendix B.7. All machining of new components was done by the Instrument makers on the North-West University Potchefstroom campus.

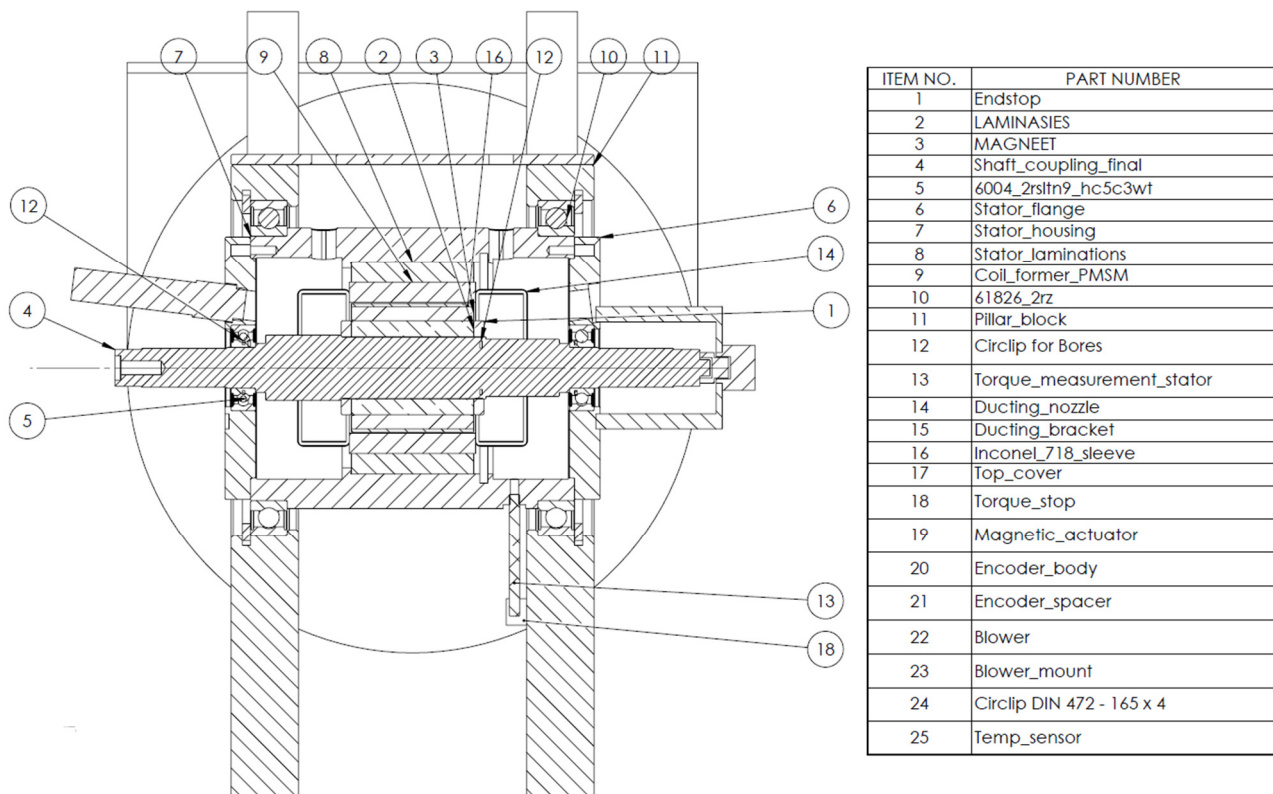


Figure 4-1: Machine drawing of current assembly [2]

With the modifications that were made to the TWINS, components that were removed and not reused includes the components in Figure 4-1 with part numbers 5,6,10-15,18-24. The parts that will be reused include the rotor (part numbers: 1, 2, 3, 16), the Stator_housing and everything that is permanently attached to it (part numbers:7, 8, 9), the Top_cover (17) plate as this part houses the cables that come out of the motor, and the Pillar_blocks (11) as they can be modified to be used on the new assembly. The big ball bearing (10) was intended to make it possible to measure the torque that is produced by the motor; this did not work and will not be implemented into the new design.

The modifications that were made to the rotor will be explained first. As stated in section 3.10, the journals of the AMBs, as well as endstops, need to be shrink-fitted onto the current rotor of the TWINS. Figure 4-2 shows a drawing of the TWINS's rotor, with the modifications that were made to it.

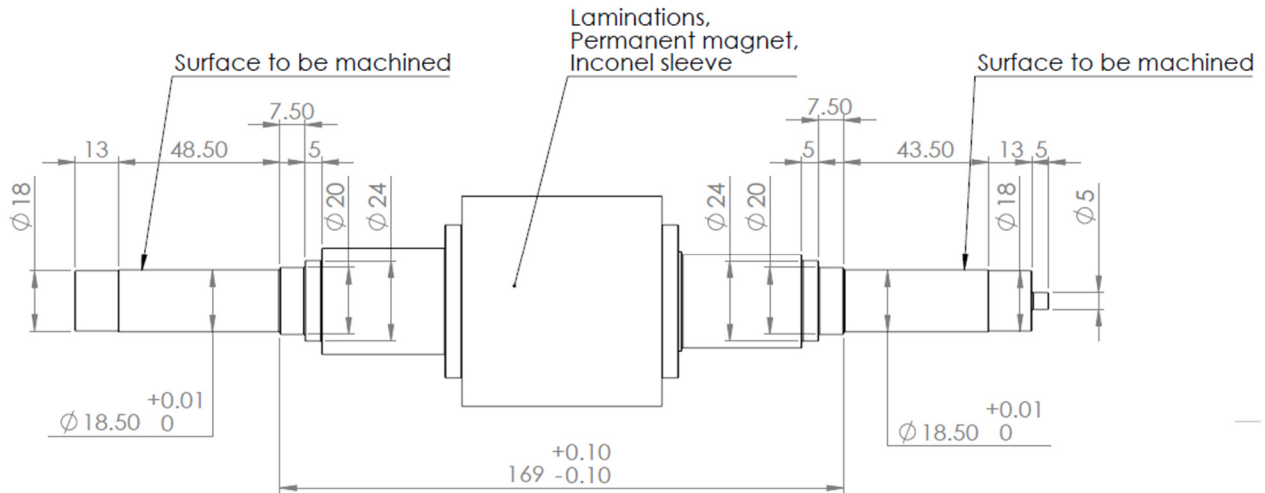


Figure 4-2: Rotor machining

The surfaces that are indicated as “*Surface to be machined*” were machined down from 19 mm to 18.5 mm with a tolerance of +0.01 and -0 mm. This is to ensure that the surface is smooth as the laminations and endstops need to be fitted onto these surfaces. As it would be impossible to shrink-fit the laminations and the endstops to the shaft individually, they were assembled beforehand and fitted to the shaft as a whole. Two endstops and a sleeve were manufactured and assembled as shown in Figure 4-3. This was done to keep the 57 laminations in a manageable stack and to keep the laminations from bending and buckling.

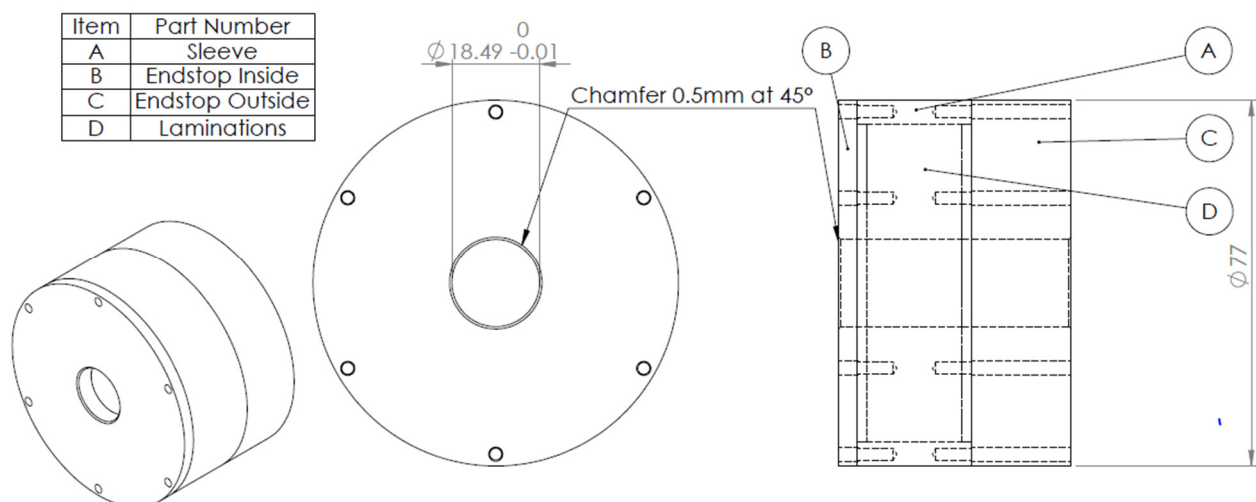


Figure 4-3: Laminations press fit jig

The laminations and endstops were manufactured with an inner diameter smaller than what it should be before it is shrink-fitted to the rotor, and machined to its final size after assembly. This is to ensure that their inner diameters are exactly the same size and line up perfectly. The laminations and endstops

were also manufactured with a greater diameter than its final desired size. This was done to ensure that the final outer diameter is the same for all three components. As well as to enable one to bolt it to the Sleeve to keep the parts together. The sleeve was fully stacked with laminations and one extra lamination to ensure that the lamination stack can be properly compressed by the endstops. This assembly was machined to an inner diameter of 18.49 mm with a tolerance of +0 and -0.01 mm. The assembly can now be added to the machined rotor and also machined to its final outer diameter to produce the rotor as shown in Figure 4-4.

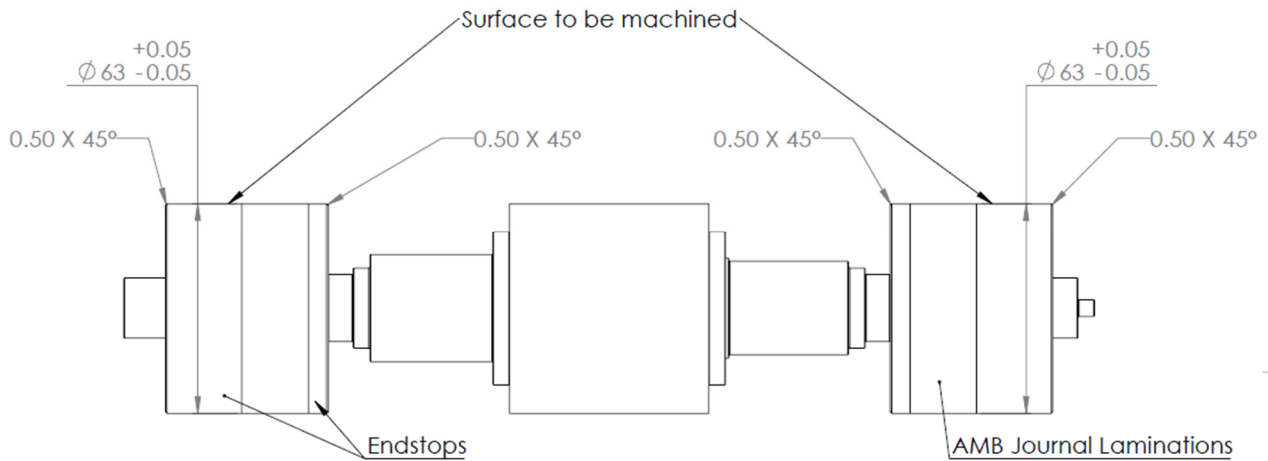


Figure 4-4: Rotor final

The outer endstops were made thicker to allow for rotor position measurements to be taken on them. As well as to create a surface to drop the rotor on backup pins in case the AMBs fail, while the motor is in operation. The backup pins will be shown later in this section. This concludes the modifications that were made to the rotor. The modifications that were made to the stator will be explained next.

The stator of the TWINS was used as is, and is shown in Figure 4-5.

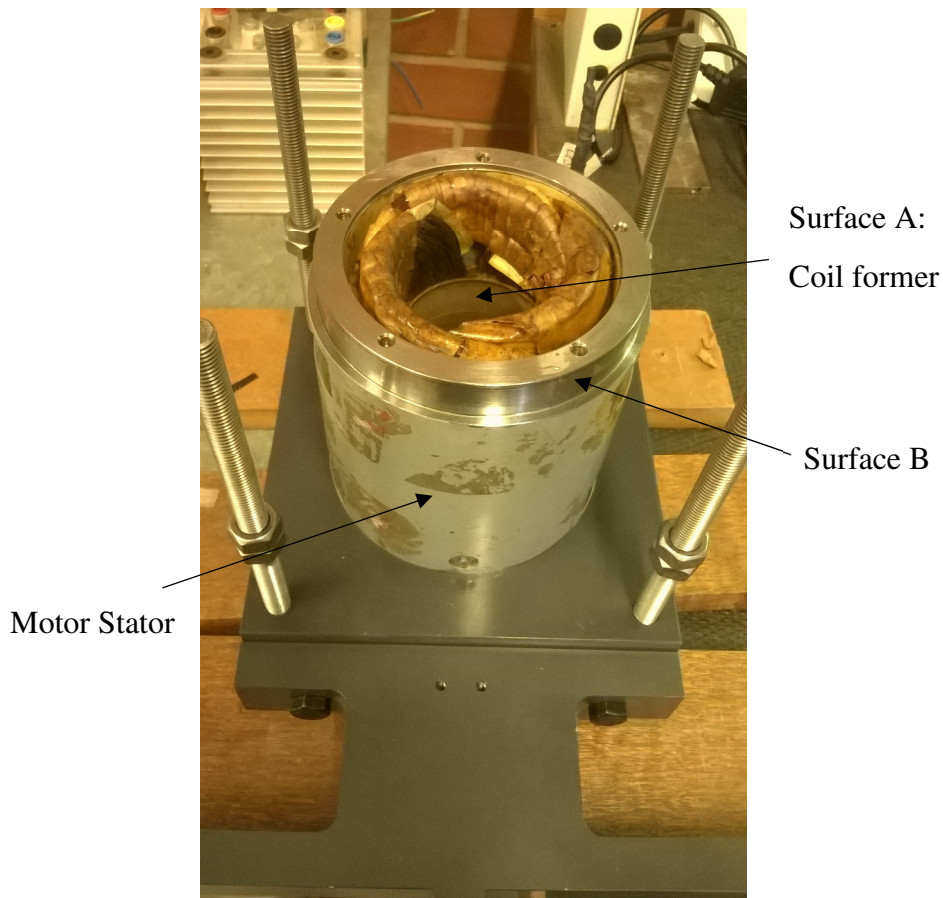


Figure 4-5: Motor stator

Surface B was used to fit the AMB housing to the stator. This was done due to the fact that Surface B was made concentric with the coil former inside the stator (Surface A). This is to ensure that the laminations inner diameter is concentric with that of the coil former. This will happen as Surface C and D of the AMB housing shown in Figure 4-6 were made to be concentric with each other. This means that if Surface C of the AMB housing fits over Surface B of the motor stator housing with a good sliding fit, that the inner diameter of the laminations and coil former will be concentric with respect to each other to a great extent.

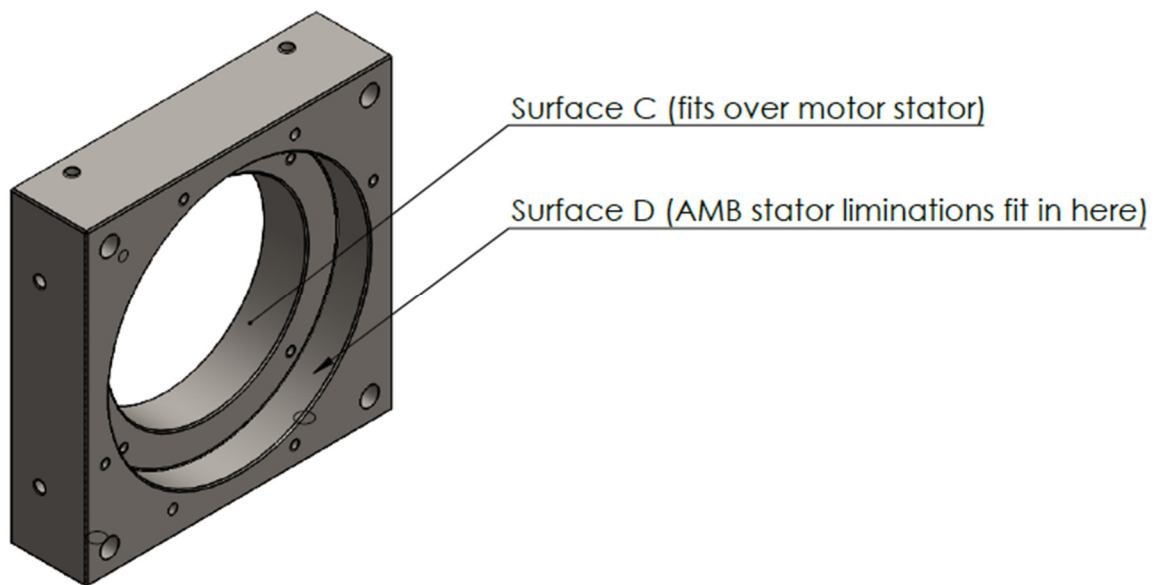


Figure 4-6: AMB stator

Surfaces C and D were made concentric by clamping the part in a lathe and machining both surfaces without reclamping the part. This was important to ensure that the pole tips of the AMB's electromagnet line up with the motor stators coil former. If they don't line up the rotor may make contact with the coil former or one of the electromagnets, as the air-gap is only 0.5 mm, between the three stator parts and the rotor. The assembly of the HS-PMSM with the AMBs is shown in Figure 4-7.

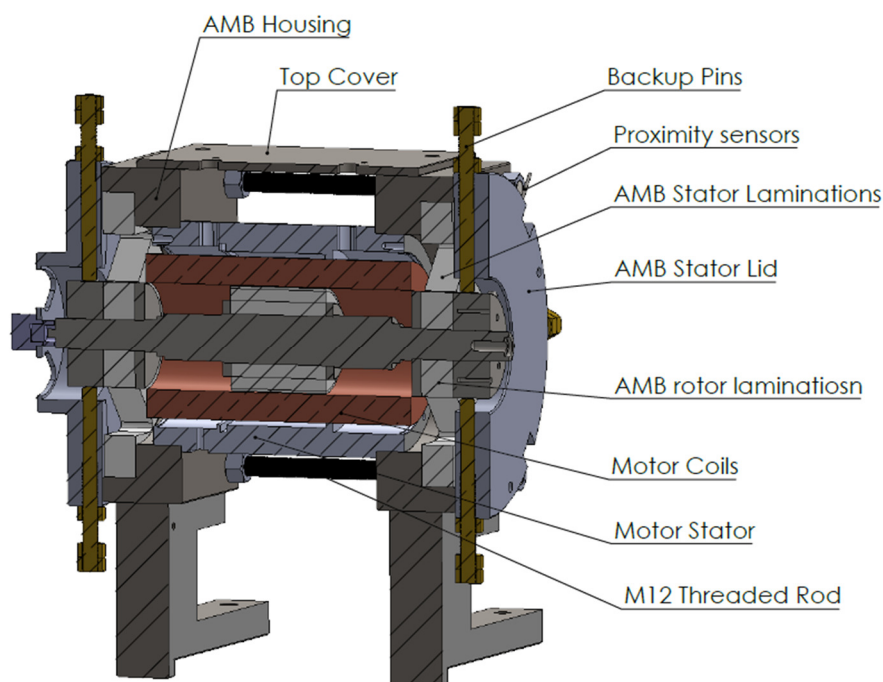


Figure 4-7: Motor assembly with AMBs

The two AMB housings were secured to the motor stator by using four M12 threaded rods as shown in Figure 4-7. The backup bearings and positioning sensors were put in place by bolting lids on the AMB stator housing. Figure 4-8 shows a 3D drawing of the Lid that is holding the rotor positioning sensors and backup pins in place.

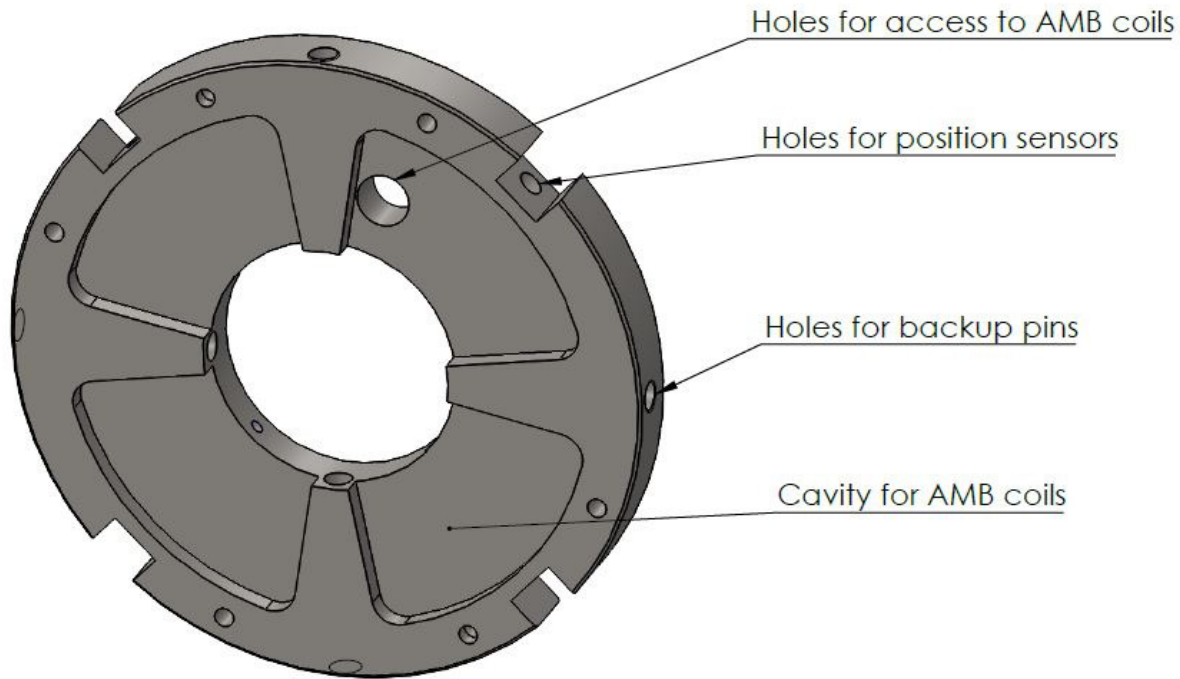


Figure 4-8: AMB lid1

The cavities that were machined out of the lid is to make space for the AMB's coils. This is to save axial space as there wasn't much axial space on the rotor to add the lamination/endstop assembly. It also enables the backup pins to be axially in line with the AMB's end coils. All the detailed drawings of the new motor assembly can be found in Appendix B.2.

Balancing holes were tapped into the two outer endstops of the rotor, as shown in Figure 4-9 a). Grub screws can be inserted into these holes to add mass to one side of the rotor to balance it. The balancing of the rotor was not done. This was left for future work that needs to be done. Figure 4-9 b) shows the dimensions of these holes. This will be useful information if the balancing of the rotor is done.

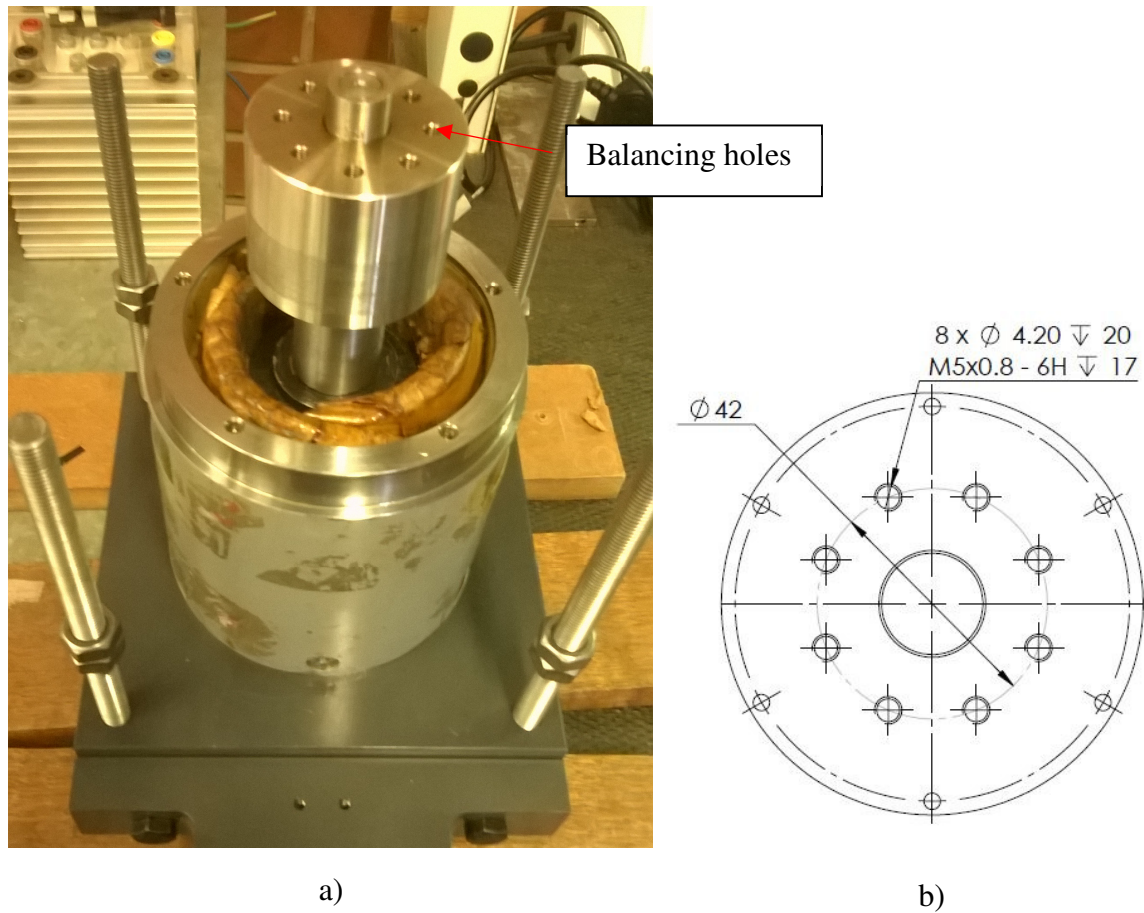


Figure 4-9: Balancing holes

Lastly, laminations (used for the electromagnet) were cut to shape by using a wire cutter. To do this, a stack 50 mm thick were made from rectangular lamination plates of 244x180 mm. This stack was bolted and welded together to ensure that all the laminations are electrically connected. This is necessary as, during the wire cutting process, electrical energy is converted into thermal energy during spark discharges used to erode the material away [29]. While the lamination stack was being cut, the lamination stack flared open and broke the wire. When the cutting process was restarted, the machine didn't start on the exact same place that it stopped. This caused a cutting error in one of the poles. Figure 4-10 a) shows the error that was made, the error is not to scale; only the shape of the error is illustrated. Due to time constraints, it was decided that the laminations will still be used as the error was not that significant. The effect that the error has on the force produced will be investigated in section 4.4.

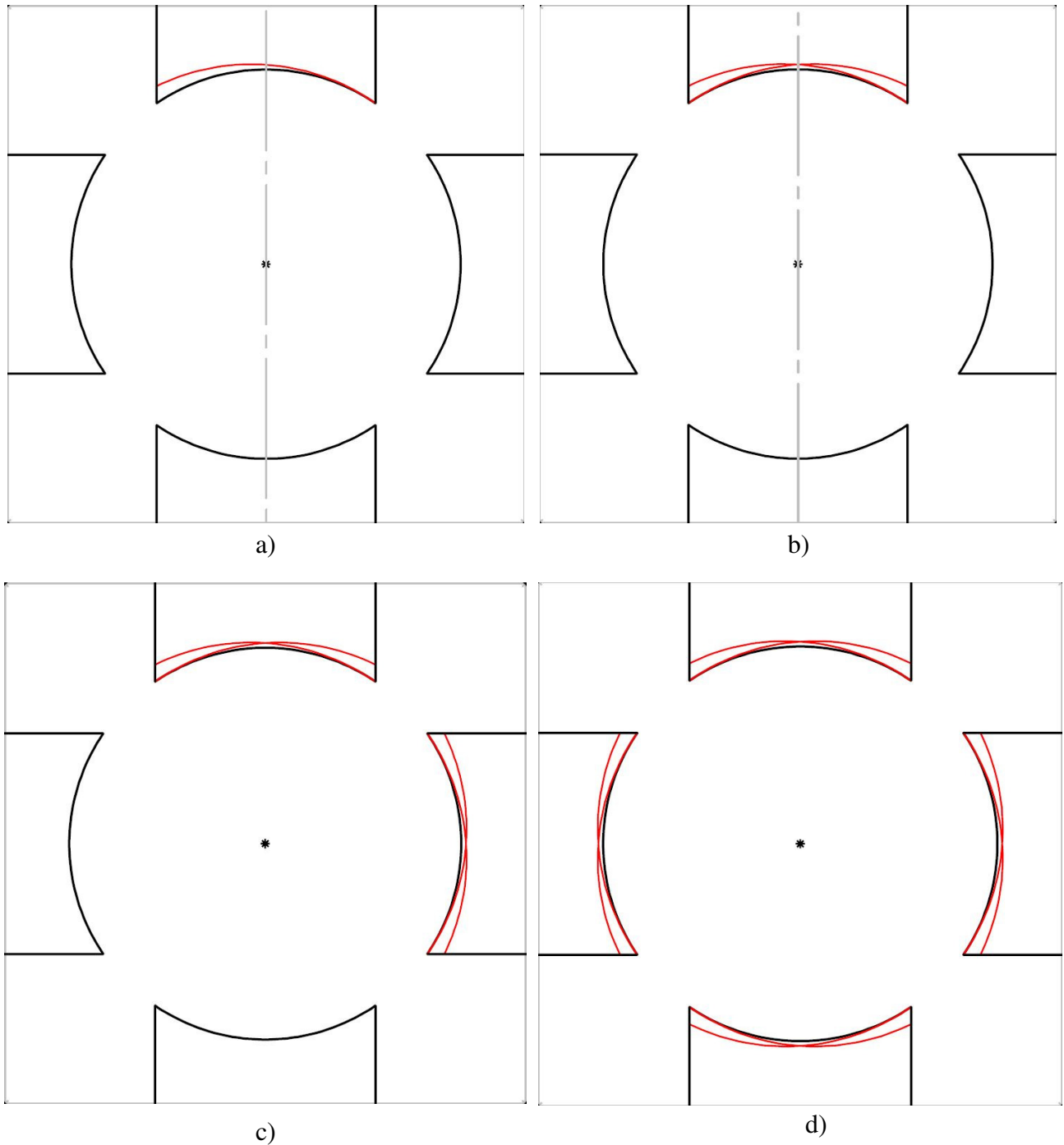


Figure 4-10: Cut error in laminations

An attempt was made to minimize the effect of the error. This was done by creating pairs of laminations that were flipped around the centreline, as shown in Figure 4-10 a) to Figure 4-10 b). These pairs of laminations were rotated by 90° every time they were stacked onto each other. This means that one complete pole, will consist of two wrongly cut laminations followed by 6 correctly cut laminations. This pattern will continue until a stack of 20 mm is produced. The effect of the manufacturing error will be investigated later in this chapter.

The next section will explain the Simulink model that was used to control the levitation of the rotor.

4.2 SIMULINK MODEL

A PID control strategy was implemented with dSPACE® in order to levitate the rotor. The integrator was added to reduce the steady-state error to zero. This was desirable as force measurements needed to be taken at accurately known positions while the rotor is at a standstill. However, the addition of the integrator will change the dynamic behaviour of the controller and needs to be investigated before levitating the rotor with it at high-speeds. Figure 4-11 shows the Simulink control model of both the radial AMBs. The rotors radial position was captured by four eddy current displacement sensors, two on each AMB, one for the x and one for the y -position of the rotor. Two Micro-epsilon DT3005-U1-A-C1 sensors were bought while two Micro-epsilon DT3701-U1-A-C3 sensors from previous projects were used.

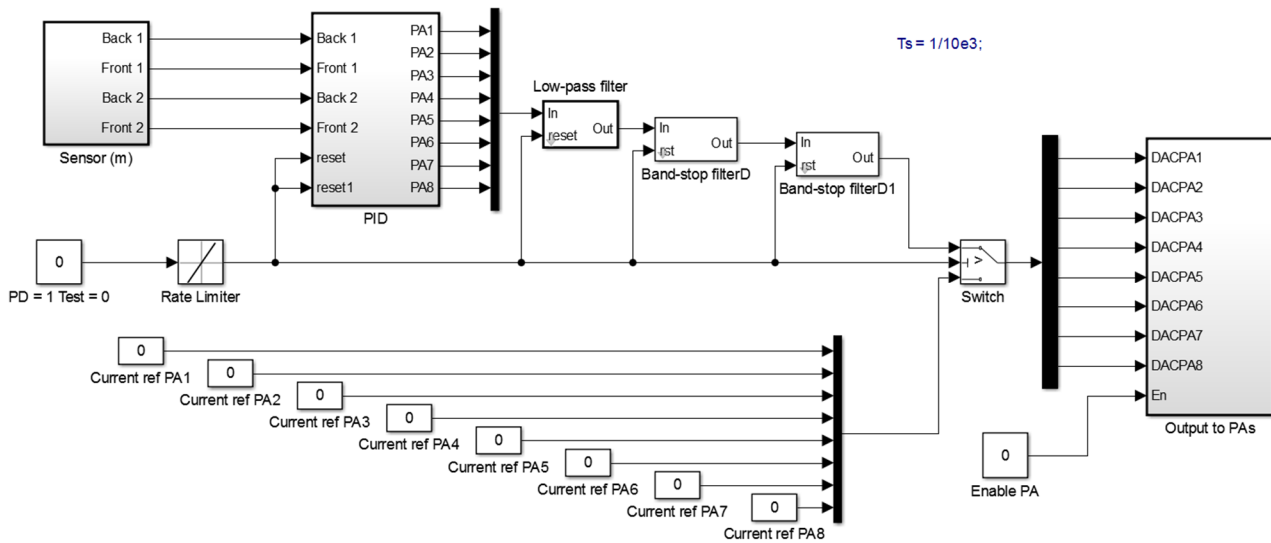


Figure 4-11: Simulink® model of two radial AMBs

The dSPACE® discretizes a voltage between 0 and 10V to a value between 0 and 1. This value has to be converted to represent a value in meters, as shown in Figure 4-12. The sampling frequency of the dSPACE® was chosen as 10 kHz. According to the Nyquist theorem: the sampling must be at least twice that of the highest frequency sampled. To achieve this, the output of the sensors was filtered by an anti-aliasing filter with a cutoff frequency of 5 kHz, ensuring that aliasing does not happen.

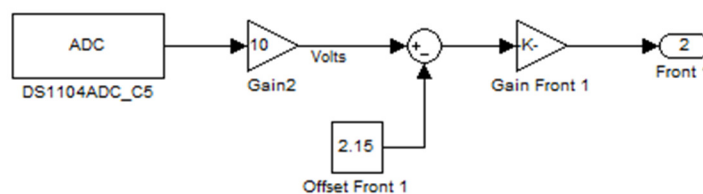


Figure 4-12: Simulink® Sensor (m)

Figure 4-12 shows the conversion of one of the sensor's output voltage to a value in metres. **Gain2** was inserted to convert the output of the ADC's back to a voltage out of 10. Whereas **Gain Front 1** describes the sensitivity of the sensor which was determined to be 126.16 $\mu\text{m}/\text{V}$. This was done by firstly approximating the sensitivity as the maximum travelling distance of the rotor (1mm) divided by the voltage difference at the rotors maximum and minimum displacement. This yielded a

sensitivity of $143.37 \mu\text{m}/\text{V}$, which is not an accurate value as the sensor is not linear at its extreme measuring distances due to saturation effects.

The true sensitivity was calculated by varying the position of the rotor in increments of $50 \mu\text{m}$ with the approximated sensitivity and using a dial gauge to measure the actual displacement of the rotor as shown in Figure 4-13.

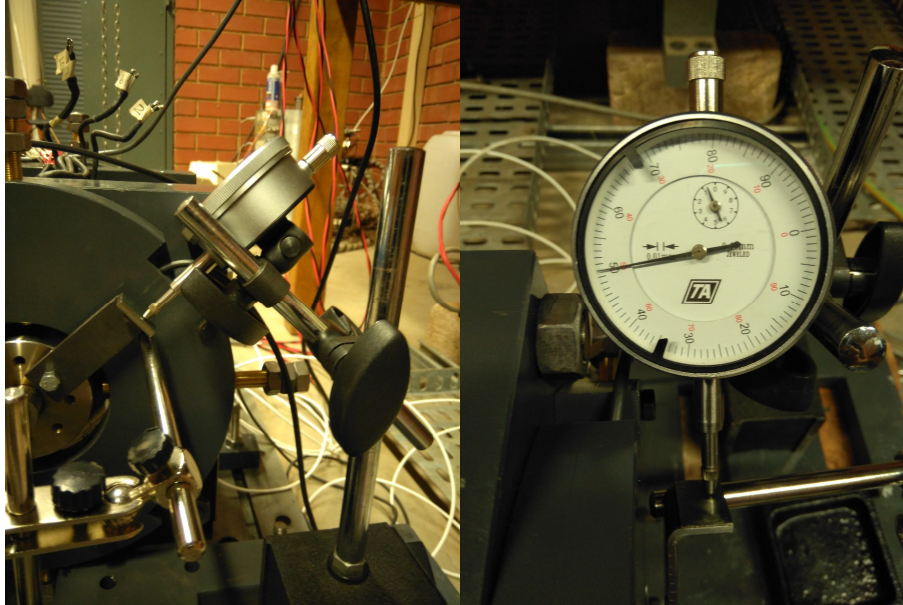


Figure 4-13: Sensor calibration

The measurements that were taken of the sensor calibration is shown in Table 4-1.

Table 4-1: Sensor calibration

Output Voltage of sensors (V)	Change in voltage	Dial gauge measurement (μm)	Change in μm
4.86		675	
5.21	0.35	630	45
5.56	0.35	589	41
5.91	0.35	545	44
6.25	0.34	500	45
6.6	0.35	455	45
6.95	0.35	410	45
7.3	0.35	369	41
7.65	0.35	323	46
Average change in voltage	: 0.34875	Average change in displacement (μm)	: 44

The sensitivity can now be calculate by using the average change in position and the average change in voltage as:

$$\text{Sensitivity} = \frac{44 \mu\text{m}}{0.34875 \text{ V}} \quad (4.1)$$

$$\Delta x = 126.164875 \times 10^{-6} \cdot \Delta V$$

This sensitivity was tested after implementation and produced a good correlation between the change in position implemented in the dSPACE® and the actual change measured by the dial gauge. This only ensures that the desired change in position is correct. However, the actual centre point of the rotor is not necessarily at the (0.5,0.5) mm coordinates as shown by the (**Sensor (m)**) dSPACE® block. This is due to the fact that the output voltage of the sensor is not 0 V if the rotor touches the pole closest to the sensor. This was rectified by subtracting an offset voltage from the voltage measured by the sensors. This offset value was determined by using the electromagnet to pull the rotor to the AMB pole, and subtracting that voltage from the measured voltage. The centre point was determined by finding the magnetic centre, as explained in Section 4.3. This was done as it was still unclear where the true centre point lies.

The **PID** controller block produces a reference signal that is sent to the power amplifiers controlling the rotor to levitate at the desired position. This signal is filtered and scaled before it is sent to the power amplifiers.

The PID block is shown in Figure 4-14 below, note that the control for just one AMB is shown. At the top of the figure, a sine wave generator is shown. This was used to add a sinusoidal current to two opposing coils; this was used to find the magnetic centre and will be explained in Section 4.3. Gain **id_Back1** and **id_Front 1** controls the amplitude of the sine wave and is used to switch it off. Gain **fd1** and **f2w** are used to set the frequency of the sine wave.

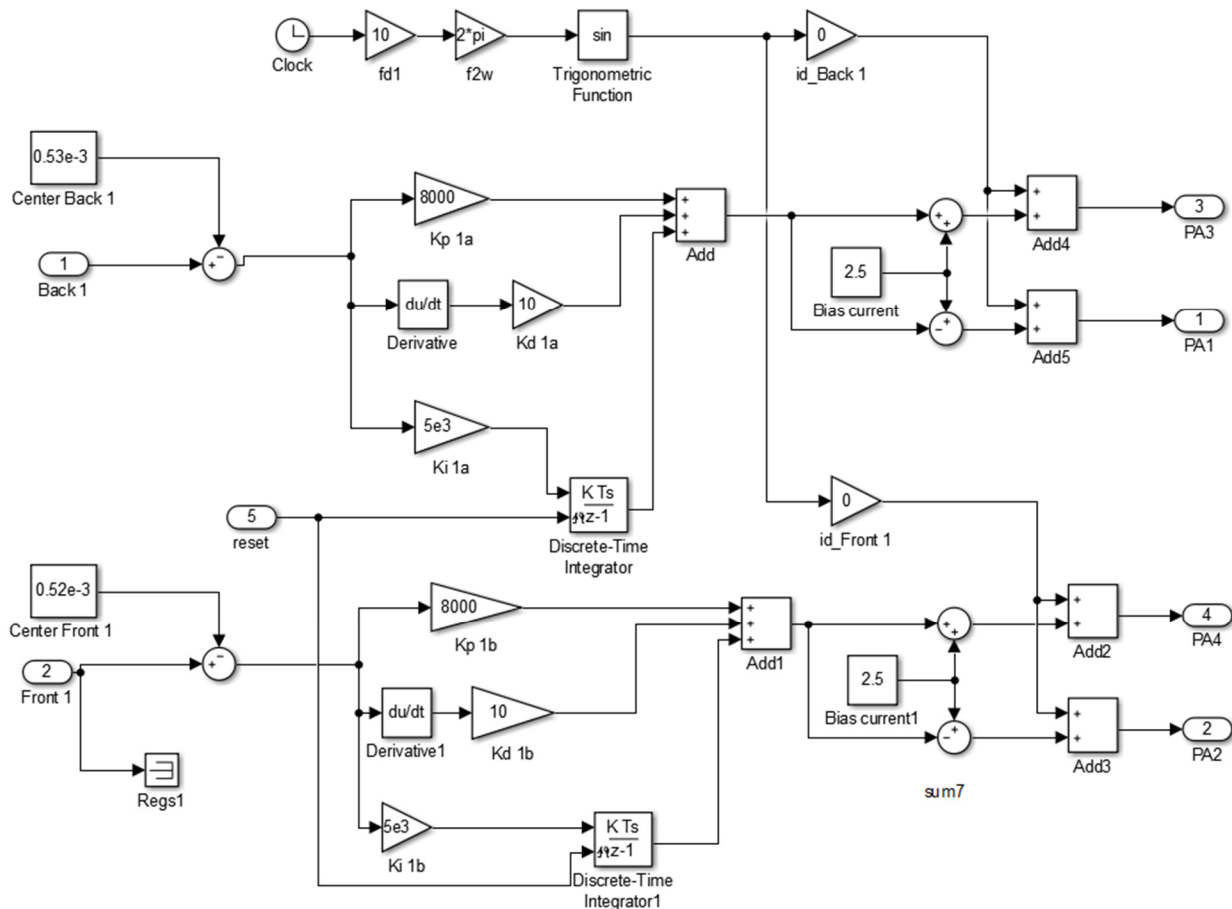


Figure 4-14: PID control for one radial AMB

Centre Back 1 and **Centre Front 1** are used to set the coordinates that the rotor will levitate at. Currently, these values are set to the magnetic centre point of the electromagnet. Input **Back 1** and **Front 1** are the sensors measurements as converted by the **Sensor (m)** block. An error signal that is sent to the PID controller is obtained by subtracting the reference position from the measured position. Gains K_p , K_d and K_i are the constants for the proportional, differential and integral parts of the PID controller respectively. The integrator was inserted to reduce the steady-state error to zero, as it was desirable to set the rotor's position accurately. This was mainly used to determine the sensitivity of the sensors. If the integrator was not inserted the steady-state error of the PD controller would have caused the measured sensitivity to be inaccurate. It should, however, be noted that the integrator will change the dynamic behaviour of the rotor. Thus the influence of the integrator should firstly be analysed before it is used with the motor in operation. The integrator was only used in static tests.

In Figure 4-11, the low-pass filters are used to filter out high frequency noise as the sensor operates in a noisy environment. Two band-stop filters were used to filter out frequencies that were causing an audible vibration when the rotor was levitated. It is suspected that these were the critical frequencies of the rotor that were being excited. These frequencies were determined by using a phone app called *Advanced Spectrum Analyzer*.

In Figure 4-11 a switch was implemented to switch between controlling the power amplifiers' reference current manually or with the PID controller. This switch is referred to as the **PD=1 Test =0** constant block. The ability to control the PAs' current manually were used for the purposes of pulling the rotor to the pole tip that is in line with the sensor. This was used to install the sensors at a position, that ensures the rotor could not damage the sensors. The ability to control the PAs' current manually were also used to calibrate their current outputs. Another switch was inserted to disable the output current of the power amplifier, this is the **Enable PA** block.

The current reference signal was scaled by the **Output to PAs** Simulink block and is shown in Figure 4-15.

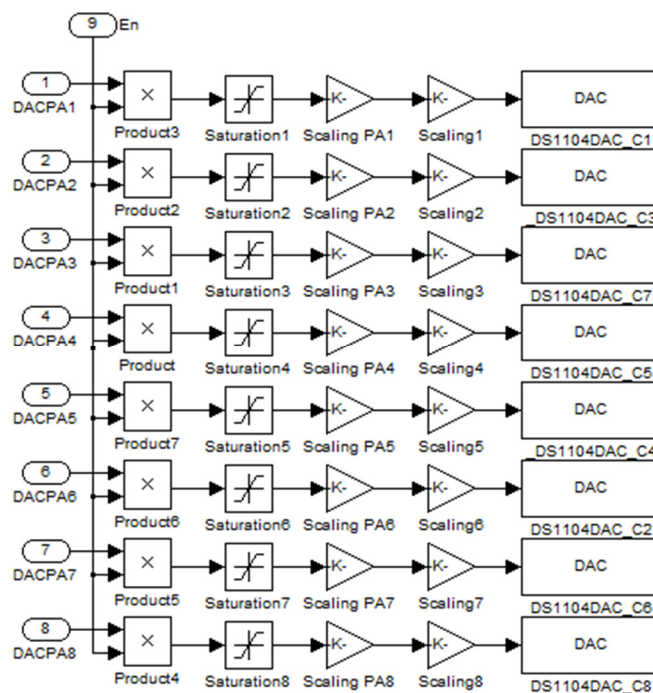


Figure 4-15: Scaling of PAs reference signal

Saturation blocks 1-8 are used to limit the output current of the PAs to 6 A. This is the maximum continuous current rating of the PAs and is above the maximum designed current needed for the electromagnets. The gains **Scaling PA1-8** scales, the current reference signal to match the sensitivity of the PAs. These gains differ slightly for the different power amplifiers, but is in the range of 0.75 V/A. Lastly, the DACs only accept values between -1 and 1 which represents an output voltage of between -10 V to 10 V. This means that the current reference signal had to be attenuated by 10. This is done by the gain blocks **Scaling1-8**. These outputs are connected to the DACs, which produces analogue reference signals sent to each PA.

4.3 MAGNETIC CENTRE POINT

As stated in section 4.2, the rotor is intended to levitate around its magnetic centre point. The magnetic centre point was used as it is difficult to determine the actual centre point of the rotor accurately. This is because it is unclear how far the sensors are from the actual centre point. The magnetic centre point is the point where the force produced by the same amount of current in coils on opposite sides of the rotor will cancel out.

Consider Figure 4-16, if the rotor is in its centre position, and all the coils are biased with 5 A, the resultant force on the rotor will be zero as shown in the top figure. This is because the magnetic flux density in all the poles will be the same as the reluctance of all four paths, as well as all four mmf's are the same. However, if the rotor is offset from the centre, as shown in the bottom figure, the force on the rotor will be towards the pole on the left side. Regardless of the fact that the mmf produced by all the coils are the same. This is due to the smaller reluctance between the left pole and the rotor vs the other three poles.

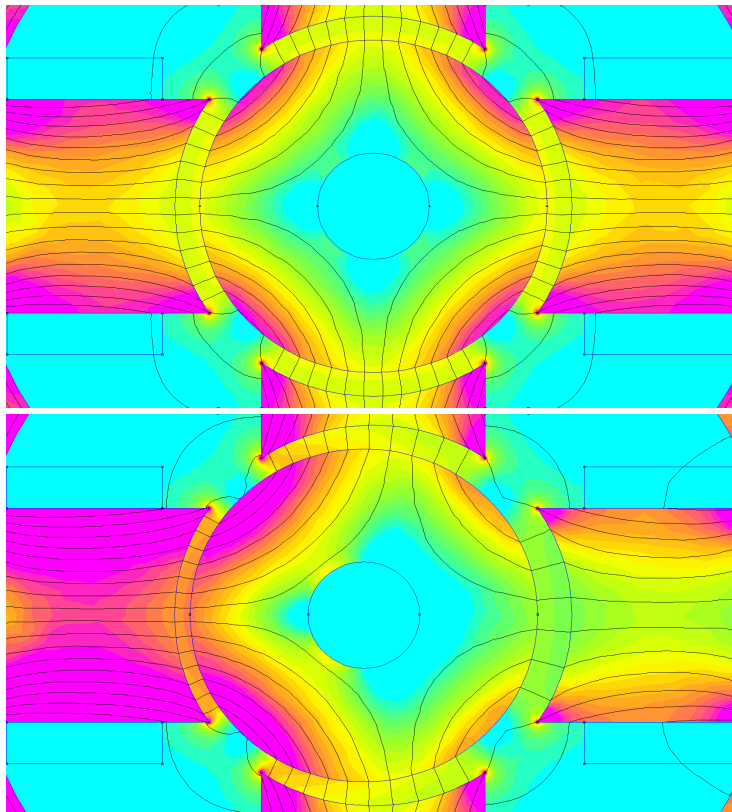


Figure 4-16: Magnetic flux vs rotor position

The magnetic centre point was determined as follow; the rotor was levitated with the PID controller at a point believed to be the centre. Two in-phase sinusoidal currents with an amplitude of 1.5 A and a frequency of 10 Hz were superimposed on the control currents of the coils in the x -axis. If the rotor is in the middle of the poles in the x -axis the amplitude of the two forces will be the same as shown in Figure 4-17, as the reluctance of the two magnetic paths are the same.

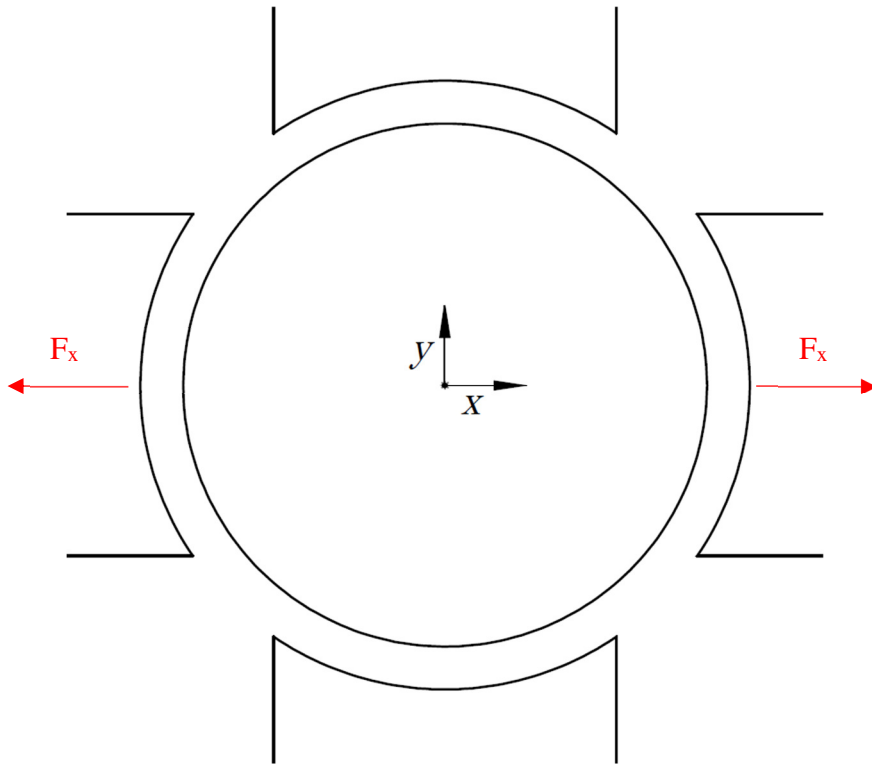


Figure 4-17: Magnetic centre

However, if the rotor is not located in the centre of the x -axis as shown in Figure 4-18 the forces caused by the superimposed sinusoidal currents will not cancel out. This will cause the rotor to start oscillating in the x -direction as the magnetic flux will be higher at the left pole tip, hence the force on the rotor will be to the left. This is because the reluctance in the left coils magnetic path is less than that of the right coil's magnetic path although the mmf produced by the two is the same. This is the same as with an electrical circuit if the resistance reduces the current will increase, although the voltage stays the same.

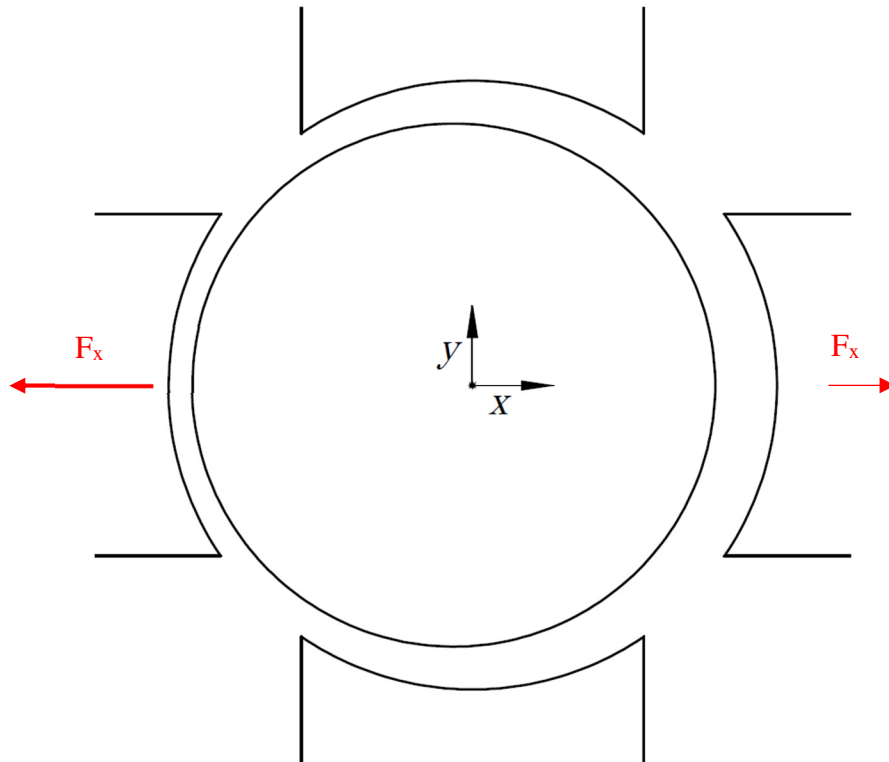


Figure 4-18: Offset from magnetic centre

Two methods can be used to determine the magnetic centre point of the rotor. Namely by using the position signal of the rotor and by using the current reference signal to the power amplifiers. These two methods will be explained in the next two paragraphs.

Method 1: While superimposing a sinusoidal current signal to the coils in the x -direction, the rotors reference position in the x -direction can be varied, until the oscillation of the rotor in the x -direction position signal is at its lowest. This same procedure can be followed to obtain the magnetic centre point in the y -direction.

Method 2: While superimposing a sinusoidal current on the coils in the y -direction, the rotors reference position in the x -direction can be varied, until the oscillation of the currents in the coils that control the x -direction is a minimum. This method will however, not work in the 8-pole configuration with a NSSN configuration as there is no magnetic flux that is shared between the x and y -direction. However, it will work with the 4-pole electromagnet that was implemented, due to the fact that the x and y -axis shares a magnetic path. With the 4-pole configuration, the change in magnetic flux in the x -direction will only be zero if the current in the y -direction is being controlled differentially i.e. if the current in the top coil is increased the current in the bottom coil has to be decreased by the same amount. However, if the current in the top and bottom coils are both increased at the same time, as is the case while determining the magnetic centre, the flux in the x -direction will also change. If the rotor's x -position is in the middle, the magnetic flux in the left and right coils will both increase by the same amount, resulting in a net force of zero on the rotor. However, if it is not in the centre, as shown in Figure 4-18, the magnetic flux in the left pole will increase more, as this magnetic path has less reluctance in it. This results in a net force on the rotor to the left. This net force will be counteracted by the control current in the left and right coils, resulting in an oscillation of the rotor in the x -direction, as well as an oscillation in the current of the left and right coils.

Method 2 was the preferred method as the current reference signals had less noise on it than the position signals. This made it easier to see when the oscillation was at its minimum.

Figure 4-19 and Figure 4-20 shows the current in the four coils. Firstly with the rotor away from its centre position, and secondly with the rotor at its centre position.

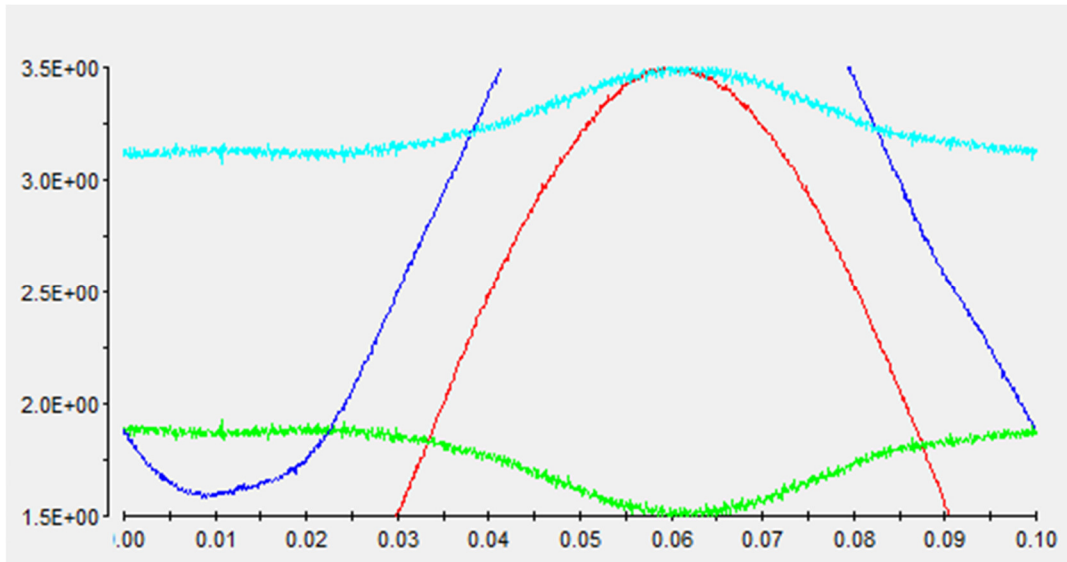


Figure 4-19: Current oscillation with rotor not at magnetic centre

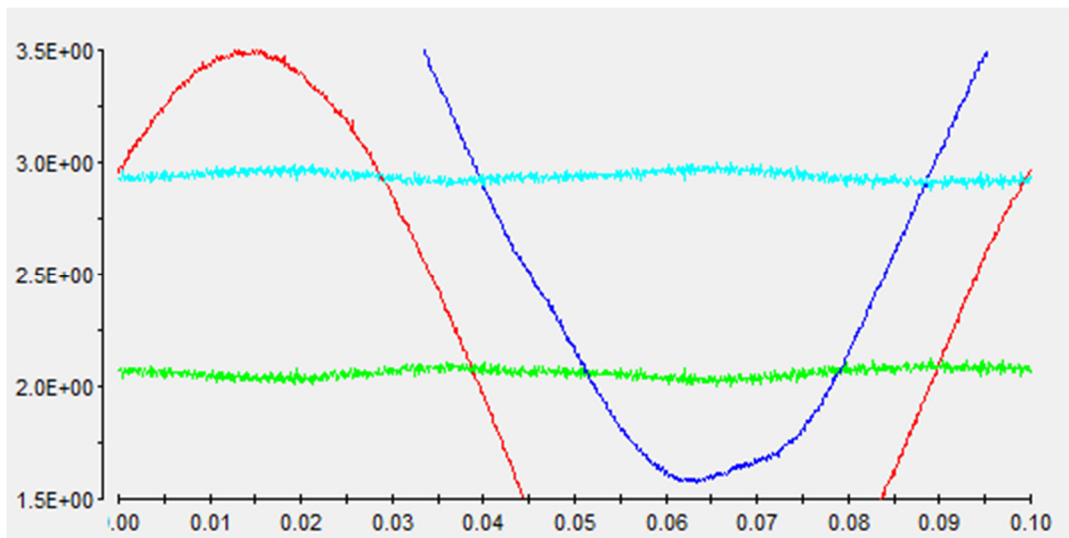


Figure 4-20: Current oscillation with rotor at magnetic centre

The dark blue and red lines represent the current in the top and bottom coils that were superimposed with a sinusoidal current while the light blue and green lines represent the current in the left and right coils of the perpendicular axis. Between the two figures, the reference position was moved from 590 μm to 520 μm in the x -direction. The reduction in the current oscillation can clearly be seen. The same procedure was followed for the other y -axis.

4.4 FORCE VERIFICATION

The force that the Electromagnets can produce was measured to verify that it can produce the force that it was designed for. The force was measured by levitating the rotor at a certain reference position and using a spring scale to apply a force on the rotor. The force was applied on a bolt that was screwed into one of the balancing holes, as shown in Figure 4-21. The force was applied at point C, and the AMB is applying a counterforce in line with point B.

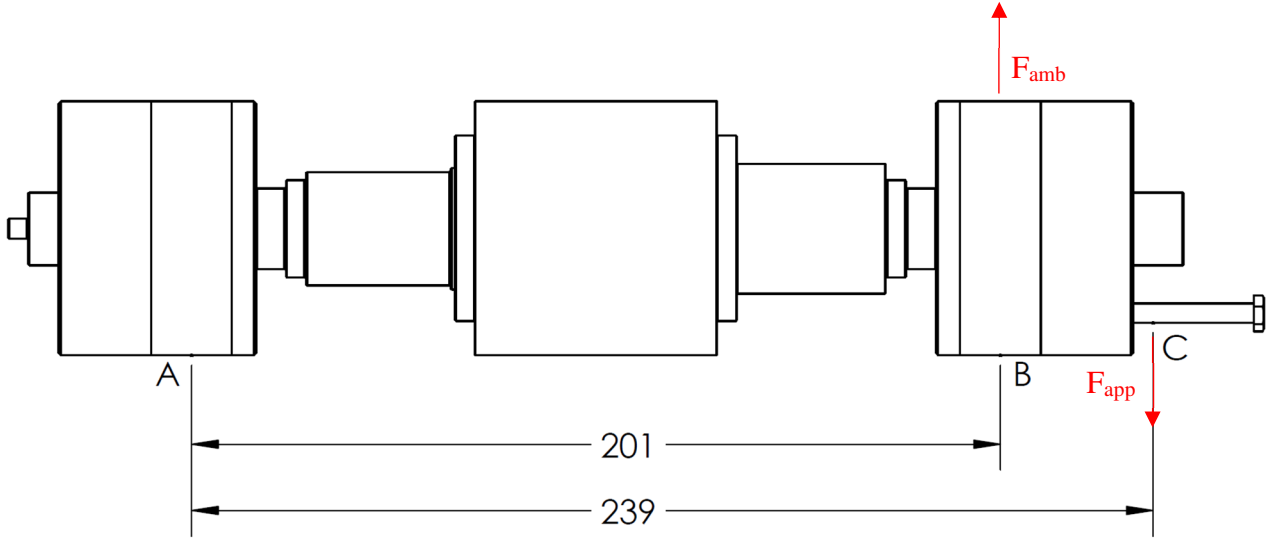


Figure 4-21: Force conversion

The force that is being measured by the spring scale (F_{app}) is not equal to the force that is being produced by the AMB (F_{amb}). For the rotor to be in static equilibrium, the torques produced by the forces need to cancel each other out. Point A was considered as a pivot point. This was done because the left AMB is holding the rotor at its centre point in the stator. This means that the torque produced by forces F_{app} and F_{amb} around point A need to be the same:

$$F_{app} \cdot 239 = F_{amb} \cdot 201 \quad (4.2)$$

Thus the force applied by the AMB can be calculated by using:

$$F_{amb} = 1.189 \cdot F_{app} \quad (4.3)$$

As can be seen, the conversion increases the measured results by 18.9%, which is a substantial amount. For the measurements, the force was applied until the current in the coil to which the rotor is being pulled is about 0 A and the current in the opposing coil is 5 A. Consider that the force was applied in the x -direction. For the first test, the rotor was suspended at its centre in the y -direction, and its x -position was varied to measure the achievable force at different air-gaps. The measurements are shown in Table 4-2, with the simulated forces. The negative displacement means that the rotor is away from the pole that is producing the force by the indicated amount i.e. the air-gap is increased. The applied force by the scale opposes the force produced by the AMB.

Table 4-2: AMB Force with y at 0

Displacement from centre(μm)	F_{amb} measured (N)	F simulated(N)
-200	117	87
-150	126	100
-100	138	115
-50	149	133
0	163	154
50	169	179
100	181	210
150	184	250
200	192	301

The variations between the measured and simulated force are due to the pole that was not cut properly as well as the fringing in the axial direction that is taken into account by the FEMM simulation, which is a 2D simulation package. However, the force is still above the calculated required force. The effects of fringing that made the force larger will also increase the DC bus voltage required by the power amplifiers to produce the desired force slew rate. This is however not a problem as the amplifiers can handle up to 80 V and the calculated required voltage is only 47 V. It can be seen that the forces measured for a decrease (positive displacements) in the air-gap between the pole producing the force and the rotor, is much lower than the simulated value. This is however, not a problem as a pole will always try to decrease an air-gap that is larger than nominal.

These measurements were retaken to prove that this force can also be produced at y reference positions other than the centre. These measurements are shown in Table 4-3

Table 4-3: AMB Force with variations in y

Displacement form centre(μm)	F_{amb} (N) $y=0 \mu\text{m}$	F_{amb} (N) $y=100 \mu\text{m}$	F_{amb} (N) $y=200 \mu\text{m}$
-200	117	117	108
-150	126	125	122
-100	138	138	134
-50	149	151	146
0	163	163	161

It can be seen that if the rotors centre point varies in the y-direction the achievable force in the x-direction will not be influenced by displacements of less than 100 μm in the y-direction. This was however expected and is not a problem as the maximum orbit of the rotor is not expected to be larger than 100 μm .

The force measurements were taken with an integration constant of 5000. This is to reduce the steady-state error to zero. This integration constant was only used for static tests and was not calculated but rather experientially determined. The mechanical stiffness was also tested and will be discussed in the next section.

4.5 STIFFNESS VERIFICATION

The mechanical stiffness produced by the AMB was tested by reducing the integration constant of the PID controller to zero. This is to allow the rotor to be pulled away from its centre position. A force was again applied to the rotor by using a spring scale at the same point as in the previous section. As a force is being applied to the rotor, it will deviate from the AMB's centre. A camera was used to display the measurements of the spring scale next to the position measurements obtained by the dSPACE® controller. From this information, the mechanical stiffness of the rotor can be calculated, as the stiffness is defined by the change in force on the rotor due to a change in the position of the rotor.

$$k_{eq} = \frac{\Delta F}{\Delta x} \text{ (N/m)} \quad (4.4)$$

The results of two tests are shown in Table 4-4 and Table 4-5. The tests were performed with a proportional gain of 10 000 and a derivative gain of 10. According to (3.18) and (3.19) in section 3.5, the stiffness of the AMB should be 303 600 N/m. The stiffness was calculated between two successive points, for example, between point 1 and 2, 2 and 3 etcetera as well as between the origin (point 1) and the other points.

Table 4-4: Measured AMB stiffness: Test no.1

Point	Position (μm)	Force Amb (N)	Stiffness (N/m) Calculated between two successive points	Percentage Deviation	Stiffness (N/m) Calculated to origin	Percentage Deviation
1	0	0.00				
2	50	11.65	233054.73	-23.24%	233054.73	-23.24%
3	85	23.31	332935.32	9.66%	274182.03	-9.69%
4	114	34.96	401818.49	32.35%	306650.96	1.00%
5	151	46.61	314938.82	3.73%	308681.76	1.67%
6	184	58.26	353113.22	16.31%	316650.44	4.30%
7	224	69.92	291318.41	-4.05%	312126.87	2.81%
Average:			321196.50	5.80%	291891.13	-3.86%

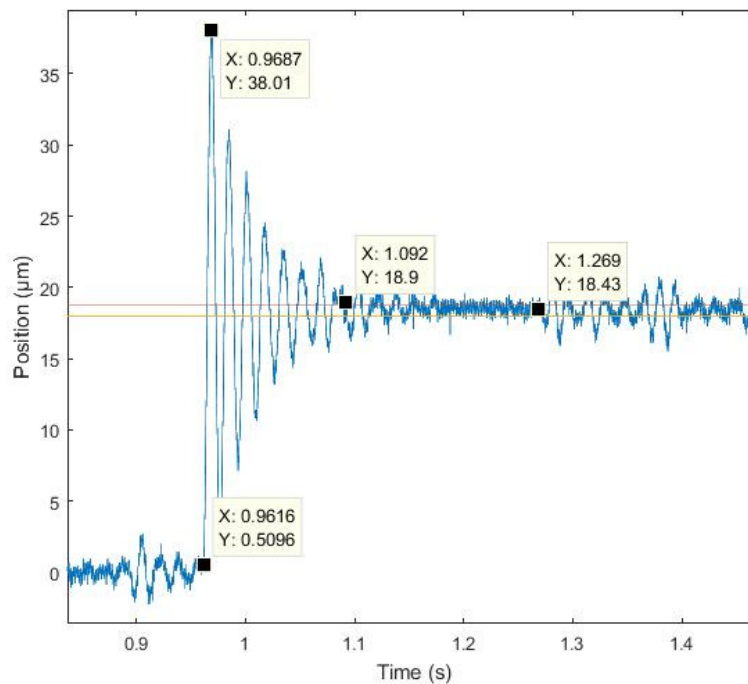
From Table 4-4 it can be seen that the measured stiffness varies a lot from the calculated stiffness at some points if calculated between two points with a 50μm difference. However, the stiffness that was calculated between the origin and points further than 100 μm from it, the measured stiffness showed a correlation within 5% of the calculated stiffness. This is because the values can't be read accurately from the spring scale. However, if the displacement size is increased, the error made is less significant. The same conclusion was made with the second test shown in Table 4-5.

Table 4-5: Measured AMB stiffness: test no.2

Point	Position (μm)	Force Amb (N)	Stiffness (N/m) Calculated between two successive points	Percentage Deviation	Stiffness (N/m) Calculated to origin	Percentage Deviation
1	0	0.00				
2	48	11.65	242765.34	-20.04%	242765.34	-20.04%
3	80	23.31	364148.01	19.94%	291318.41	-4.05%
4	123	34.96	270993.87	-10.74%	284213.08	-6.39%
5	156	46.61	353113.22	16.31%	298788.11	-1.58%
6	189	58.26	353113.22	16.31%	308273.45	1.54%
7	231	69.92	277446.10	-8.61%	302668.48	-0.31%
Average:			310263.29	2.19%	288004.48	-5.14%

The stiffness and damping were also measured by using the system's response to a disturbance in its reference position, as will be explained in the next.

According to literature, the AMB stiffness and damping can be measured by introducing a step disturbance to the rotors reference position while the rotor is being levitated [25]. This will be explained by using an example. Consider Figure 4-22, it shows the step response of the rotor's position with a step change of 10 μm in its reference position. The proportional gain of the PD controller is set to 10000 while the differential gain is set to 10.

Figure 4-22: 10 μm step response

The percentage overshoot can be determined as:

$$P.O. = \frac{18.35 - 38.39}{18.35} \times 100 \% \quad (4.5)$$

$$P.O. = 106.24 \%$$

The settling time to 2% of its final value can be seen as $1.092 - 0.9616 = 0.1304$ s. The damping ratio can be determined by rearranging (4.6):

$$P.O. = 100e^{\frac{-\pi\zeta}{\sqrt{1-\zeta^2}}} \quad (4.6)$$

This gives a damping ratio (ζ) of:

$$\zeta = \sqrt{\frac{[\ln(P.O.)]^2}{\pi^2 + [\ln(P.O.)]^2}} \quad (4.7)$$

$$\zeta = 0.019263$$

Next, the natural frequency can be determined as:

$$\omega_n = \frac{4}{T_s \zeta} \quad (4.8)$$

$$\omega_n = 1592.393 \text{ rad/s}$$

By using this, the stiffness can be calculated by using (4.9), with $m=2.22$ kg (Half the mass of the rotor as one AMB is only responsible for moving half of the rotor's mass.)

$$k_{eq} = \omega_n^2 \cdot m \quad (4.9)$$

$$k_{eq} = 5\,629\,290 \text{ N/m}$$

This is 18.54 times the calculated value of 303 600 N/m.

The damping can be calculated by using:

$$b_{eq} = \zeta \left(2 \sqrt{k_{eq} \cdot m} \right) \quad (4.10)$$

$$b_{eq} = 136 \frac{\text{N} \cdot \text{s}}{\text{m}}$$

This is 4.46 times smaller than the calculated value of 608 Ns/m. This value is however, not reliable as the stiffness was verified with another method as explained with Table 4-4. Another method of verifying the damping of the AMB could not be found. The verification of the AMB's damping was left for future work as no time was left to do this.

4.6 RESISTANCE AND INDUCTANCE MEASUREMENTS

The inductances and resistance of the coils were measured by using an RLC meter. The RLC meter used was the *digimiss® RLC 300 Programmable Automatic RLC meter*. The rotor was levitated at its magnetic centre and clamped using the backup bearings. These values were compared to the calculated values as specified in section 3.3. Firstly the inductance and resistance of the coils were measured while the other coils were left unconnected. These measurements were done for 50, 100 and 1000 Hz, the results are shown in Table 4-6.

Table 4-6: Resistance and inductance measurements

Coil	Inductance (50 Hz)	Resistance (50 Hz)	Inductance (100 Hz)	Resistance (100 Hz)	Inductance (1000 Hz)	Resistance (1000 Hz)
1	4.15 mH	157 mΩ	4.1 mH	206 mΩ	3.9 mH	1.41 Ω
2	4.14 mH	158 mΩ	4.07 mH	209 mΩ	3.8 mH	1.28 Ω
3	3.926 mH	152 mΩ	3.882 mH	198 mΩ	3.7 mH	1.33 Ω
4	3.934 mH	152 mΩ	3.887 mH	195 mΩ	3.6 mH	1.19 Ω

The increase of the resistance with frequency is due to skin effect, while the reduction of the inductance with frequency is due to the complex permeability of the core. The calculated resistance value of 101 mΩ relates closely to the measured values. The measured value will be higher than the calculated value as the calculated resistance is the DC resistance, the conductivity of copper used in the calculations can also vary from the conductivity of the wire used. The inductance measured in Table 4-6 shows the self-inductance of the coils as the measurements were taken with the other coils left open. The self-inductance was simulated using the same FEMM simulation used in section 3.6. The simulation was done by injecting 2.5 A in one coil and leaving the others open, the results of the simulation are shown in Figure 4-23.

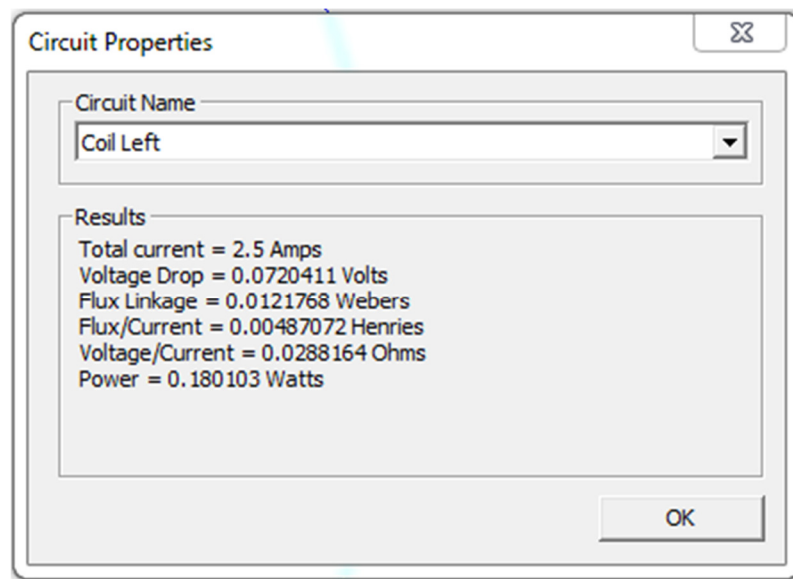


Figure 4-23: FEMM simulation of coil's self inductance

It can be seen that the self-inductance of the coil is 4.8 mH, this is close to the measured value. The variation can be due to the fact that the pole tips were not cut correctly increasing the effective air-gap, or the fact that the permeability of the core was not taken into account when calculating the inductance of the coil. The difference in inductance in the different coils can be attributed to the fact that it is very difficult to centre the rotor precisely.

The inductance of the coil was also measured with different current values passing through the coil. The inductance of coil 1 was measured with different current values passing through it, as well as with different current values in the other coils. The results are shown in Table 4-7. The method that was used to measure the inductance is shown in Figure 4-24. The capacitor C_∞ is used to protect the RLC meter from the DC source. A large capacitor will be an open circuit for DC, meaning that the

DC will only flow through the coils. Capacitor C_∞ will be a short circuit for the AC produced by the RLC meter, where the inductor L_∞ will have a very large reactance at 100 Hz meaning that the AC produced by the RLC meter will mostly just flow through coil 1. The reactance that the RLC meter measures will be the small reactance of the capacitor in series with the parallel combination of the two reactances of the coils. The inductor L_∞ will have a large reactance compared to the reactance of Coil 1. This means that the parallel combination that is measured will have a reactance slightly more than Coil 1. It is not expected that the addition of capacitor C_∞ and inductor L_∞ will have a significant impact on the measurement of the inductance of the coil in question.

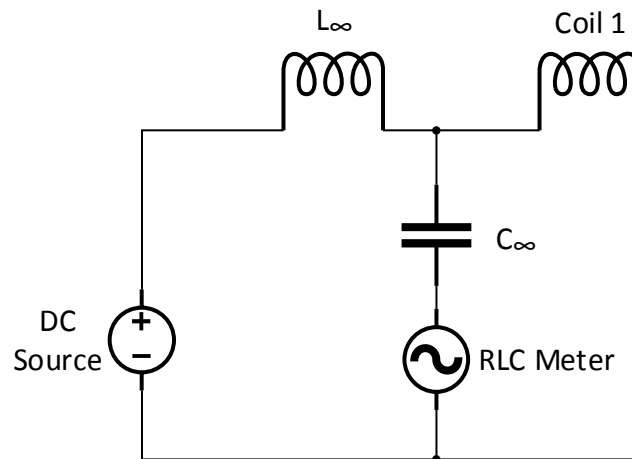


Figure 4-24: Inductance measurements vs current connection.

The inductance of Coil 1 was measured with different current values in it as well as different current values in the adjacent coils, note that coil 3 is adjacent to coil 1. All these measurements were done at 100 Hz and is shown in Table 4-7.

Table 4-7: Inductance of coil 1 vs coil current

I ₁	I _{2,3,4} =0 A	I _{2,3,4} =2.5 A	I _{2,4} =2.5 A ; I ₃ =5 A-I ₁
0	4.1 mH	6 mH	5.6 mH
0.5	4.1 mH	5.6 mH	5.6 mH
1	4.1 mH	5.2 mH	5.3 mH
1.5	4.05 mH	5.2 mH	5.2 mH
2	4 mH	5.2 mH	4.9 mH
2.5	3.8 mH	4.5 mH	4.6 mH
3	3.5 mH	4.3 mH	4.3 mH
3.5	3.3 mH	3.9 mH	3.9 mH
4	3 mH	3.5 mH	3.5 mH
4.5	2.9 mH	3.1 mH	3.2 mH
5	2.8 mH	3 mH	2.9 mH

It can be seen that the inductance of coil 1 with no DC current correlates with the inductance measured by using just the RLC meter. This shows that the extra components inserted to protect the RLC meter did not influence the measurements. It can be seen that the inductance of the coil decreases as the current in the coil increase. This can be due to saturation of the core or due to the change in the relative

permeability as the flux density of the core changes. The inductance of the coil with a change in the current is expected to behave like that in the last column of Table 4-7, as the flux in pole 1 due to coils 2 and 4 will stay the same if they are controlled differentially. Meaning that if the current in coil 2 and 4 are 2 A and 3 A, the flux in pole 1 should be the same as the if the current in coil 2 and 4 are 2.5 A. According to the control scheme coils, 1 and 3 will also be controlled differentially. This is why it is expected that the inductance in the coil will resemble that of the last column.

It can also be seen the inductance in coil 1 changes if coils 2-4 are biased. This is due to the mutual inductance effect. It can be seen that the inductance correlated closely to the calculated value of 6 mH if the adjacent coils are biased.

These inductance measurements may be useful in future projects, to assist with proper modelling of the system and controller tuning.

4.7 CONCLUSION

This chapter explained the modifications that were made to the TWINS to enable it to work on AMBs. It introduced the Simulink control model that was used to levitate the rotor during testing. It was very difficult to determine the geometrical centre point of the rotor. This is because the distance which the proximity sensor is located away from the rotor cannot be determined accurately. To overcome this, the magnetic centre point was determined. It does not necessarily coincide 100% with the geometrical centre but should be close enough. The force that the AMB can produce with the rotor in its centre was determined to be 162 N. This compares to a designed value of 150 N. The increase in force is not a problem, all this means is that the DC bus voltage on the power amplifiers will have to be more than 47 V. This is to ensure that the desired force slew rate can be achieved. The stiffness that the AMB produces with the PD control values set at $P=10\ 000$ and $D=10$ was determined to be within 7% of the calculated value. The damping that the AMB produces could not be verified as the method used didn't produce believable results.

CHAPTER 5: CONCLUSION

This project focused on the replacement of the TWINS' ball bearings with AMBs. This was done to solve the problem of the rotor overheating. Active magnetic bearings were chosen as they produce fewer losses than mechanical bearings as there are no mechanical friction between the rotor and stator. The only losses that will be present in the rotor are eddy current and hysteresis losses. The eddy current losses were reduced by laminating the core with 0.35 mm electrical steel. According to calculations, the AMBs will reduce the losses from 75 W to 8 W per bearing. The losses weren't measured as the rotor couldn't be spun up. This is due to the fact that the control of the PMSM still need to be completed, as it was not in the scope of this project due to time constraints. Only static tests were done to verify the design of the AMB.

This chapter will describe the mechanical feasibility that was considered to see whether the retrofit could be done. It describes the electromagnet that was designed and comparisons that were made when designing the electromagnet. The results of the tests that were performed on the AMBs to verify the design will be discussed, as well as the influence that the cut error in the laminations had on the performance of the AMBs. Lastly, future work that needs to be done will be discussed, followed by a closing statement.

5.1 MECHANICAL FEASIBILITY

As discussed in section 3.10, a stress analysis was done to determine whether the journal of the AMB will be able to be shrink-fitted to the rotor of the TWINS. The stress analysis determined that the interference between the outer diameter of the shaft and inner diameter of the journal should have a minimum value of 10 μm and a maximum value of 30 μm . A minimum of 10 μm interference is to ensure that the journal will stay in contact with the rotor while it rotates at 30'000 r/min. Due to manufacturing capabilities, the minimum tolerance that could be specified on the interference was 20 μm . This resulted in a maximum possible interference of 30 μm . It was seen that for an interference of 30 μm , the material would deform plastically. However, the material is not expected to deform more, as to produce an effective interference of less than 10 μm . From this analysis, it was concluded that the journal of the AMB will be able to be shrink-fitted to the rotor of the TWINS.

Section 3.11 investigated the locations of the rotor-AMB system's critical frequencies, to ensure that they don't fall within the operating speed of the motor. The 1st bending critical frequency was of most concern as it can't be moved by changing the control parameters of the AMB. It was calculated that the 1st bending mode occurs at 1100 Hz, which is more than double the operating frequency of the motor. This means that the unbalance of the rotor shouldn't excite the 1st bending frequency of the rotor. It also means that the rotor can be balanced by using two balancing planes, as it is considered a rigid rotor at its maximum operating frequency. This is because its maximum operating frequency is lower than half of its 1st bending critical frequency. Balancing holes at two different plains were manufactured into the rotor, as shown in section 4.1. This can be used in future projects to balance the rotor by adding grub screws. It was seen that the 1st and 2nd rigid mode locations could be moved by changing the stiffness of the AMBs. This should be taken into account when operating the motor, to ensure that the motor isn't operated close to one of these frequencies.

5.2 ELECTROMAGNET DESIGN

The electromagnet of the AMB was designed to be able to attract the rotor with a force of 150 N. A 4-pole and 8-pole electromagnet was designed and compared to each other. A full design of the electromagnet and comparison between the 4-pole and 8-pole electromagnet was done in Section 3.

The comparison was done to determine if there is cross-coupling between the forces in the x and y -direction and if so, is it more in the 4-pole than in the 8-pole configuration. When controlling the force produced by the electromagnet, the bottom and top coils are used to control the force in the y -direction, while the left and right coils are used to control the force in the x -direction. This simplifies the control of the rotor's position. For this reason, it is not desirable for the force on the rotor in the x -direction to change if the force in the y -direction is being controlled. This was tested by setting the x and y -position of the rotor both at 50 μm in all three different configurations, and applying a control current of 1 A to the top and bottom coils. This was done to apply a force on the rotor in the y -direction. The results are shown in . The second 4-pole(B) electromagnet is one that was designed with the same pole width as the 8-pole electromagnet. This was done to determine whether the curvature of the pole tip has an influence on the cross-coupling between the forces in the x and y -direction.

Table 5-1. The second 4-pole(B) electromagnet is one that was designed with the same pole width as the 8-pole electromagnet. This was done to determine whether the curvature of the pole tip has an influence on the cross-coupling between the forces in the x and y -direction.

Table 5-1: Change in force with a change in control current.

Electromagnet configuration	Control current (A)	F_x (N)	F_y (N)
4-pole (A)	0	16.4	16.4
	1	17.85	79.8
Percentage increase in Force		8.8%	386.6%
4-pole (B)	0	14.4	14.4
	1	15	76.68
Percentage increase in Force		4.2%	432.5%
8-pole (C)	0	15.4	15.4
	1	15.8	78.5
Percentage increase in Force		2.6%	409.7%

It can be seen that while trying to control the force in the y -direction the force in the x -direction also changed in all the topologies. It can be seen that the cross-coupling between the x and y -direction is the least for the 8-pole configuration. This is however, not just due to the number of poles as a large decrease in the 4-pole's cross-coupling was seen when the pole width was decreased. This means that the cross-coupling is also due to the curvature of the pole. The comparison between the designed electromagnets were explained in more detail in section 3.7. Regardless of the fact that the 4-pole electromagnet showed to have more cross-coupling than the 8-pole electromagnet, the 4-pole electromagnet was used as it has a simpler construction (fewer winding to turn), and the fact that it showed to produce fewer losses in the rotor. The 4-pole electromagnet was also desirable to enable the McTronX research group to do more research on the 4-pole configuration. It should be noted that the 4-pole(B) configuration couldn't be used as its axial length as to long. The 4-pole(A) electromagnet was manufactured. The force that it can produce was tested and will be shown next.

5.3 FORCE AND STIFFNESS VERIFICATIONS

The force that the electromagnet can produce was measured by levitating the rotor at different reference positions and using a spring scale to apply an opposing force onto the rotor. These measurements were explained in detail in Section 4.4.

The AMB was designed to be able to produce a Force of 150 N onto the rotor, if the rotor is situated at its centre position. The measured force at the centre position was 163 N, 8% higher than the designed value. This is not a problem as this would only mean that the DC voltage on the power amplifiers needs to be increased to ensure that the desired force slew rate can still be produced. The increase in the force is due to fringing effects that weren't taken into account with the design calculations. As explained in section 4.1 an error occurred when the laminations for the electromagnet were manufactured. This error effectively increased the air-gap in one of the poles. This error was reduced when stacking the laminations by spreading the error onto all 4 poles. According to the force measurements, this didn't have a big effect as the increase in the air-gap should have reduced the force that the AMBs can produce.

The mechanical stiffness produced by the AMBs was tested to determine if it correlates to the amount defined by the control parameters. When the test was performed, the control parameters were set to values that should produce a stiffness of 303'600 N/m. The stiffness was measured by applying a force to the rotor with a spring scale and measuring the displacement of the rotor. These results are shown in Table 4-4 and Table 4-5. It was seen that the stiffness correlates to within 6% of the calculated value.

The damping couldn't be measured as the method used to measure the damping and stiffness didn't produce believable results. The method considered the rotor-AMB system to be a second-order system, and using the rotor's overshoot and settling time to a step response to determine the stiffness and damping of the system. As this method didn't produce a stiffness even close to the previous method that correlated with the expected results, it was deemed to be inaccurate.

5.4 FUTURE WORK

The AMB system still needs to be operated at full speed to test the stability of the system and to measure the losses that are produced by the AMBs. Before this can be done the rotor should be balanced to a G1 balancing specification at a lower speed to ensure that the forces produced by the unbalance at 30'000 r/min are less than the maximum force than can be produced by the AMBs. The current control loop of the power amplifiers also needs to be investigated to determine whether it would be able to control its output current with stability at 500 Hz. The control system of the PMSM needs to be designed and implemented. The damping of the AMB system still needs to be verified as this couldn't be done reliably. The PD control topology can be replaced with unbalance control to control the rotor to rotate around its centre of mass rather than around its geometric centre.

BIBLIOGRAPHY

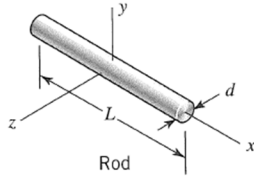
- [1] A. Grobler, S. Holm, and G. van Schoor, "Thermal modelling of a high speed permanent magnet synchronous machine," *Proceedings of the 2013 IEEE International Electric Machines & Drives Conference (IEMDC)*, pp. 319–324.
- [2] C. Ranft, "Stress in a multi-ring high speed rotor of a permanent magnet synchronous machine," *North-West University, Tech. Rep*, 2008.
- [3] Z. Huang and J. Fang, "Multiphysics design and optimization of high-speed permanent-magnet electrical machines for air blower applications," *IEEE Transactions on Industrial Electronics*, vol. 63, no. 5, pp. 2766–2774, 2016.
- [4] A. Muszynska, "Rotordynamics," CRC press, 2005
- [5] N. Uzhegov, A. Smirnov, C. H. Park, J. H. Ahn, J. Heikkinen, and J. Pyrhönen, "Design aspects of high-speed electrical machines with active magnetic bearings for compressor applications," *IEEE Transactions on Industrial Electronics*, vol. 64, no. 11, pp. 8427–8436, 2017.
- [6] A. H. Ahmad and D. A. Smit, "Modeling and design of a prototype radial magnetic bearing system," in *Universities Power Engineering Conference, 2006. UPEC'06. Proceedings of the 41st International*, 2006, vol. 2, pp. 679–684.
- [7] G. Schweitzer, H. Bleuler, and A. Traxler, "Active magnetic bearings-basics, properties and application of active magnetic bearings," *vdf, Hochschulverlag an der ETH Zürich, Zürich, Switzerland*, 1994.
- [8] H. Bleuler, M. Cole, P. Keogh, R. Larssonneur, E. Maslen, Y. Okada, G. Schweitzer, A. Traxler, G. Schweitzer, E. H. Maslen, "Magnetic bearings: theory, design, and application to rotating machinery," Springer, 2009.
- [9] C. Meeks, P. McMullen, D. Hibner, and L. Rosado, "Lightweight magnetic bearing system for aircraft gas turbine engines," *Proceedings of the 4th International Symposium on Magnetic Bearings*, August , 1994, pp. 429–434.
- [10] S. Xu and J. Fang, "A novel conical active magnetic bearing with claw structure," *IEEE Transactions on Magnetics*, vol. 50, no. 5, pp. 1–8, 2014.
- [11] C. A. da Fonseca, H. I. Weber, P. F. Fleischer, and I. Santos, "Analyzing the use of pins in safety bearings," *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, vol. 37, no. 4, pp. 1425–1434, 2015.
- [12] S. S. Rao, T. Quinn, "Mechanical vibrations", Fifth Edition, Pearson, 2011.
- [13] R. C. Dorf and R. H. Bishop, "Modern control systems," Twelfth Edition, Pearson, October, 2010.
- [14] J. L. M. Morrison and B. Crossland, "An introduction to the mechanics of machines" Longmans, 1982.

- [15] K. Adler, C. Schalk, R. Nordmann, and B. Aeschlimann, "Active balancing of a supercritical rotor on active magnetic bearings" *Proceedings of the 10th International Symposium on Magnetic Bearings*, 2006, pp. 49–54.
- [16] J. Bean, "Vibration characterization of an active magnetic bearing supported rotor," *Master's thesis, North-West University*, 2011.
- [17] J. J. Janse Van Rensburg, "Development of a Flywheel Energy Storage System: Uninterrupted Power Supply (FLY-UPS)," *Master's thesis, North-West University*, 2007.
- [18] E. O. Ranft, "The development of a flexible rotor active magnetic bearing system," *Master's thesis, North-West University*, 2005.
- [19] H. H. Mabie and F. W. Ocvirk, "Mechanisms and dynamics of machinery," Third Edition, Wiley, 1975.
- [20] R. W. Fitzgerald, "Mechanics of materials," Second Edition, Addison Wesley, 1982.
- [21] "Balance quality requirements of rigid rotors: the practical application of ISO 1940/1," IRD Balancing, 2009.
- [22] R. Larssonneur, "Design and control of active magnetic bearing systems for high speed rotation," *Ph.D thesis, Swiss federal institute of technology Zürich*, 1990.
- [23] C. Ranft, "Mechanical design and manufacturing of a high speed induction machine rotor," *Ph.D thesis, North-West University*, 2010.
- [24] P. Alaire, C.R. Knospe, "Short course on magnetic bearings," *University of Virginia*, 1997.
- [25] S. Myburgh, "The development of a fully suspended AMB system for a high-speed flywheel application," *Master's thesis, North-West University*, 2007.
- [26] D. F.B. David, J. A. Santisteban and A. C. Del Nero Gomes, "Interconnected four poles magnetic bearings simulation and testing," *Proceedings of the 14th International Symposium on Magnetic Bearings*, August , 2014, pp. 30–35.
- [27] E. O. Ranft, "An improved model for self-sensing heteropolar active magnetic bearings," *Ph.D thesis, North-West University*, 2007.
- [28] G. Genta and F. F. Ling, "Dynamics of Rotating Systems," Springer, 2005.
- [29] Y. Xu, "Kilowatt Three-phase Rotary Transformer Design for Permanent Magnet DC Motor with On-rotor Drive System," *Master's thesis, MID Sweden University*, 2016.
- [30] R. C. Juvinall and K. M. Marshek, "Fundamentals of machine component design," 4th Wiley international student edition, Wiley New York, 2005.

APPENDIX A:

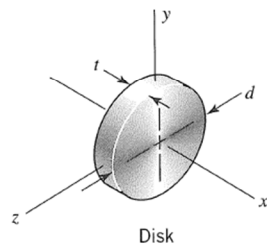
This Appendix will provide equations to calculate the mass and mass moment of inertia of homogeneous [30] solids as well as equations of typical beam deflections [20].

ρ = mass density



$$m = \frac{\pi d^2 L \rho}{4}$$

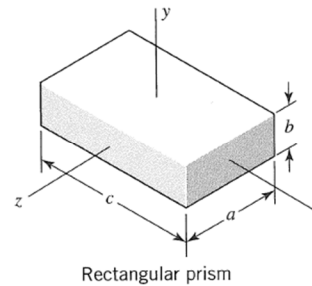
$$I_y = I_z = \frac{m L^2}{12}$$



$$m = \frac{\pi d^2 t \rho}{4}$$

$$I_x = \frac{m d^2}{8}$$

$$I_y = I_z = \frac{m d^2}{16}$$

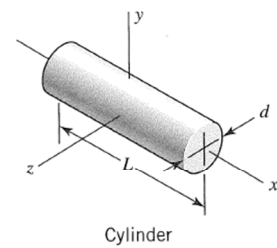


$$m = abc\rho$$

$$I_x = \frac{m}{12}(a^2 + b^2)$$

$$I_y = \frac{m}{12}(a^2 + c^2)$$

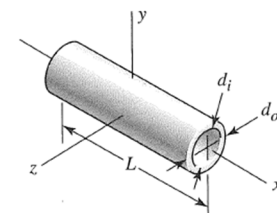
$$I_z = \frac{m}{12}(b^2 + c^2)$$



$$m = \frac{\pi d^2 L \rho}{4}$$

$$I_x = \frac{m d^2}{8}$$

$$I_y = I_z = \frac{m}{48}(3d^2 + 4L^2)$$



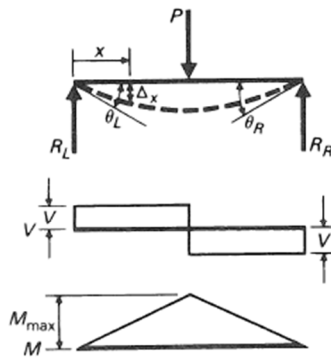
$$m = \frac{\pi L \rho}{4}(d_o^2 - d_i^2)$$

$$I_x = \frac{m}{8}(d_o^2 + d_i^2)$$

$$I_y = I_z = \frac{m}{48}(3d_o^2 + 3d_i^2 + 4L^2)$$

Figure A-1: Mass and mass moments of inertia of homogeneous solids [30]

Beam Diagrams



Case 1.

Case 1 Simply supported beam—concentrated load at center.

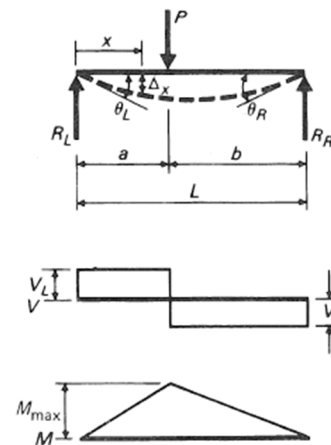
$$R_L = R_R = V = \frac{P}{2},$$

$$M_{\max} = \frac{PL}{4} \quad \text{at} \quad x = \frac{L}{2},$$

$$\theta_L = \theta_R = \frac{PL^2}{16EI},$$

$$\Delta_x = \frac{Px}{48EI} (3L^2 - 4x^2) \quad 0 < x < \frac{L}{2},$$

$$\Delta_{\max} \text{ (at centerline)} = \frac{PL^3}{48EI}.$$



Case 2.

Case 2 Simply supported beam—concentrated load at any point.

$$R_L = V_L = \frac{Pb}{L},$$

$$R_R = V_R = \frac{Pa}{L},$$

$$M_{\max} = \frac{Pab}{L} \text{ (at point of application of load),}$$

$$\theta_L = \frac{Pb(L^2 - b^2)}{6LEI},$$

$$\theta_R = \frac{Pab(2L - b)}{6LEI},$$

$$\Delta_x = \frac{Pbx}{6LEI} (L^2 - x^2 - b^2) \quad 0 < x < a,$$

$$\Delta_x = \frac{Pb}{6LEI} \left[\frac{L}{6} (x - a)^3 + (L^2 - b^2)x - x^3 \right] \quad a < x < L,$$

$$\Delta_{\max} = \frac{Pab(a + 2b)\sqrt{3a(a + 2b)}}{27LEI}.$$

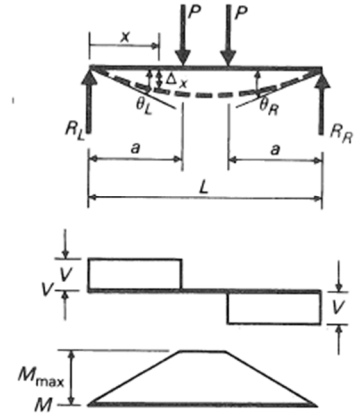
514

Figure A-2: Beam deflection diagrams 1 [20]

APPENDIX D / BEAM DIAGRAM 515

Case 3 Simply supported beam—two equal concentrated loads symmetrically placed.

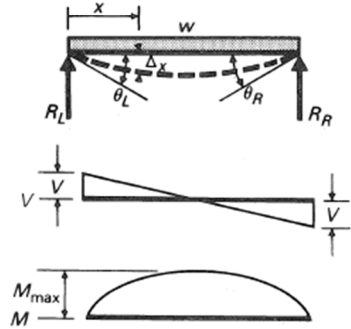
$$\begin{aligned}
 R_L &= R_R = V = P, \\
 M_{\max} &= Pa, \\
 \theta_L &= \theta_R = \frac{Pa}{2EI}(L-a), \\
 \Delta_x &= \frac{Px}{6EI}(3La - 3a^2 - x^2) \quad 0 < x < a, \\
 \Delta_{\max} &= \frac{Pa}{24EI}(3L^2 - 4a^2) \quad \text{at centerline.}
 \end{aligned}$$



Case 3.

Case 4 Simply supported beam—uniformly distributed load.

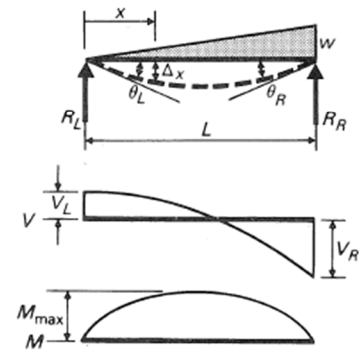
$$\begin{aligned}
 R_L &= R_R = V = \frac{wL}{2}, \\
 M_{\max} &= \frac{1}{8}wL^2 \quad \text{at centerline,} \\
 \theta_L &= \theta_R = \frac{wL^3}{24EI}, \\
 \Delta_x &= \frac{wx}{24EI}(L^3 - 2Lx^2 + x^3), \\
 \Delta_{\max} &= \frac{5wL^4}{384EI} \quad \text{at centerline.}
 \end{aligned}$$



Case 4.

Case 5 Simply supported beam—uniformly varying distributed load.

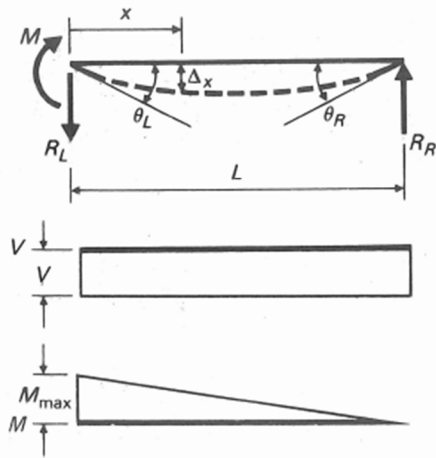
$$\begin{aligned}
 R_L &= V_L = \frac{wL}{6}, \\
 R_R &= V_R = \frac{wL}{3}, \\
 M_{\max} &= 0.0642wL^2 \quad \text{at } x = 0.577L, \\
 \theta_L &= \frac{7wL^3}{360EI}, \\
 \theta_R &= \frac{wL^3}{45EI}, \\
 \Delta_x &= \frac{wx}{360LEI}(7L^4 - 10L^2x^2 + 3x^4), \\
 \Delta_{\max} &= 0.00652 \frac{wL^4}{EI} \quad \text{at } x = 0.519L.
 \end{aligned}$$



Case 5.

Figure A-3: Beam deflection diagrams 2 [20]

516 APPENDIX D/BEAM DIAGRAM



Case 6.

Case 6 Simply supported beam—couple at one end.

$$R_L = R_R = V = \frac{M}{L},$$

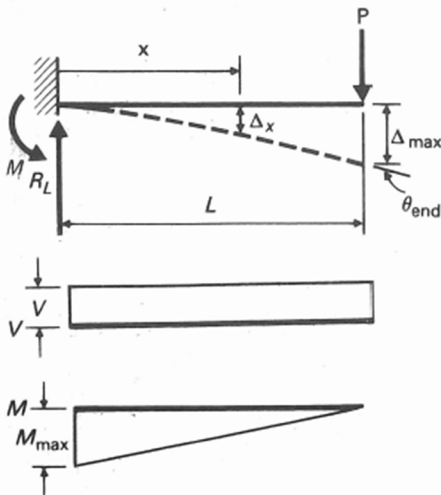
$$M_{\max} = M,$$

$$\theta_L = \frac{ML}{3EI},$$

$$\theta_R = \frac{ML}{6EI},$$

$$\Delta_x = \frac{Mx}{6LEI} (2L^2 - 3Lx + x^2),$$

$$\Delta_{\max} = \frac{0.064ML^2}{EI} \text{ at } x = 0.423L.$$



Case 7.

Case 7 Cantilever beam—concentrated force at the free end.

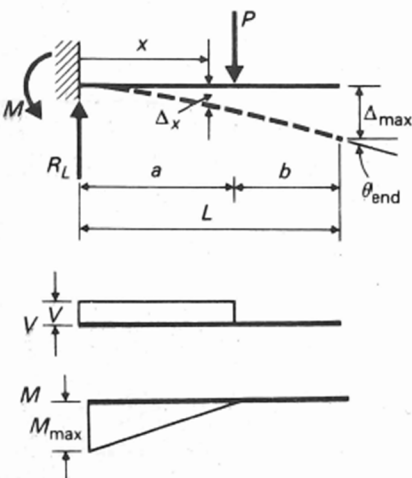
$$R_L = V = P,$$

$$M_{\max} = PL \text{ at } x = 0,$$

$$\theta_{\text{end}} = \frac{PL^2}{2EI},$$

$$\Delta_x = \frac{Px^2}{6EI} (3L - x),$$

$$\Delta_{\max} = \frac{PL^3}{6EI} \text{ at free end.}$$



Case 8.

Case 8 Cantilever beam—concentrated force at any location.

$$R_L = V = P,$$

$$M_{\max} = Pa \text{ at } x = 0,$$

$$\theta_{\text{end}} = \frac{Pa^2}{2EI},$$

$$\Delta_x = \frac{Px^2}{6EI} (3a - x) \quad 0 < x < a,$$

$$\Delta_x = \frac{Pa^2}{6EI} (3x - a) \quad a < x < L,$$

$$\Delta_{\max} = \frac{Pa^2}{6EI} (3L - a) \text{ at free end.}$$

Figure A-4: Beam deflection diagrams 3 [20]

APPENDIX D / BEAM DIAGRAM 517

Case 9 Cantilever beam—uniformly distributed load.

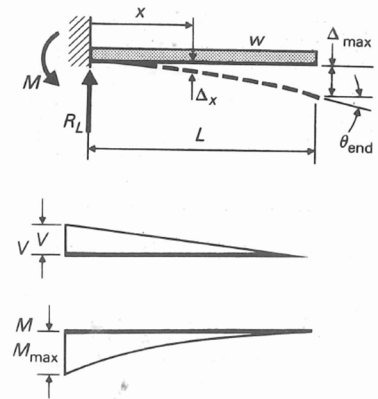
$$R_L = V = wL,$$

$$M_{\max} = \frac{wL^2}{2},$$

$$\theta_{\text{end}} = \frac{wL^3}{6EI},$$

$$\Delta_x = \frac{wx^2}{24EI} (6L^2 - 4Lx + x^2),$$

$$\Delta_{\max} = \frac{wL^4}{8EI}.$$



Case 9.

Figure A-5: Beam deflection diagrams 4 [20]

APPENDIX B: DATA CD

B.1 DOCUMENTATION

B.2 SOLIDWORKS[®] OF NEW ASSEMBLY

B.3 MATLAB SOURCE CODE

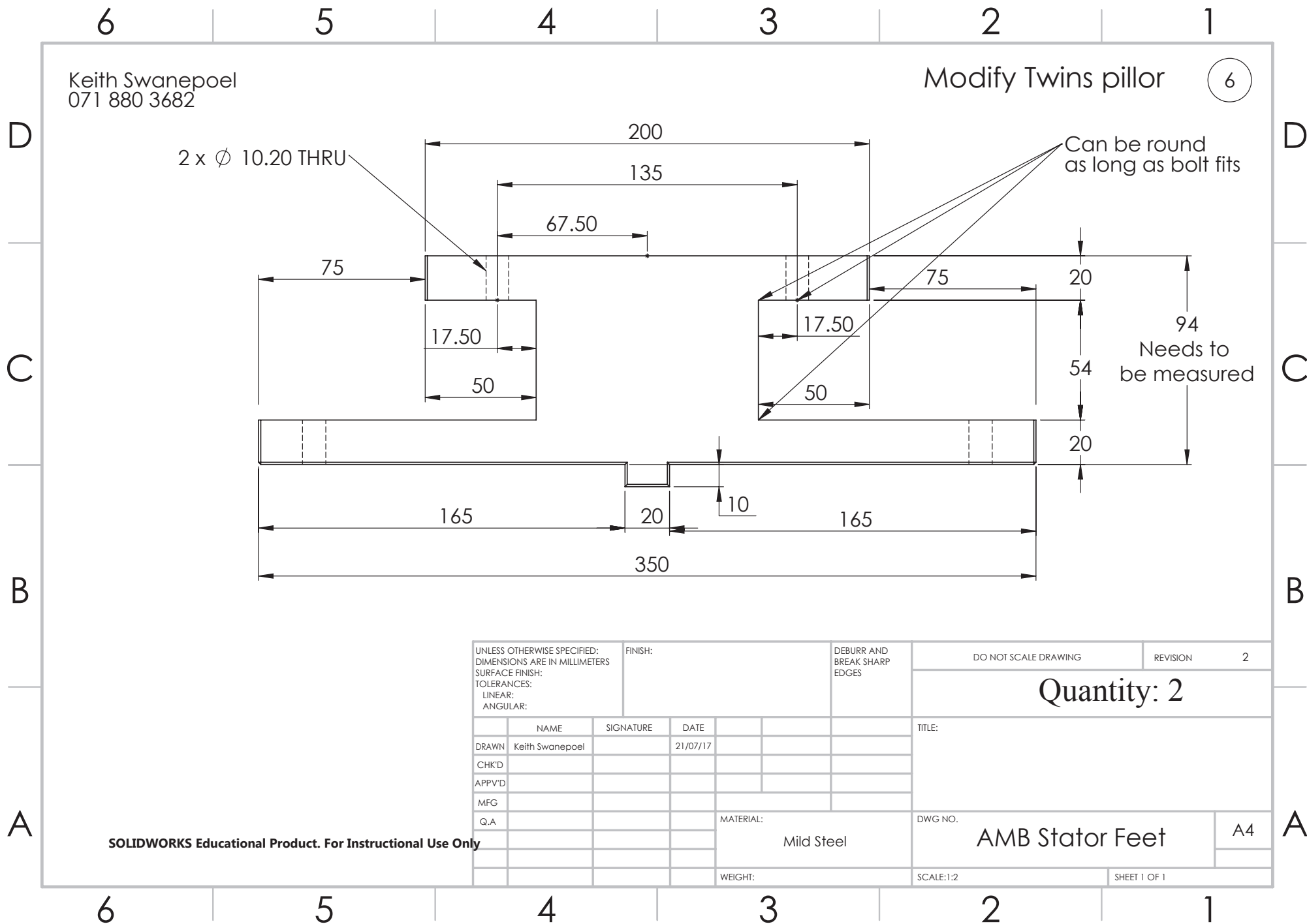
B.4 PICTURES

B.5 DATA SHEETS

B.6 LITERATURE

B.7 TWINS OLD DRAWINGS

APPENDIX B.2: NEW ASSEMBLY DRAWINGS



Keith Swanepoel
071 880 3682

Modify Twins pillar

6

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

2

Quantity: 2

TITLE:

	NAME	SIGNATURE	DATE			
DRAWN	Keith Swanepoel		21/07/17			
CHK'D						
APP'D						
MFG						
Q.A						

MATERIAL:

Mild Steel

DWG NO.

AMB Stator Feet

A4

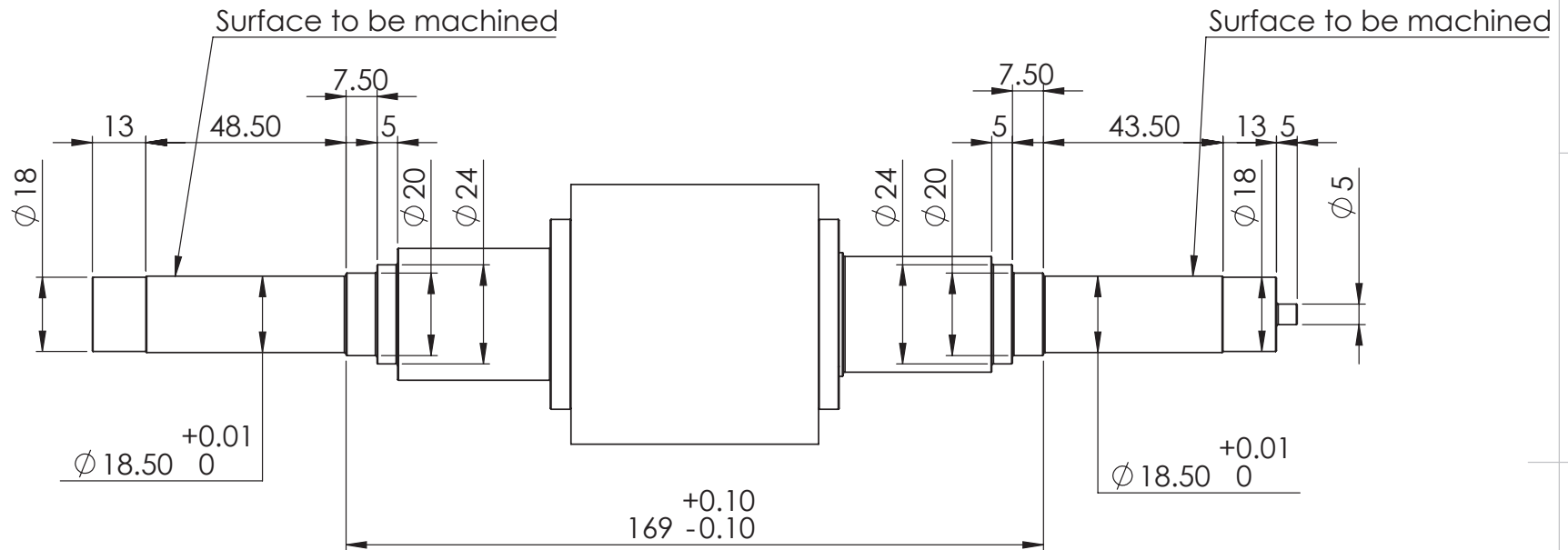
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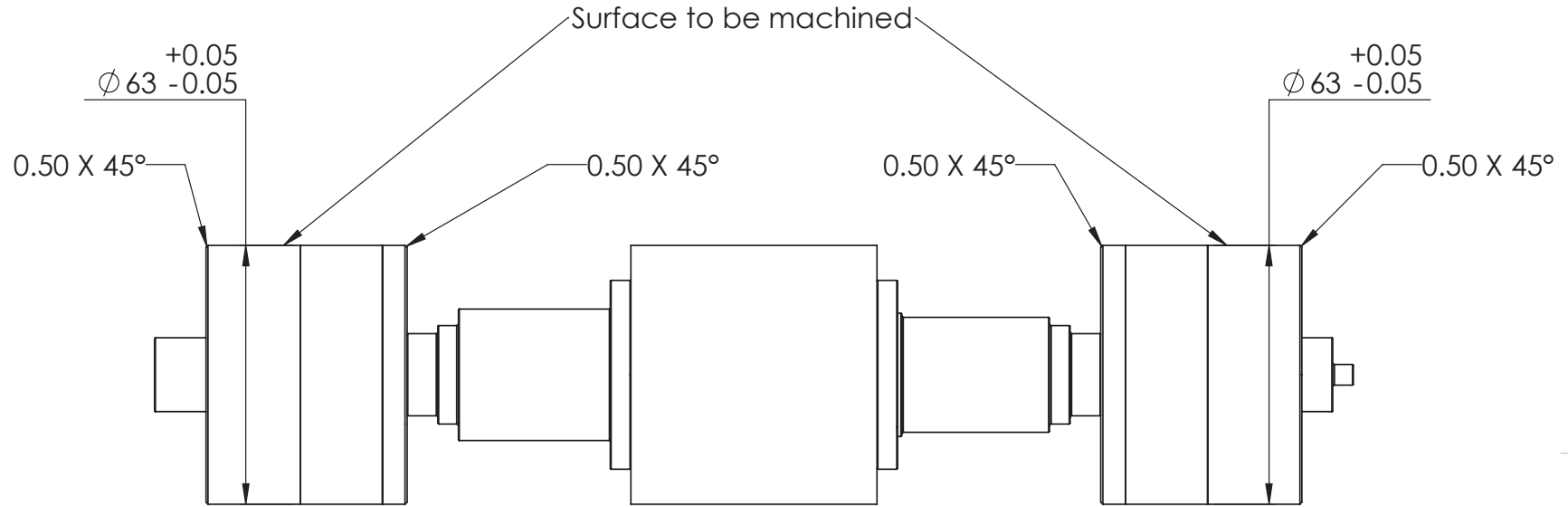
SHEET 1 OF 1

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071 880 3682



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN				NAME		SIGNATURE		DATE		TITLE:	
CHK'D											
APP'D											
MFG											
SOLIDWORKS Educational Product. For Instructional Use Only								MATERIAL:		DWG NO.	
										Rotor machined	
								WEIGHT:		SCALE:1:1.5	
										SHEET 1 OF 1	

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071 880 3682



UNLESS OTHERWISE SPECIFIED:
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TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

1

Quantity 1

TITLE:

	NAME	SIGNATURE	DATE			
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CHK'D						
APP'V'D						
MFG						

MATERIAL:

DWG NO.

Rotor Final

A4

WEIGHT:

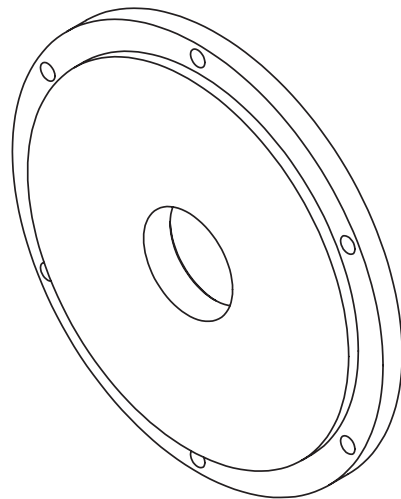
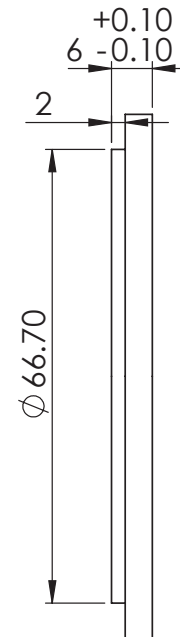
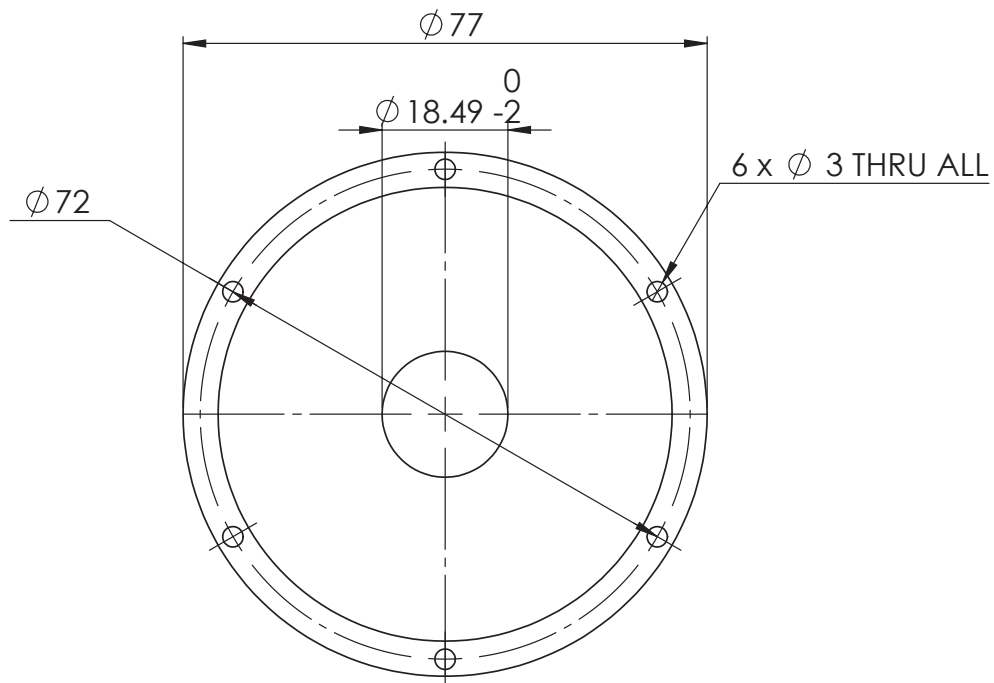
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SHEET 1 OF 1

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B



SOLIDWORKS Educational Product. For Instructional Use Only.

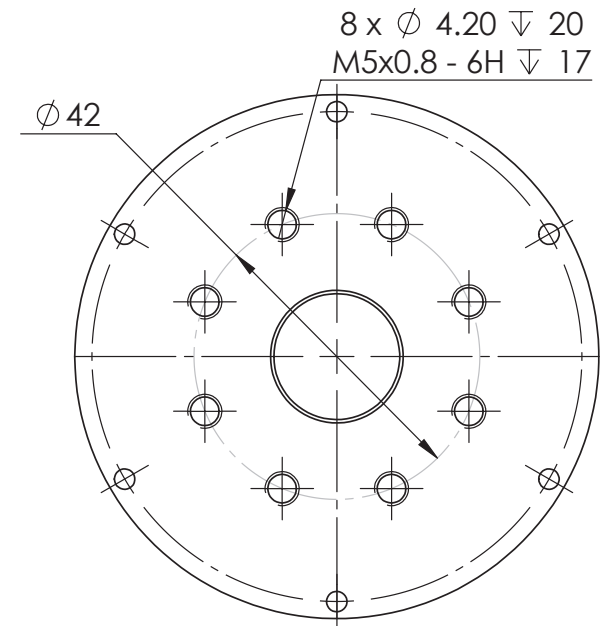
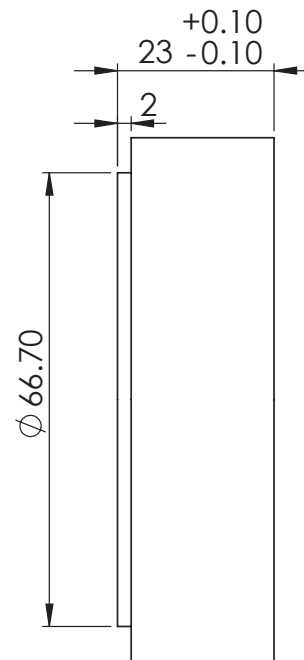
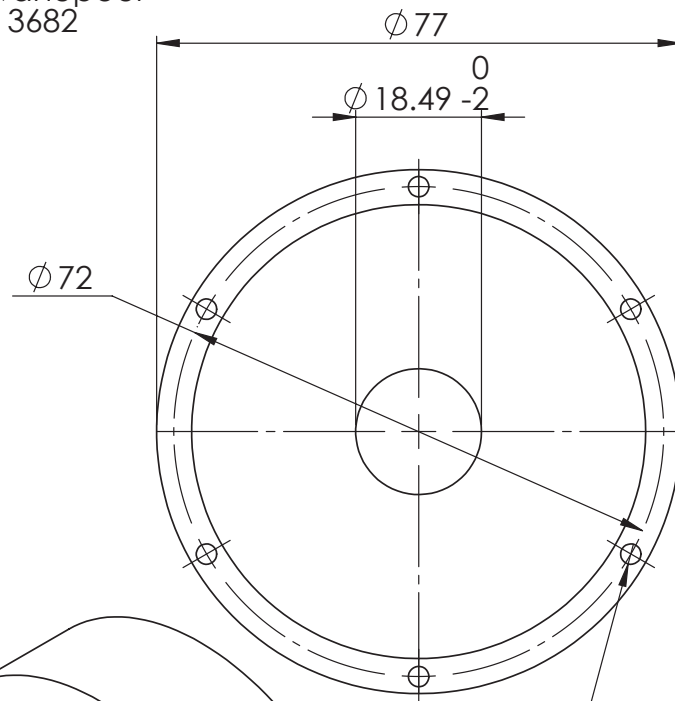
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION 2	
								Quantity: 2			
								TITLE:			
								DWG NO.			
								Endstop Inside			
								A4			
								SCALE: 1:1			
								SHEET 1 OF 1			

NAME	SIGNATURE	DATE
DRAWN Keith Swanepoel		21/07/17
CHK'D		
APPV'D		
MFG		

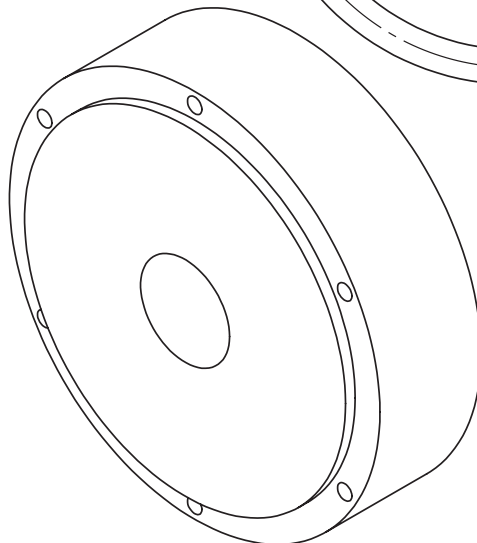
MATERIAL:	
AISI 304	
WEIGHT:	

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071 880 3682

C



6 x ϕ 3 THRU ALL



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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

3

Quantity 2

	NAME	SIGNATURE	DATE			
DRAWN	Keith Swanepoel		21/07/17			
CHK'D						
APP'D						
MFG						

MATERIAL:

AISI 304

WEIGHT:

TITLE:

DWG NO.

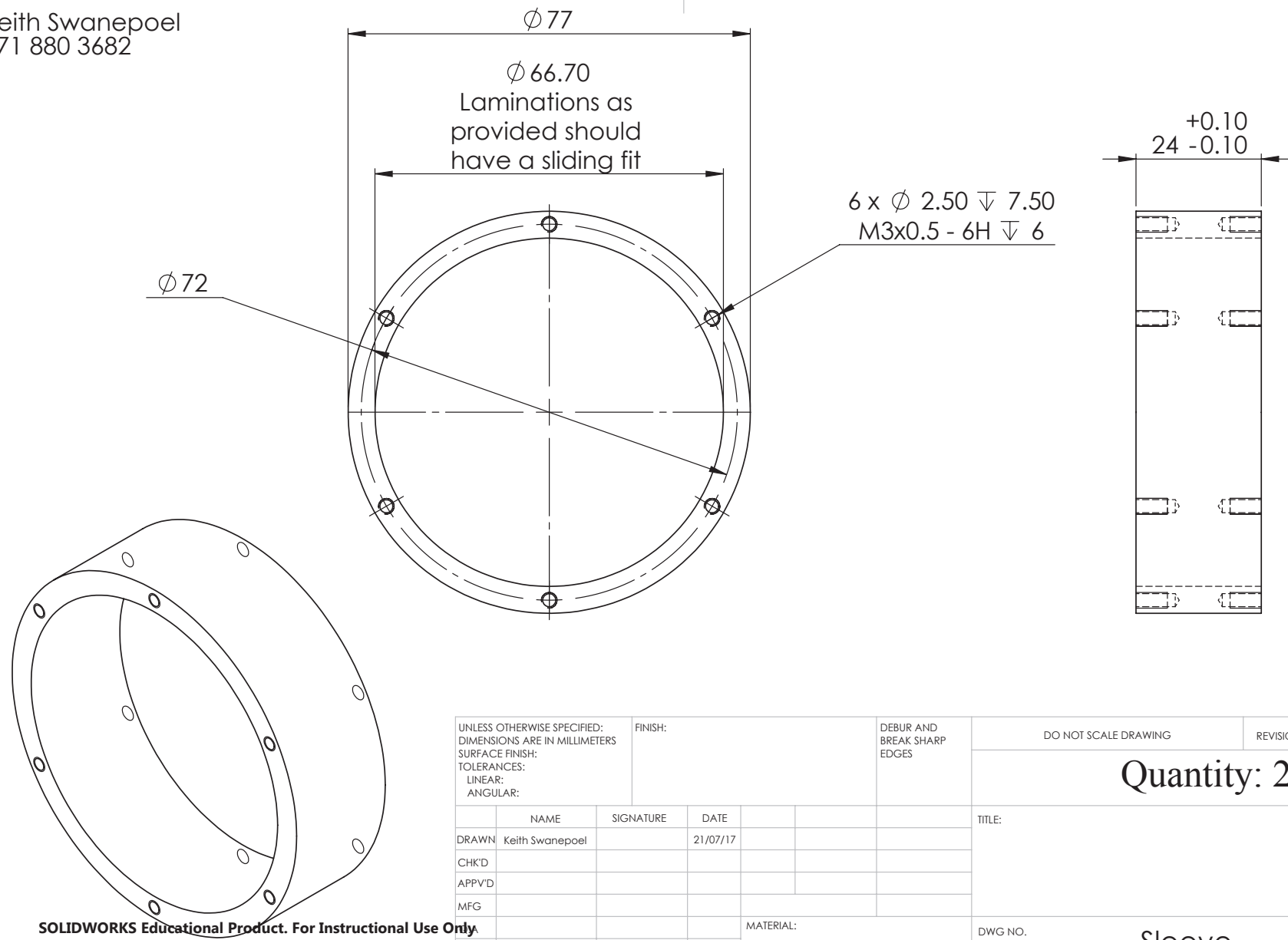
Endstop Outside

A4

SCALE:1:1

SHEET 1 OF 1

Keith Swanepoel
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UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION 2	
								Quantity: 2			
								TITLE:			
								DWG NO. Sleeve A4			
								SCALE: 1:1			
								SHEET 1 OF 1			

	NAME	SIGNATURE	DATE
DRAWN	Keith Swanepoel		21/07/17
CHK'D			
APP'D			
MFG			

MATERIAL:	
AISI 304	
WEIGHT:	

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6 5 4 3 2 1

D

C

B

A

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$\varnothing 18.49 \begin{matrix} 0 \\ -0.01 \end{matrix}$

Chamfer 0.5mm at 45°

B

A

C

D

Item	Part Number	Comments
A	Sleeve	Manufacture
B	Endstop Inside	Manufacture
C	Endstop Outside	Manufacture
D	Laminations	Supplied

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DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION 1

Quantity: 2

TITLE:

DWG NO.
Laminations_press_fit_jig A4

MATERIAL:

WEIGHT:

SCALE:1:1 SHEET 1 OF 1

6 5 4 3 2 1

6

5

4

3

2

1

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9

Needs to fit into M10 holes in Lid1 and Lid2

110

45

R15

Ø 9

M10X1

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

2

Quantity: 8

TITLE:

	NAME	SIGNATURE	DATE			
DRAWN	Keith Swanepoel		22/08/17			
CHK'D						
APP'V'D						
MFG						
Q.A						

MATERIAL:

Stainless

DWG NO.

Backup Bearing (BB)

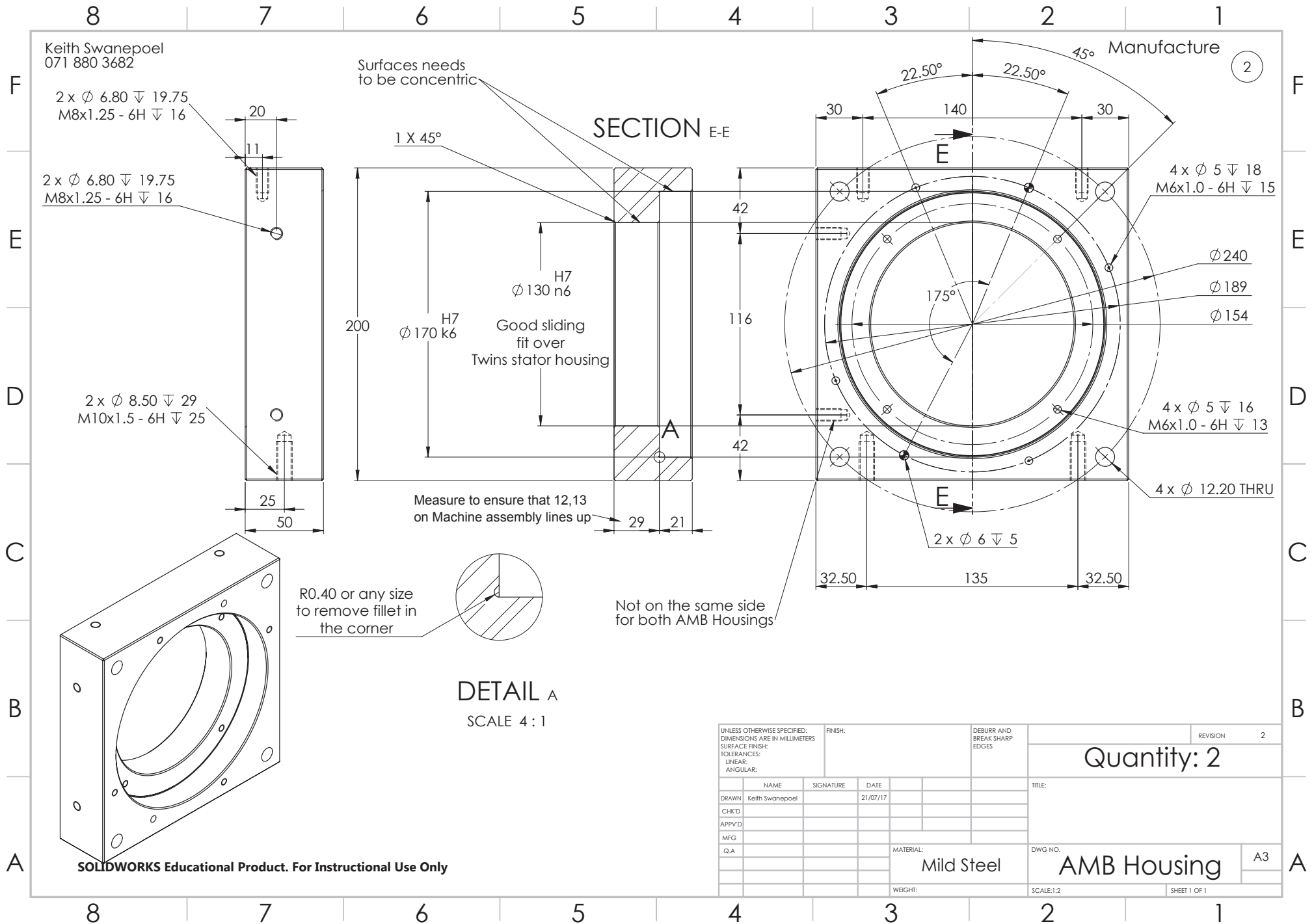
A4

WEIGHT:

SCALE:1:0.5

SHEET 1 OF 1

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4 x ϕ 6.20 THRU

ϕ 17 THRU

ϕ 189

22.50°
18°
59

ϕ 80

2 x ϕ 6 ∇ 5

ϕ 189

ϕ 16 or
bit size
smaller
than ϕ 20

Manufacture

3

ϕ 172

ϕ 206

175°

Size can be adjusted
for tool size

2 x ϕ 5.50 THRU
M6x0.5 - 6H THRU

45°

18
12
38
50
63

4 x ϕ 8 ∇ 50

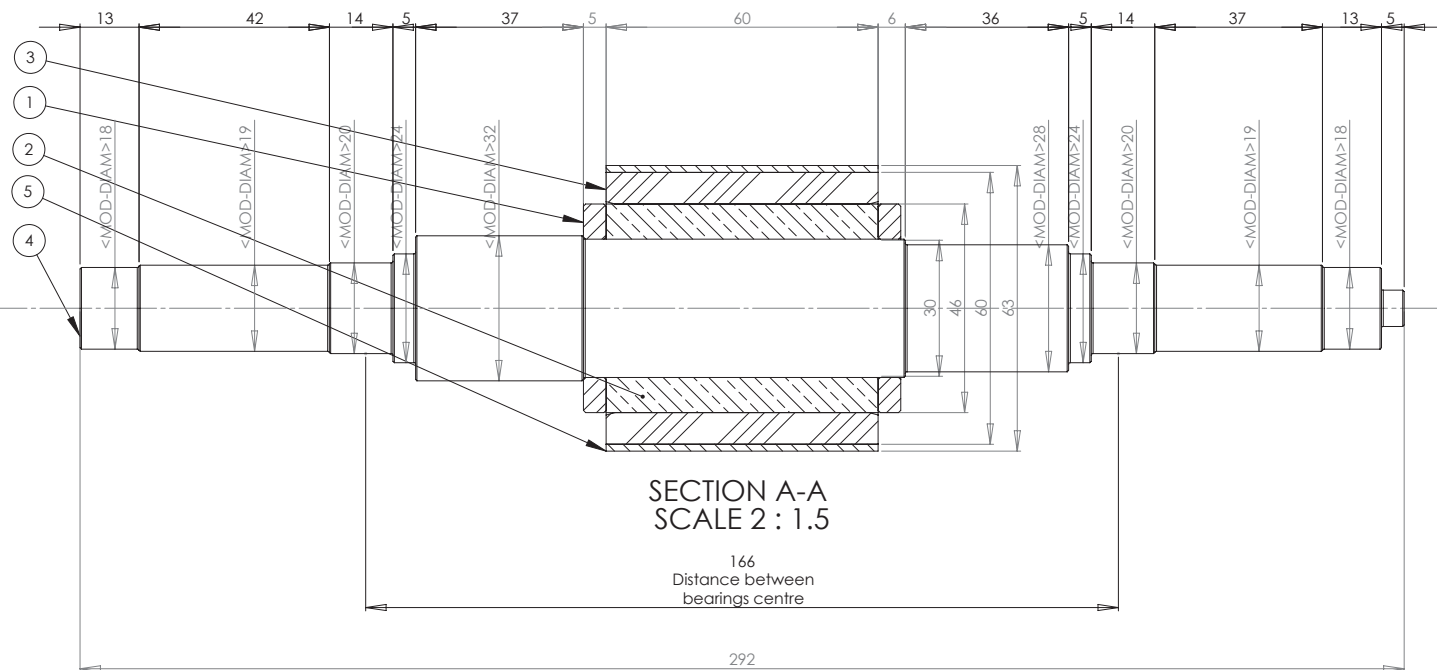
2 x ϕ 4.20 THRU
M5x0.8 - 6H THRU

4 x ϕ 9 THRU
M10x1.0 - 6H ∇ 30

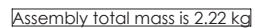
73

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS				FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION 2	
SURFACE FINISH:								Quantity: 1			
TOLERANCES:								TITLE:			
LINEAR:											
ANGULAR:											
NAME		SIGNATURE		DATE							
DRAWN Keith Swanepoel				21/07/17							
CHK'D											
APPV'D											
MFG											
Q.A											
MATERIAL:						Mild steel		DWG NO.		A3	
WEIGHT:								SCALE:1:2.5		SHEET 1 OF 1	

APPENDIX B.7: TWINS OLD DRAWINGS



166
Distance between
bearings centre



UNLESS STATED OTHERWISE:
REMOVE ALL BURRS AND
SHARP EDGES

SCREW THREAD: ISO 6H 6g

GEOMETRIC TOL. TO BS 308 1972

WELDING SYMBOLS TO BS 499/2

MACHINING TOLERANCES
UNLESS STATED OTHERWISE:

OVER - TO	±	±	±
0 - 6	0.1	0.2	0.5
6 - 30	0.2	0.5	0.8
30 - 100	0.3	0.8	1.5
100 - 300	0.5	1.2	2.0
300 - 1000	0.8	2.0	3.0
1000 - 3000	1.2	3.0	5.0
3000 PLUS:	2.0	4.0	8.0
ANGLES:	1°	1°	1°
SURFACE FIN:	1.6	6.3	12.5

Technical drawing of a tapered shaft with a hole. The front view (left) shows a circle with a crosshair indicating the hole. The side view (right) shows a trapezoid representing the tapered shaft.

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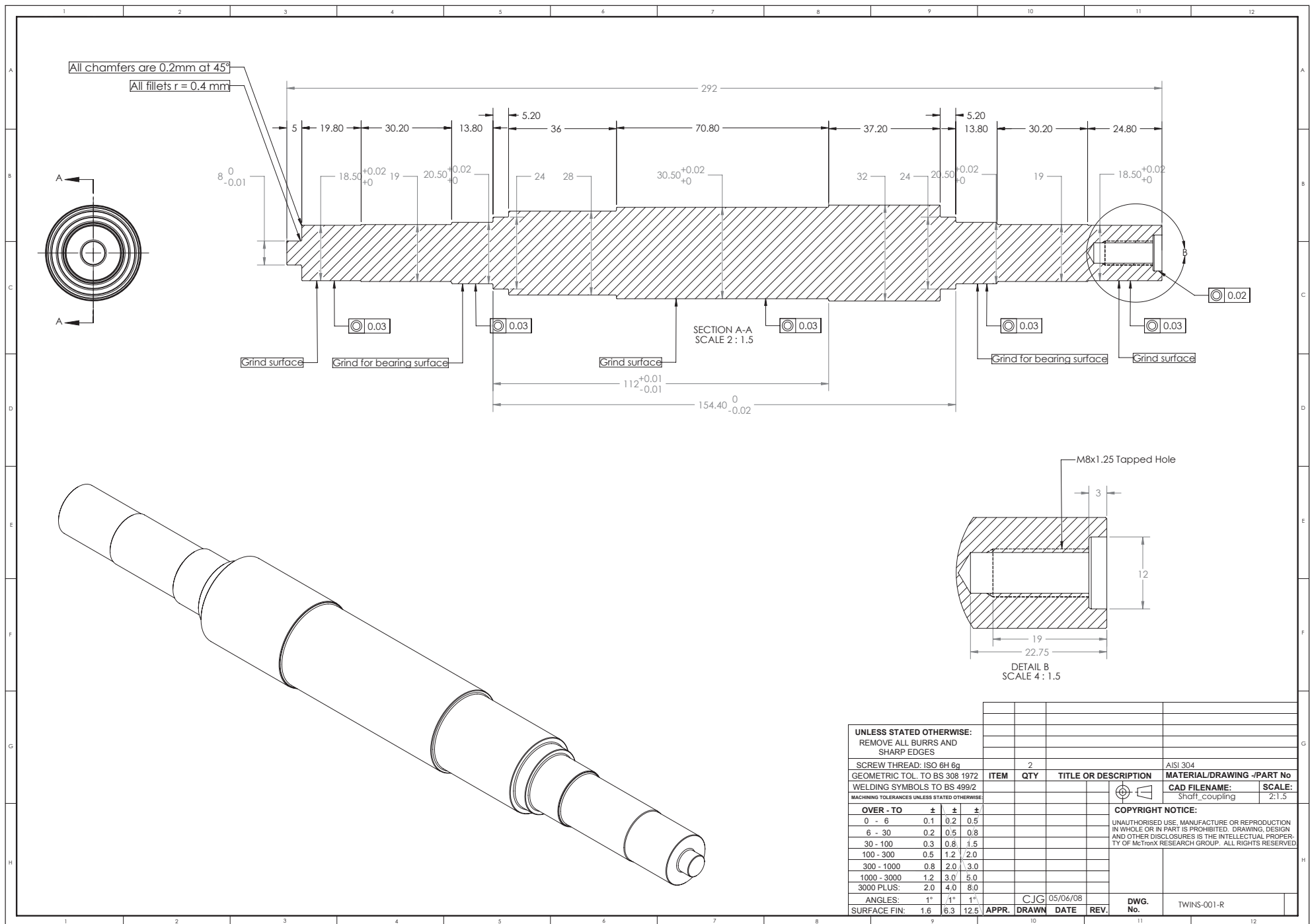
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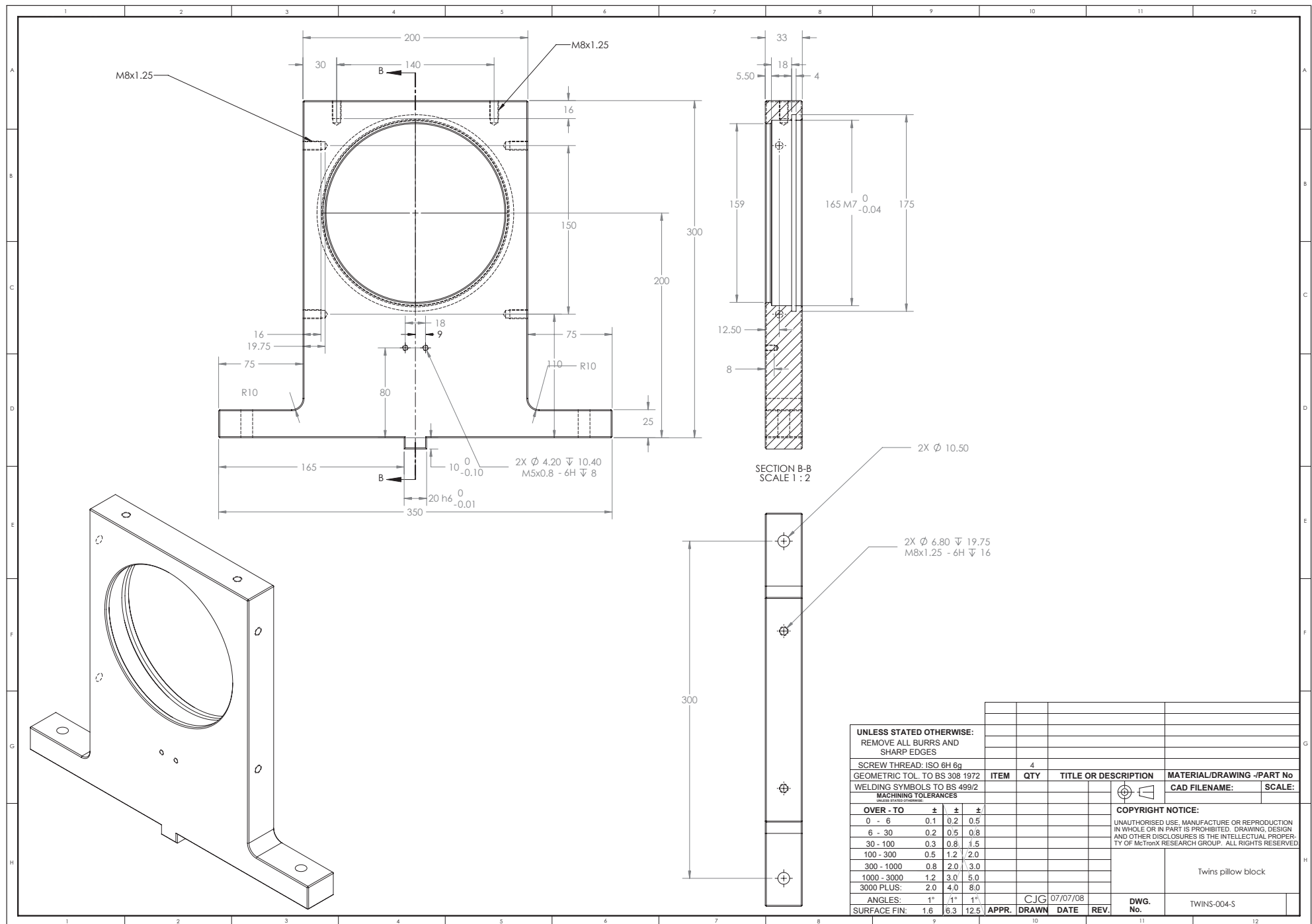
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DWG.		
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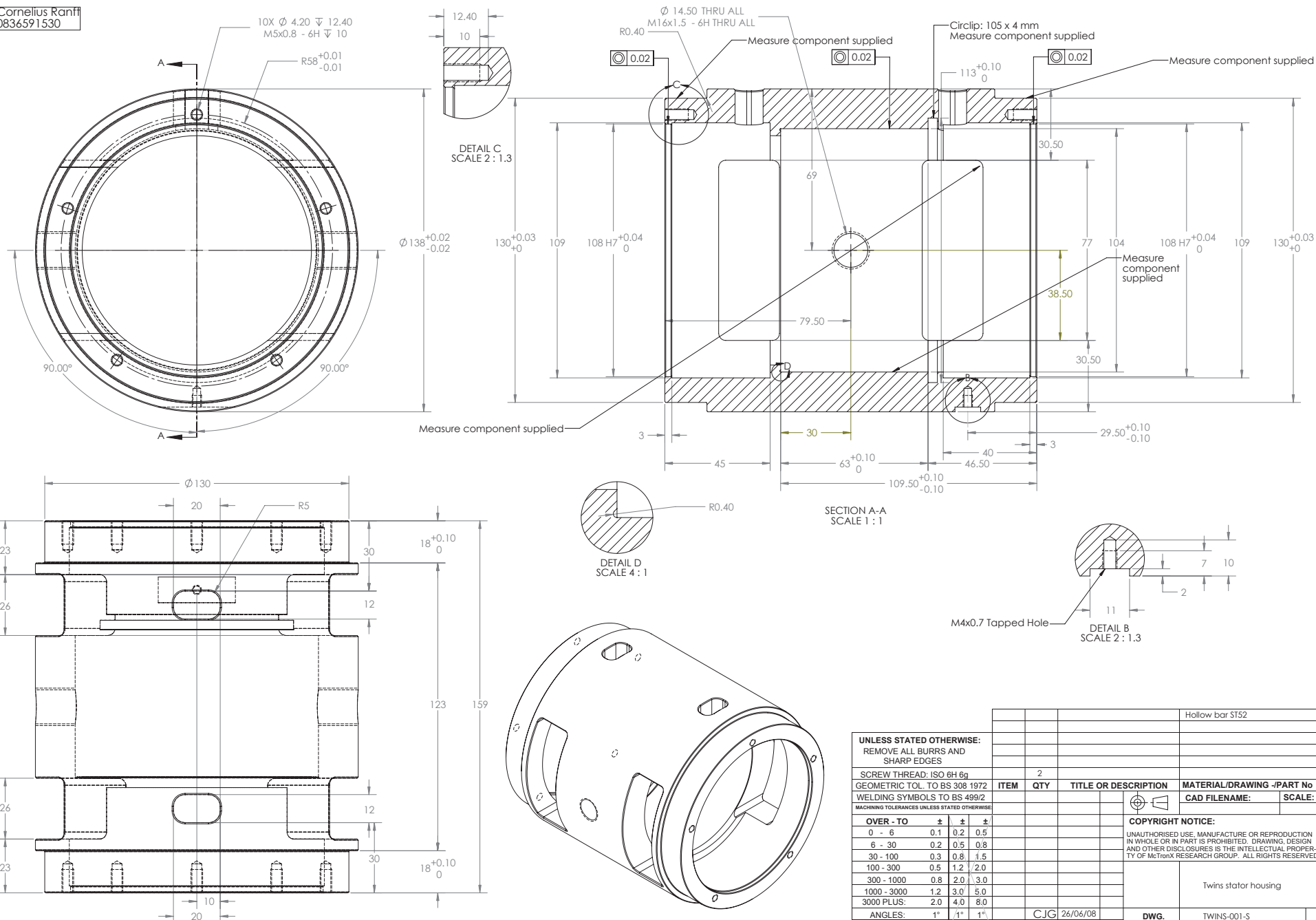
V.	No.						
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11	12
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Cornelius Ranft
0836591530



					Hollow bar ST52																																	
UNLESS STATED OTHERWISE: REMOVE ALL BURRS AND SHARP EDGES																																						
SCREW THREAD: ISO 8H 6g				2																																		
GEOMETRIC TOL. TO BS 308 1972			ITEM	QTY	TITLE OR DESCRIPTION	MATERIAL/DRAWING -PART No																																
WELDING SYMBOLS TO BS 499/2						CAD FILENAME:																																
MACHINING TOLERANCES UNLESS STATED OTHERWISE						SCALE:																																
<table><tr><th>OVER - TO</th><th>±</th><th>±</th><th>±</th></tr><tr><td>0 - 6</td><td>0.1</td><td>0.2</td><td>0.5</td></tr><tr><td>6 - 30</td><td>0.2</td><td>0.5</td><td>0.8</td></tr><tr><td>30 - 100</td><td>0.3</td><td>0.8</td><td>1.5</td></tr><tr><td>100 - 300</td><td>0.5</td><td>1.2</td><td>2.0</td></tr><tr><td>300 - 1000</td><td>0.8</td><td>2.0</td><td>3.0</td></tr><tr><td>1000 - 3000</td><td>1.2</td><td>3.0</td><td>5.0</td></tr><tr><td>3000 PLUS:</td><td>2.0</td><td>4.0</td><td>8.0</td></tr></table>			OVER - TO	±	±	±	0 - 6	0.1	0.2	0.5	6 - 30	0.2	0.5	0.8	30 - 100	0.3	0.8	1.5	100 - 300	0.5	1.2	2.0	300 - 1000	0.8	2.0	3.0	1000 - 3000	1.2	3.0	5.0	3000 PLUS:	2.0	4.0	8.0			COPYRIGHT NOTICE: UNAUTHORISED USE, MANUFACTURE OR REPRODUCTION IN WHOLE OR IN PART IS PROHIBITED. DRAWING, DESIGN AND OTHER DISCLOSURES IS THE INTELLECTUAL PROP- erty of McTronX RESEARCH GROUP. ALL RIGHTS RESERVED	
OVER - TO	±	±	±																																			
0 - 6	0.1	0.2	0.5																																			
6 - 30	0.2	0.5	0.8																																			
30 - 100	0.3	0.8	1.5																																			
100 - 300	0.5	1.2	2.0																																			
300 - 1000	0.8	2.0	3.0																																			
1000 - 3000	1.2	3.0	5.0																																			
3000 PLUS:	2.0	4.0	8.0																																			
ANGLES: 1° (1° 1°) SURFACE FIN: 1.6 3's 12's			CJG	26/06/08		TWINS-001-S																																
APPR.	DRAWN	DATE	DATE	REV.																																		
					DWG. No.																																	

