



ANALYSIS OF LINEAR MOTION SYSTEMS FOR A LARGE SCALE FDM 3D PRINTER

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Reprap is an open source 3D printer that can self-replicate a lot of its own parts, has helped greatly in bringing 3D printing into the price range and homes of hobbyists. Current models are small and not sturdy looking, which is not good for scaling up so that it would be able to do a large quality print. How can a printer be made larger but still print quality prints? The linear system of a normal size Reprap MendelMax would have to be replaced to achieve this. Two linear systems that were relatively inexpensive and easy to acquire, the Makerslide and the Open Rail linear systems were good candidates. Calculating and experimenting with these systems revealed that the Makerslide would only deflect 0.12 mm on a span of 1 meter. Compared to the normally used 8 m from a load of 2.21 Kg, which is roughly the weight of 3 Greg's Tilt Extruder's, for multi-color prints. These numbers are acceptable for model building in an architectural firm for a derivation of a Reprap MendelMax with a print size of 400x400 mm. This information also implies that one can go even bigger than that, but will probably need to upgrade electronics and motors.

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Abstract

3D printers have been around for quite some time in one way or another. Only in recent years with programs such as Reprap, an open source 3D printer that can self replicate a lot of its own parts, has helped greatly in bringing 3D printing into the price range and homes of hobbyists. When constructing one, the current models are small and not sturdy looking, which is not good for scaling up so that it would be able to do a large quality print. This of course does not matter if detail in the print is not important, but if the printer is to be used in an architectural firm for model building, detail is important. How can a printer be made larger but still print quality prints? The linear system of a normal size Reprap MendelMax would have to be replaced to achieve this. Two linear systems that were relatively inexpensive and easy to acquire, the Makerslide and the Open Rail linear systems were good candidates. Calculating and experimenting with these systems revealed that the Makerslide would only deflect 0.12 mm on a span of 1 meter . Compared to the normally used 8 mm steel rod that deflected 12.96 mm , which is a difference of 10.700% . This is from a load of 2.21 Kg , which is roughly the weight of 3 Greg's Tilt Extruder's, for multi color prints. So if only using 1 color and there for 1 extruder the weight would be reduced substantially and the deflection as well. These numbers are acceptable for model building in an architectural firm for a derivation of a Reprap MendelMax with a print size of $400\times 400\text{ mm}$. This information also implies that one can go even bigger than that, but will probably need to upgrade electronics and motors.

Contents

Abstract	ii
Table of Contents	iv
List of Figures	v
List of Tables	vi
1 Introduction	1
1.1 Background	2
1.2 3D printing at home	5
2 Methods & Experiments	5
2.1 Design	5
2.2 Test samples	6
2.2.1 Makerslide	6
2.2.2 OpenRail	7
2.2.3 Stainless steel rod	8
2.3 Experiments	9
2.3.1 Location and conditions	9
2.3.2 Test frame	9
2.4 Studies	10
2.4.1 Statics	10
2.4.2 Strain calculations	12
2.4.3 Natural frequency	13
2.4.4 Strain gauge in general	14
2.4.5 Piezoelectric Accelerometer in general	18
2.5 Instruments and experiments	19
2.5.1 Strain gauges	19
2.5.2 Accelerometers	20
2.5.3 Data acquisition units	21
2.5.4 LabVIEW programs	22
2.6 Requirements	23
3 Results and Discussion	24
3.1 Results	24
3.1.1 Makerslide	24
3.1.2 OpenRail	25
3.1.3 Stainless steel rod	26
3.2 Discussion	27

4	Conclusion	28
4.1	Future work	28
5	Appendix	31
5.1	Makerslide drawing	32
5.2	OpenRail drawing	33
5.3	V-wheel drawing	34
5.4	Monitran accelerometer data sheets	35
5.5	Makerslide hand calculations	36
5.6	OpenRail hand calculations	38
5.7	Steel rod hand calculations	40
5.8	Strain data from Makerslide experiment	42
5.9	Strain data from OpenRail experiment	43
5.10	Strain data from 8 <i>mm</i> rod experiment	44
5.11	Hand calculations with strain data for Makerslide	45
5.12	Hand calculations with strain data for OpenRail	47
5.13	Hand calculations with strain data for steel rod	49
5.14	Solidworks Simulation Pro settings	51
5.15	Test frame drawing	52
5.16	Solidworks Model	54

List of Figures

1	Difference between derivations.	5
2	Cross section of Makerslide	6
3	Cross section of 2 OpenRail on KJN 20x20 mm extrusion	7
4	Cross section of stainless steel rod	8
5	Test Frame	10
6	Simple supported beam with load F applied	10
7	Simple supported beam with uniformly distributed load	11
8	Representation of a Single-Degree-Of-Freedom system	13
9	Strain gauge scematic [1]	15
10	Basic Wheatstone Bridge Circuit [2]	15
11	2 wire quarter-bridge version	16
12	3 wire quarter-bridge version	17
13	Double 3 wire quarter-bridge version	17
14	Placement of strain gauges on beam samples	18
15	How the piezoelectric accelerometer looks and works.	18
16	Straingauge in place	19
17	Look and placement of sensors.	20
18	National Instruments signal processing models.	21
19	LabVIEW user interface for strain experiments	22
20	LabVIEW user interface for natural frequency experiments	23
21	Screen shot from LabVIEW from a Natural frequency experiment	24
22	Screen shot from LabVIEW from a Natural frequency experiment	25
23	Strain gage reading from when someone walking by experiment in progress	27
24	Fixtures and loads	51
25	Mesh set to max	51
26	Split line inserted in the middle of the beams	52
27	Result sample	52
28	Axis movement on new printer	54
29	Solidworks assembly of the modelled parts so far	55

List of Tables

1	Pros and cons of each of the methods described in chapter 1.1	4
2	Relevant numbers for calculations	7
3	Relevant numbers for calculations of the assembly	8
4	Relevant numbers for calculations	8
5	Experiment environment data	9
6	Results of Makerslide experiments and calculations	24
7	Results of Open Rail experiments and calculations	25
8	Results of Stainless steel 8 mm rod experiment and calculations	26

1 Introduction

In the Reprap 3D printer projects [3] the most commonly used type of linear system for all axis in the printers are 8 mm diameter smooth steel rods with some sort of linear bearing that runs along it. In this paper I will be looking at the possibilities of using other than the usual smooth steel rods for a linear system for all axis in a derivation of a MendelMax [4] FDM printer. The main reason being that the usual print size is around 200x200 mm on a normal MendelMax but my derivation will have around 400x400 mm print size. I will be looking at two systems, the so called Makerslide system [5] and the Open Rail system [6]. Both of them use dual V wheels that slide along guided paths, see drawing in appendix 5.3. The Makerslide system is a 20x40 mm aluminium extrusion with notches on top and bottom that act as a guide for dual V wheels that slide along them. The Open Rail system is a rail that you can screw onto an existing aluminium extrusion i.e 20x20 mm as seen in Figure 3, or any size extrusion that you have for that matter. This is then the guide for the dual V wheels to slide along them. I will be comparing the deflection in a 1 meter long sample of the systems while they are subject to a load weighing 2.210 kg as well as looking at the normally used 8 mm steel rod under the same conditions. I will also look at the natural frequencies of all three systems and comparing the results to the natural frequency of the stepper motors. This is to be better informed which system is best suited for a large scale printer that will be able to print models for an architectural firm where detail is preferred and to find out what frequency's will have to be avoided when designing the printer.

The general procedure for this is as follows:

1. Analyse deflection in all 3 beams using traditional beam equations from reference material [7].
2. Perform experiments with strain gauges on all 3 types of beams and compare to no. 1.
3. Make a CAD model of all 3 beams and use FEA method in Solidworks simulation 2013 to see the deflection and compare to no. 1 and no. 2.
4. Perform closed form solutions from reference material [8], to find the natural frequencies of the 3 beams.
5. Perform experiments with accelerometers on all 3 beams and compare to no. 4.
6. Use the CAD models from no. 3 and perform Modal analysis in Solidworks simulation 2013 to get the natural frequency's and compare to no. 4 and no. 5.
7. Try to implement the results into a working derivative of the MendleMax within the given time frame.

1.1 Background

A 3D printer is a machine that you can feed Computer Aided Design (CAD) data to. With the professional machines you can often feed the Standard Tessellation Language (STL) file straight to the printer and its software then slices the data into layers. Sometimes this is performed by a separate program and then fed to the printer. The number of layers determines the resolution of the object. More layer usually equals more resolution, hence a better looking object. These layers define a tool path for the printer in the X and Y axis, it creates objects layer by layer from plastic and/or other material from the bottom up on the Z axis. This is called an additive manufacturing process. More common are traditional subtractive manufacturing processes, where material is cut away from a block to create the desired object.

3D printing is a branch of rapid prototyping methods and is a generalized term over making an object in a "3D printer". There are a few different methods that a 3D printer can use [9] [10] :

- Stereolithography, (SLA)

SLA was invented in 1986 by Charles Hull [11]. The process involves a platform immersed in a vat of liquid photopolymer resin, the platform is placed just below the surface of the resin so that only a thin layer of resin covers it. Then the resin is exposed to ultra-violet light from a low-power, highly focused laser which turns the resin from liquid to solid. The laser draws out the bottom layer of the object being created and when it finishes the platform lowers just so that liquid resin flows over the layer. This process is then repeated until the model is complete. The laser beam can harden the layer from 0.05 mm to 0.15 mm depending on the resolution required for the model [12].

- Selective Laser Sintering, (SLS)

SLS was invented at the University of Texas in 1986 by Dr. Carl Deckard [13]. This process is similar to SLA. With it you can create 3D models from plastic, metal or ceramic powder. The powder is fused together with heat from a carbon dioxide infrared emitting laser. As in the SLA process, we have a platform that is immersed in a thin layer of uncured material. In this case in powder form, the laser heats a thin layer of powder so that it reaches its melting point and fuses together to form the bottom layer of the object being created. When this is done the platform lowers just a bit and a powder recoater system deposits a fresh layer of powder ranging from 0.08 mm to 0.15 mm [14]. Then the process is repeated with every layer fusing it to the previous one until the model is complete. Usually when referring to use of metal powder the term SLM is used, which is short for selective laser melting. It is capable of a layer height of 0.02 mm to 0.1 mm [15].

- Three-Dimensional Printing, (3DP)

3DP was invented in 1989 by Emanuel Sachs, John Haggerty, Michael Cima and Paul Williams of the Massachusetts Institute of Technology (MIT) [16]. This method is similar to SLS except that a multichannel ink-jet head and liquid adhesive supply are used instead of a laser. Starch and cellulose powder are then, in a thin layer, spread across the platform. Then the multichannel ink-jet head sprays a water-based liquid adhesive onto the surface of the powder to bond it in the shape of the bottom layer, then a fresh layer of powder is added and the process repeated until the object is complete. Layer thickness is between 0.08 mm to 0.15 mm [17]

- Polyjet

Polyjet was invented by Hanan Gothait in 1999 [18]. It is somewhat like a ink-jet process when you print ink on paper, but instead of ink the print head deposits photo polymer that is immediately cured with UV bulbs following behind the print head. Then the build platform lowers a bit and the process is repeated until the object is complete. This technology is capable of a layer thickness of 0.016 mm to 0.6 mm [19]

- Fused Deposition Modelling, (FDM)

FDM was developed in 1988 by Scott Crump [20]. FDM uses a different method than all of the above, it uses thermoplastic filament that is fed into a temperature controlled FDM extrusion head. The head is heated so that the filament reaches semi liquid state and is extruded and deposited in ultra thin, precise layers on a platform. Each layer adheres with the previous layer by thermal fusion and the platform lowers just a bit. This is repeated until the object is done. Layer thickness can be between 0.178 mm to 0.33 mm [21].

Table 1: Pros and cons of each of the methods described in chapter 1.1

	Pros	Cons
▶ SLA	<ul style="list-style-type: none"> • Surface finish • Mechanical properties • Resolution • Durable • Choice of materials 	<ul style="list-style-type: none"> • Accuracy • Not fully cured, needs UV oven • Support removal process • Not office friendly
▶ SLS	<ul style="list-style-type: none"> • Strong materials • No support material • Living hinges • Production ready 	<ul style="list-style-type: none"> • Not office friendly • Slow and 2 hour warm-up time • Very expensive • Surface quality • Environmental waste • Time consuming • Run for 30 hours = Cool down for 30 hours
▶ 3DP	<ul style="list-style-type: none"> • Full colour spectrum • Low cost (sys + mat) • No support material • Ease of use 	<ul style="list-style-type: none"> • Accuracy • Rough surface • Only one material • Resolution • Expensive to run • Heavy post processing • Constant running
▶ Poly-Jet	<ul style="list-style-type: none"> • High resolution • High accuracy • Smooth surfacing • High build speed • Office friendly • Variety of materials • Small to big size printers 	<ul style="list-style-type: none"> • Sun may alter part • Expensive material • Post processing • UV-cure • Heavy post processing • Superglue dip
▶ FDM	<ul style="list-style-type: none"> • Functional performance • Real thermoplastics • Strong, tough and durable • Stable over time • Economical • Print working assemblies • Hands off support removal • Office friendly 	<ul style="list-style-type: none"> • Surface finish • Longer build times • Post processing • Constant running

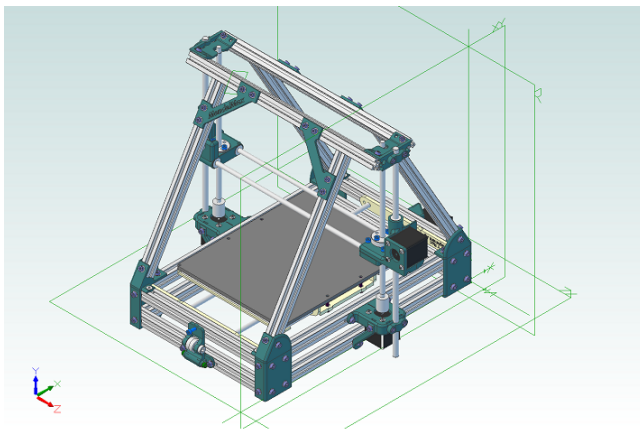
1.2 3D printing at home

With technological advances over the years the cost of 3D printing has been lowered substantially. This has had the affect that people at home have been able to use the technology and for projects like RepRap [3] to thrive. RepRap is an open source project started by Adrian Bowyer of Bath University in 2005 which was designed around the idea of creating a low cost home 3D printer that could replicate a large proportion of its own parts. Meaning if you had one already, you could print on it parts for another machine and make another one for a relatively low cost.

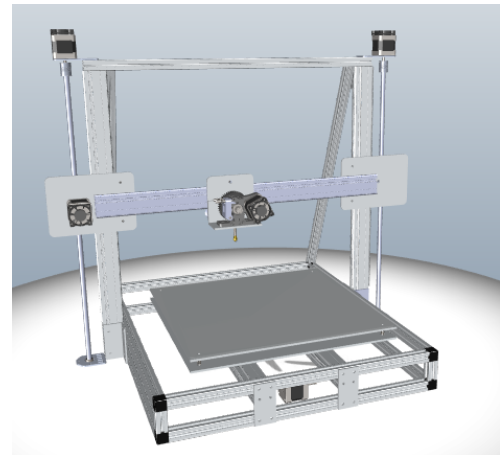
2 Methods & Experiments

2.1 Design

At first the idea about doing the same A type structure as seen in Figure 1a on the Mendle-Max came up, but if the Z axis is properly fastened to the base there is no need for the 45° braces for structural support as seen in Figure 1b.



(a) RepRap MendelMax 1.5 [22]



(b) New derivation

Figure 1: Difference between derivations.

A decision was made to have them only on the back side of the printer, so that the electronics could be mounted to them and a filament spool could be hung between them.

A conclusion was reached and the optimal design solution for the movement of the axis on the printer is to have them independent from each other. So that if there is a problem with one of them it would not affect the other two axis. This makes fine tuning and troubleshooting less complicated. To accomplish this the print bed will be designed to move in the Y direction, the print head will move in the X direction and then the whole X axis will be moved in the Z direction, this can be seen in Figure 28 in appendix 5.16, [23].

The printer size will be around $500 \times 500 \times 600 \text{ mm}$ so that the print area will be $400 \times 400 \times 400 \text{ mm}$. This is because there has to be room on both sides of the print bed and above it to fit

the extruder plus linear system.

2.2 Test samples

Both the Makerslide and OpenRail aluminium extrusions are open source linear bearing systems designed for the hobbyist, to use in his projects. They are entry level linear systems that are very cheap compared to professional ones. 20 dollars for 1 *meter* of Makerslide is cheap compared to a professional magnetic bearing system, that are designed and custom made for your application. A 8 *mm* stainless steel rod [3] was also tested as it is the the most used linear system in Reprap 3D printers today.

2.2.1 Makerslide

Makerslide as seen in Figure 2 is an aluminium extrusion made from 6105-T5 aluminium, drawing from manufacturer is in appendix 5.1.

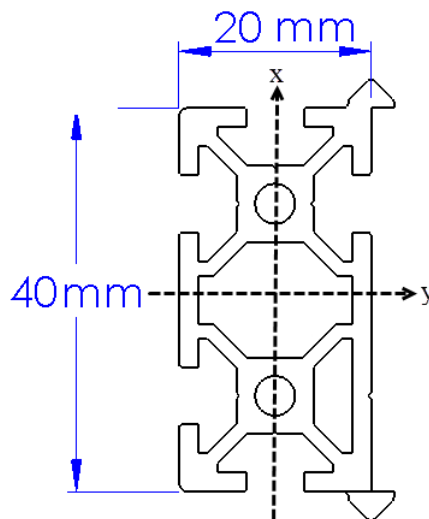


Figure 2: Cross section of Makerslide

Data in Table 2 is from Solid Works 2013. The Modulus of elasticity is from the internal material library and the moment of inertia along with the cross sectional area by using the tools/section property's command. This was then confirmed on the manufacturers web page at ¹, where further information on pricing and availability can also be found. Modulus of elasticity for both aluminium 6105-T5 and 6061-T6 which is in the OpenRail beam is 70 *Giga pascals* [GPa] [24].

¹<http://store.amberspyglass.co.uk/makerslide.html>

Table 2: Relevant numbers for calculations

Aluminium 6105-T5	Letter	Number	Unit
Modulus of elasticity	E	69	GPa
Moment of inertia x axis	I_x	16062.25	mm^4
Moment of inertia y axis	I_y	61012.91	mm^4
Cross sectional area	A	352.54	mm^2

2.2.2 OpenRail

OpenRail is a linear system designed to fit almost any aluminium extrusion that is out there, including KJN, Bosch Rexroth, 80/20 and Mitsumi. The idea is that the OpenRail on your extrusion and make your own linear system with it. You can have as long a distance as you want between the rails or you can put one on top another as seen in Figure 3 and have the distance as small as 20 mm. This is what will be used in the printer. The aluminium extrusion that I am using is a 6 mm slot profile from KJN Aluminium Profiles. They are made from 6061 aluminium. OpenRail is made from 6061-T6 aluminium and has a black type III hard anodized coat on it to make it more resistant to wear. A drawing from the manufacturer of OpenRail is available in appendix 5.2.

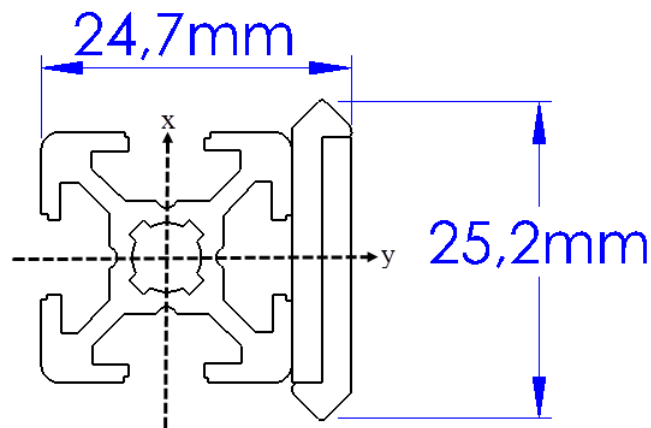


Figure 3: Cross section of 2 OpenRail on KJN 20x20 mm extrusion

Data in Table 3 is from an assembly in Solid Works 2013 using same methods as described for Table 2 and conformed on the manufacturer's website ², where further information on pricing and availability can also be found. Information regarding the KJN aluminium extrusion is available at their website ³. Even though they are made from 6060 aluminium the material properties are almost identical [24], so using modulus of elasticity as $E=70 GPa$ for the whole assembly is acceptable.

²<http://openbuildspartstore.com/openrail/>

³<http://www.aluminium-profile.co.uk/acatalog/20x20-Aluminium-Profile--KJN992888.html#SID=31>

Table 3: Relevant numbers for calculations of the assembly

Al 6061-T6 and 6060	Letter	Number	Unit
Modulus of elasticity	E	70	GPa
Moment of inertia x axis	I_x	17286.36	mm^4
Moment of inertia y axis	I_y	12012.02	mm^4
Cross sectional area	A	278.15	mm^2

2.2.3 Stainless steel rod

The most used linear system in Reprap 3D printers is an 8 mm stainless steel rod and linear bearings that travel along them. The rod that was chosen made from 304 H9 Stainless steel, below in Figure 4 is a cross section view of it.

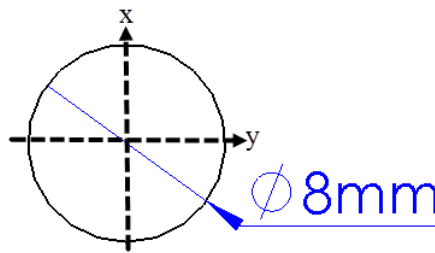


Figure 4: Cross section of stainless steel rod

Since 304 H9 steel is well known the material data from Solid Works was used to get the modulus of elasticity for data in Table 4 , moment of inertia and area information is also from there.

Table 4: Relevant numbers for calculations

304 H9 Stainless steel	Letter	Number	Unit
Modulus of elasticity	E	190	GPa
Moment of inertia x axis	I_x	201.06	mm^4
Moment of inertia y axis	I_y	201.06	mm^4
Cross sectional area	A	50.27	mm^2

2.3 Experiments

2.3.1 Location and conditions

Experiments were all conducted in the civil engineering lab at Reykjavik University. Parts that were to be tested and the frame were all kept inside the lab over night to allow everything to settle to the room temperature. The tests were conducted on four different occasions over the summer, running each test several times each session. The first test was with Indriði Sævar Ríkhartsson from the Science and Engineering department at Reykjavik University when he showed how the LabVIEW programs in the PC worked and how to collect data. The second test was conducted to get the temperature and humidity in the lab at the time of testing. The third test was to get new data because the data from test 2 was far off from data from test 1. The fourth test was to confirm data from tests 1 and 3. The fifth test was done to get the natural frequency for the 8 mm rod. In Table 5 it is shown what tests on which beam at what date and the condition of the room. Beams are: MS for Makerslide, OR for OpenRail and SR for stainless steel rod. Tests being performed are: 1 for deflection test with strain gauges and 2 is for natural frequency test with accelerometers.

Table 5: Experiment environment data

Being tested	Test performed	Date	Temperature	Humidity
MS	1, 2	14.6.2013	x	x
OR, MS, SR	1, 2	28.6.2013	24 °C	55 %
OR, MS, SR	1, 2	26.7.2013	26 °C	53 %
OR, MS, SR	1, 2	7.8.2013	25 °C	55 %
SR	2	25.8.2013	26 °C	55 %

2.3.2 Test frame

A frame was built as seen in Figure 5, made out of aluminium extrusions to conduct the tests on the different linear systems. Detailed drawings of the frame are in appendix 5.15 The sample that was to be tested was placed on the frame and fastened in place so that it would resemble a simple supported beam.

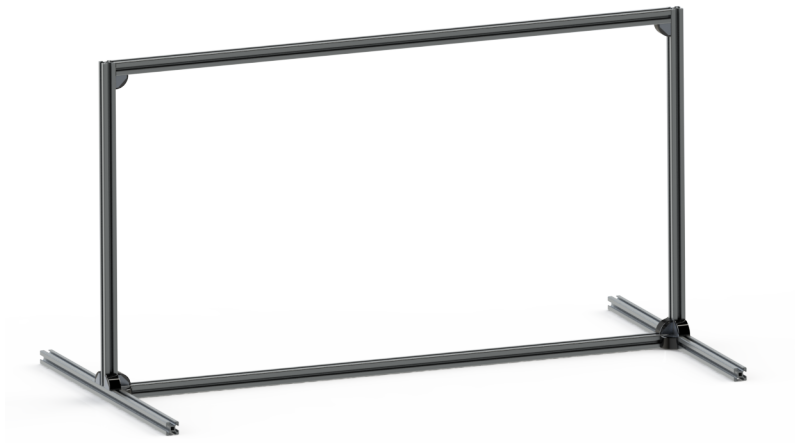


Figure 5: Test Frame

2.4 Studies

2.4.1 Statics

The beams being tested when fastened to the frame resemble a simple supported beam. In such a case the following formulas apply, where F is applied at $\frac{L}{2}$ [7].

Resultant forces are given by

$$R_A = R_B = \frac{F}{2} \quad (1)$$

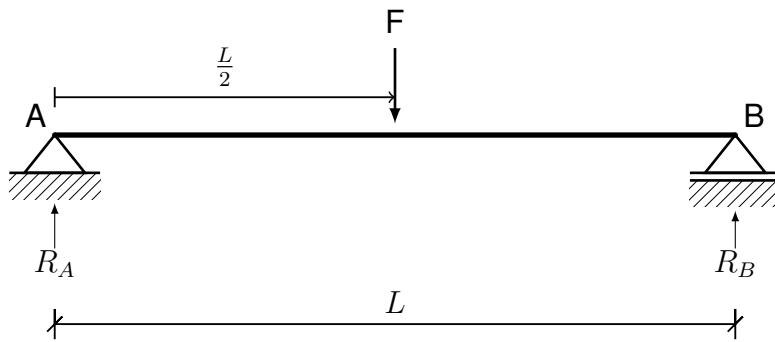


Figure 6: Simple supported beam with load F applied

Where R_A and R_B are the resultant force in newtons $[N]$ and F is the load on the beam in newtons $[N]$

Maximum moment is in the middle of the beam at $\frac{L}{2}$ and is given by

$$M_{\max \text{ Beam}} = \frac{FL}{4} \quad (2)$$

Where $M_{\max \text{ Beam}}$ is the maximum moment at $\frac{L}{2}$ newton meters $[N \cdot m]$, F is the load on the beam and L is the length of the beam in meters $[m]$

Maximum deflection of the beam occurs also at the middle of it at $\frac{L}{2}$, that is given by

$$\delta_{\max \text{ Beam}} = \frac{FL^3}{48EI} \quad (3)$$

Where $\delta_{\max \text{ Beam}}$ is the deflection of the beam in meters $[m]$, F is the load on the beam in newtons $[N]$, L is the length of the beam in meters $[m]$, E is the Modulus of Elasticity in newtons per meter squared $[\frac{N}{m^2}]$ and I is the Moment of Inertia of the cross section of the beam in quadruple meters $[m^4]$

The above equations only account for the load F being placed on the beam, to include the beam as well we have to model it as a beam with uniformly distributed load as seen below in Figure 7 where the weight of the beam works as F_q .

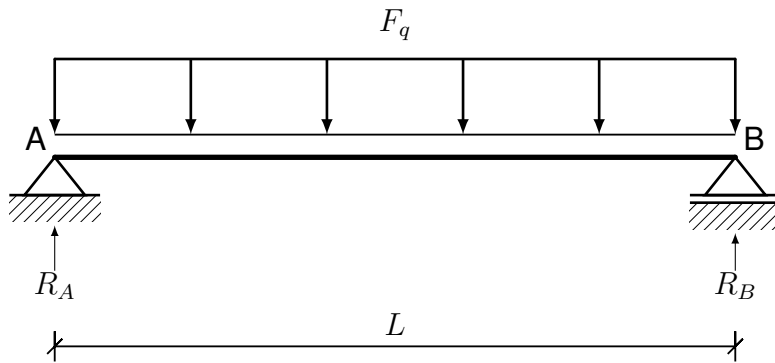


Figure 7: Simple supported beam with uniformly distributed load

To find the force F_q , which is in newtons $[N]$ per meter $[m]$, we need to take the weight of the beam W_{Beam} in kilograms $[Kg]$ and multiply it with the gravitational constant g in $[\frac{m}{s^2}]$ and finally divide that with the length of the beam in meters $[m]$

$$F_q = \frac{F_{\text{Beam}}g}{L_{\text{Beam}}} \quad (4)$$

To get the resultant forces we use

$$R_A = R_B = \frac{F_q}{2} \quad (5)$$

Where R_A and R_B are the resultant force in newtons $[N]$ and F_q is the uniform load on the beam in newtons per meter $[\frac{N}{m}]$

Maximum moment is in the middle of the beam at $\frac{L}{2}$ and is given by

$$M_{\max \text{ Unif}} = \frac{F_q L^2}{8} \quad (6)$$

Where $M_{\max \text{ Unif}}$ is the maximum moment at $\frac{L}{2}$ newton meters $[Nm]$, F_q is the uniform load on the beam in newtons per meter $[\frac{N}{m}]$ and L is the length of the beam in meters squared $[m^2]$

To find the maximum deflection of the beam, which also occurs at the middle of it at $\frac{L}{2}$ we use

$$\delta_{\max \text{ Unif}} = \frac{5F_q L^4}{384EI} \quad (7)$$

Where $\delta_{\max \text{ Unif}}$ is the deflection of the beam in meters $[m]$, F_q is the uniform load on the beam in newtons per meter $[\frac{N}{m}]$, L is the length of the beam in meters $[m]$, E is the Modulus of Elasticity in newtons per meter squared $[\frac{N}{m^2}]$ and I is the Moment of Inertia of the cross section of the beam in quadruple meters $[m^4]$

In order to get the total deflection and moment for the beam we must combine Equations (3) and (7) to form 8 to get the total deflection of the beam

$$\delta_{\max} = \frac{FL^3}{48EI} + \frac{5F_q L^4}{384EI} \rightarrow \frac{L^3}{EI} \left(\frac{5F_q}{384} + \frac{F}{48} \right) \quad (8)$$

Where δ_{\max} is the deflection of the beam in meters $[m]$, F_q is the uniform load on the beam in newtons per meter $[\frac{N}{m}]$, F is the force of the load in newtons $[N]$, L is the length of the beam in meters $[m]$, E is the Modulus of Elasticity in newtons per meter squared $[\frac{N}{m^2}]$ and I is the Moment of Inertia of the cross section of the beam in quadruple meters $[m^4]$

Likewise in order to get the total moment for the beam, we need to combine Equations (2) with (6) and form Equation (9).

$$M_{\max} = \frac{FL}{4} + \frac{F_q L^2}{8} \quad (9)$$

2.4.2 Strain calculations

In order to be able to use the data produced by the strain gauges, we need a little more. The LabVIEW program gives us data in strain ε which is unit less and plots that with unit time in seconds $[s]$. To be able to use Equation 7 and convert the strain data to deflection data, we first need to define stress σ

$$\sigma = \varepsilon E \quad (10)$$

Where σ is the stress in Pascals $[Pa]$, ε is unitless and E is the Modulus of Elasticity in newtons per meter squared $[\frac{N}{m^2}]$.

That we can relate to the following moment equation and get our new maximum moment at $L/2$

$$M_{\max \text{ DATA}} = \frac{\sigma I}{\frac{L}{2}} \quad (11)$$

$M_{\max \text{DATA}}$ is the moment in newton $[N]$ meters $[m]$, σ is the stress in Pascals $[Pa]$, c is the distance in meters $[m]$ from the neutral axis to the top of the beam and I is the Moment of Inertia of the cross section of the beam in quadruple meters $[m^4]$.

Since the weight of the beam has not changed, we can subtract Equation 6 from our new max moment Equation 11 and we are left with the moment caused by the load on the beam.

$$M_{\max \text{ Beam}} = M_{\max \text{DATA}} - M_{\max \text{ Unif}} \quad (12)$$

Then we use Equation 2 and solve for our new F_{DATA}

$$F_{\text{DATA}} = \frac{4M_{\max \text{DATA}}}{L} \quad (13)$$

Now we can plug that in to Equation 8 and get our displacement according to the data.

$$\delta_{\max \text{ DATA}} = \frac{L^3}{EI} \left(\frac{5F_q}{384} + \frac{F_{\text{DATA}}}{48} \right) \quad (14)$$

2.4.3 Natural frequency

Beams have infinite degrees-of-freedom, however beams may be modeled as single-degree-of-freedom (SDOF) systems as seen in Figure 8 to make hand calculations easier. This is the form being used in this report for approximating the first natural frequency of the beams being tested [8].

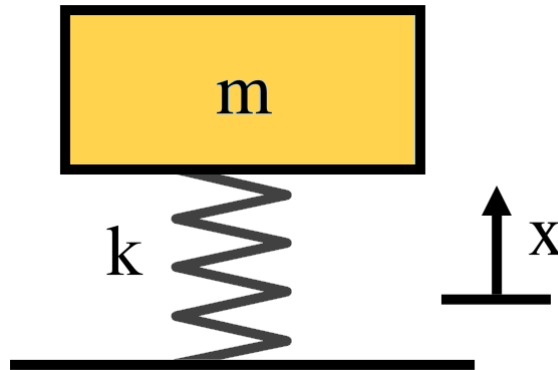


Figure 8: Representation of a Single-Degree-Of-Freedom system

Still dealing with a simple supported beam as shown in Figure 6 the following equations can be applied to find the first natural frequency of the beams being tested [8]. These formulas neglect the weight of the beam itself but are still a good indicator of what range should be considered a danger zone. We do know that the real frequency will be lower than the one from these formulas because the mass of the beam will add to the mass in Formula (15) and that will result in a lower ω

Natural frequency of the SDOF system in radians per second is given by

$$\omega = \sqrt{\frac{K_{eq}}{m}} \quad (15)$$

Where ω is the natural frequency in radians per second $[rad/sec]$, K_{eq} is the spring equivalent constant in newtons per meter $[\frac{N}{m}]$ and m is the mass in kilograms $[Kg]$.

K_{eq} is the spring equivalent constant, also known as stiffness is in newton per meter for a simple supported beam is given by

$$K_{eq} = \frac{48EI}{L^3} \quad (16)$$

Where K_{eq} is the spring equivalent constant in newtons per meter $\frac{N}{m}$, E is the Modulus of Elasticity in newtons per meter squared $[\frac{N}{m^2}]$, I is the Moment of Inertia of the cross section of the beam in quadruple meters $[m^4]$ and L is the length of the beam in meters $[m]$.

The natural frequency in cycles per second is given by

$$f = \frac{\omega}{2\pi} \quad (17)$$

Where f is the natural frequency in $[cps]$ or more commonly $[Hz]$, ω is the natural frequency in radians per second $[rad/sec]$

The period from peak to peak of the SDOF system in seconds is

$$T = \frac{1}{f} \quad (18)$$

Where T is the period in seconds $[s]$ and f is the natural frequency in hertz $[Hz]$. To get the natural frequency of a beam with load and include the weight of the beam we can use another formula that involves the deflection of a beam. If we use Formula (7) where we have combined both F and F_q , we get the total deflection

$$f_n = \frac{\sqrt{\frac{g}{\delta_{max}}}}{2\pi} \quad (19)$$

2.4.4 Strain gauge in general

The strain gauge is a common device for electrical measurement of static deformation. They rely on a proportional linear difference of resistance $[\Delta R]$ from difference in gauge length $[\Delta L]$ along its longer axis. This is referred to as Gauge Factor $[GF]$ and is usually in the

close vicinity of 2. Gauge Factor is expressed in equation form as

$$GF = \frac{\Delta R/R}{\Delta L/L} \quad (20)$$

and we define strain $[\varepsilon]$ as

$$\frac{\Delta L}{L} = \varepsilon \quad (21)$$

The strain gauges used as seen in Figure 9 are composed of a measuring grid that is formed by etching Constantan foil, which is then completely sealed in a carrier medium composed of polyimide film [25]. Strain can be positive (tensile) or negative (compression). Strain is unitless and usually very small and is often expressed as micro strain $[\mu\varepsilon]$, which is $\varepsilon \times 10^{-6}$

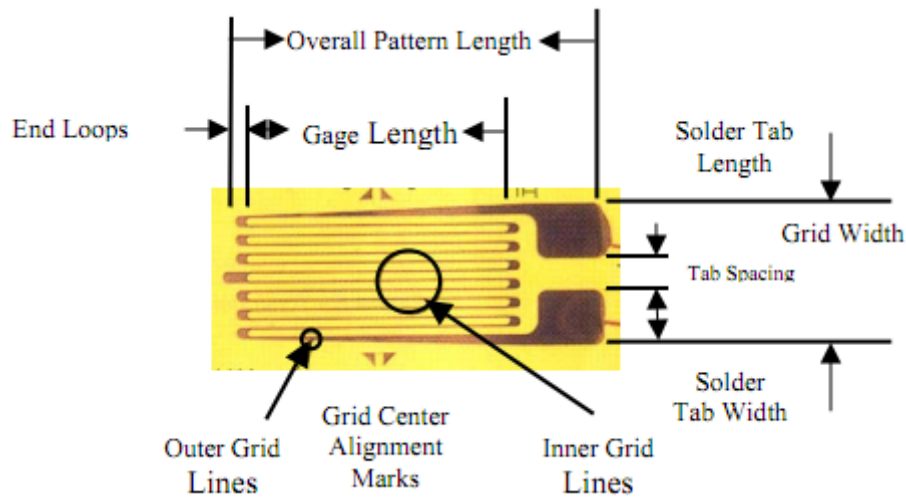


Figure 9: Strain gauge scematic [1]

Wheatstone Bridge has become the sensing circuit of choice in many commercially available strain gauge instrumentation. This is mainly because of it's inherit ability to:

- Detect the small change in resistance in the strain gauge.
- Produce a zero output voltage when the part being tested is at rest.
- Provide for compensation of temperature-induced resistance changes in the strain gauge circuit.

The Wheatstone Bridge in it's simplest form as seen in Figure 10 is the electrical equivalent of two parallel voltage divider circuits, R_1 and R_4 , which compose one voltage divider circuit and R_2 and R_3 which com-

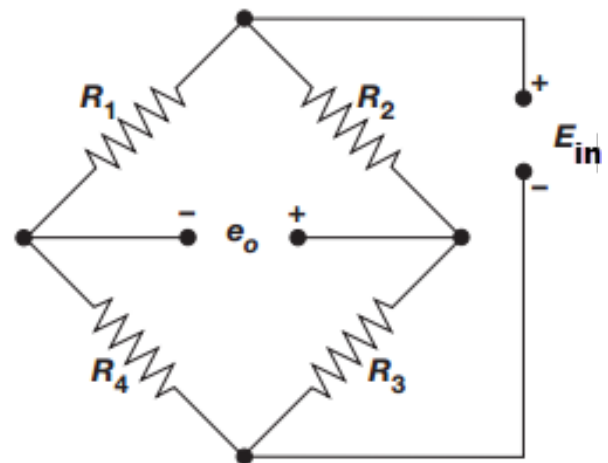


Figure 10: Basic Wheatstone Bridge Circuit [2]

pose the second one with $R_1 = R_2 = R_3 = R_4$. The output of a Wheatstone bridge is measured between the middle nodes of the two voltage dividers e_0 seen in Figure 10, when there is no force being applied then the bridge is in balance and e_0 should read 0 volts. This we can see in Formula 22, if all resistors are the same size the output voltage is 0.

$$e_0 = \left[\frac{R_3}{R_3 + R_2} - \frac{R_4}{R_1 + R_4} \right] E_{in} \quad (22)$$

Where e_0 is the bridge output in volts [V], R_{1-4} are the bridges resistor values in ohms [Ω] and $[E_{in}]$ is the input voltage in volts [V].

Then if a physical phenomena, such as a change in strain applied to a specimen or a temperature shift, changes, the resistance of the sensing elements in the Wheatstone bridge will result in an unbalanced bridge and e_0 will show the volt difference. When using a configuration of the bridge called a 2 wire quarter-bridge as seen in Figure 11, that is when you replace R_1 with a strain gauge in to the bridge that is exactly the same resistance and connect the two lead wires having negligible resistance. Then the bridge should remain in balance.

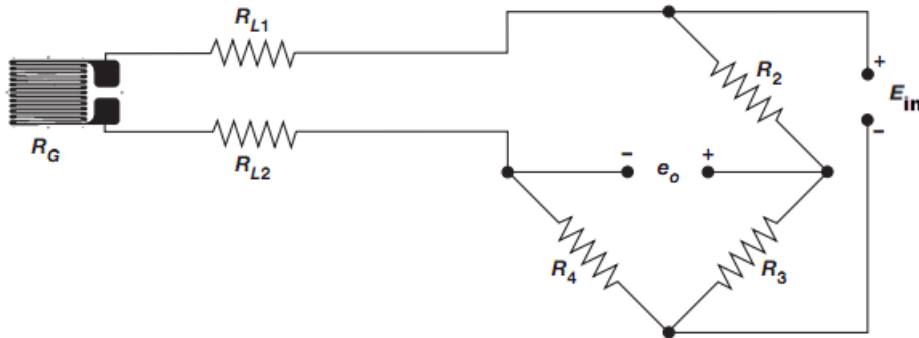


Figure 11: 2 wire quarter-bridge version

In practice though, lead wires have resistance as shown with Formula 23 and as we can see in Figure 11 above that if we take great care to put a strain gauge with exactly the same resistance as the other ones in the bridge that the lead wires resistance will add to that of the strain gauge and will make the bridge become unbalanced and a non zero voltage at e_0 . Add to that that if the temperature changes then the resistance in the wire will increase. When dealing with micro strain, lead wire resistance can have a substantial effect on the results. To counteract this problem we can use a 3 wire quarter-bridge as seen in Figure 12

The negative output bridge corner is electrically moved from the top of R_4 to the bottom strain gauge at the end of R_{L3} . In this configuration, lead wire R_{L1} and strain gauge R_{G1} make up one arm of the bridge and R_{L2} with resistor R_4 the other arm. Making a balanced bridge as long as the lead wires are same size, length and type, for then they will have the same resistance. Additionally, because only one lead wire is in series with the strain gauge, lead wire desensitization is reduced by 50% compared to the two wire configuration. Wire

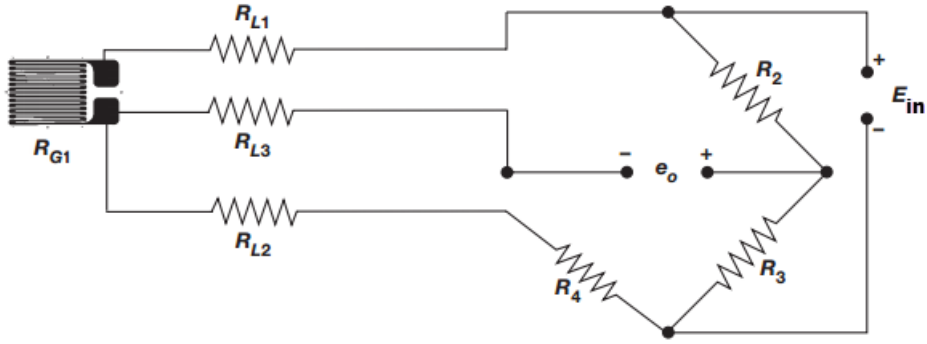


Figure 12: 3 wire quarter-bridge version

R_{L3} is only used as a voltage sensing wire and is not in series with any of the bridge arms. Therefore it has no effect on the balance of the bridge. To get an even better result from strain gauge measurements, a double 3 wire quarter-bridge was implemented as seen in Figure 13,

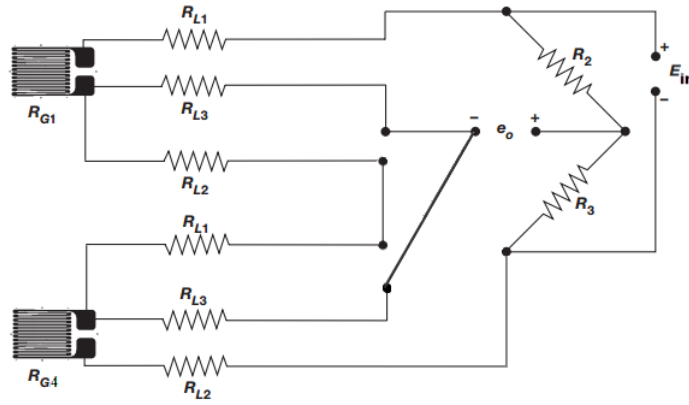


Figure 13: Double 3 wire quarter-bridge version

This configuration is the same as the single gauge setup and there for has the same advantages, but by placing one gauge on top of the specimen and the other underneath, like shown in Figure 14 you are able to get a more accurate data. Because of the way the gauges are placed, one reading is showing positive strain and the other should show very similar results only negative strain. This supports that the gauges are correct and giving reliable data [26]

Here we can see that lead wires do and can have a substantial effect on the data collected if

$$R = \frac{\rho L}{A} \quad (23)$$

Where R is the wire resistance in ohms $[\Omega]$, ρ is the resistivity of the wire in ohms per meter $[\Omega m]$, L is the length of wire in meters $[m]$ and A is the cross-sectional area of the wire in meters squared $[m^2]$

Then if used is a 1 meter long normal 24 gauge copper wire and it's 25° in the room, then it's resistivity is $1.68 \cdot 10^{-8} [\Omega m]$ and it's cross sectional area is $0.205 [mm^2]$. Now plug this into Equation 23 and we get 0.000167Ω as the wires resistance. On the other hand if

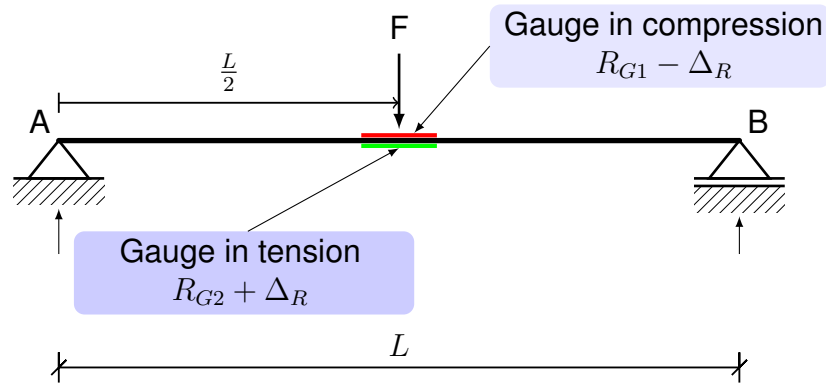


Figure 14: Placement of strain gauges on beam samples

the strain number from the Makerslide experiment calculations in appendix 5.11 is taken and substituted the into Equation 20 and solved for ΔR , it is 0.003323Ω . This is the correct value but if we take into account the wire resistance and add it to this value we get 0.00349Ω as ΔR . This 1 meter long wire produces an error of 4.8 %.

2.4.5 Piezoelectric Accelerometer in general

Accelerometers are sensors for measuring vibrations. A piezoelectric accelerometer consists of a piezoelectric crystal that is held in compression by a known mass which gets periodically compressed when the accelerometer is subjected to gravitational forces ('g' forces). This makes the crystal emit a charge. This minute signal of only a few pico-coulombs [pC] is then conditioned and amplified by its internally or externally mounted charge amplifier that produces useful voltage for analysis [27].

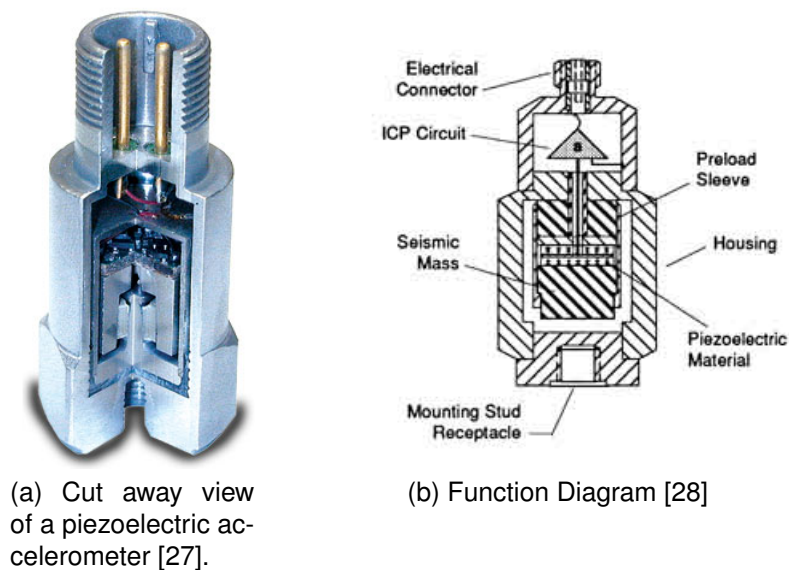


Figure 15: How the piezoelectric accelerometer looks and works.

2.5 Instruments and experiments

2.5.1 Strain gauges

Strain gauges, $120\ \Omega$, model no SG=6/120-LY11 from Omega engineering inc [29]. Each sample of beam that was to be tested was prepared the same way. The gauge itself was measured and then the middle of the beam was found. The midpoint was then marked and 2 cm were measured in each direction from it. The strain gauge was to be fastened in that 4 cm region. It is crucial that the strain gauge is well bonded to the specimens being tested so the region was first sanded with 120 grit sandpaper followed by 320 grit paper. This gave a very smooth surface, which was then cleaned with alcohol to remove any dust or grease left over. Then putting a drop of special super glue followed by the strain gauge and holding it in place for 1 minute.

A small contact pad was then glued in place just next to the strain gauge which is where the wires were soldered from the gauge to 1 meter long extension wires. This was done to not accidentally damage the wires from the gauge. A glue gun was also used to fix the extension wires to the beams just as an extra precaution not to damage the connection of the wires.

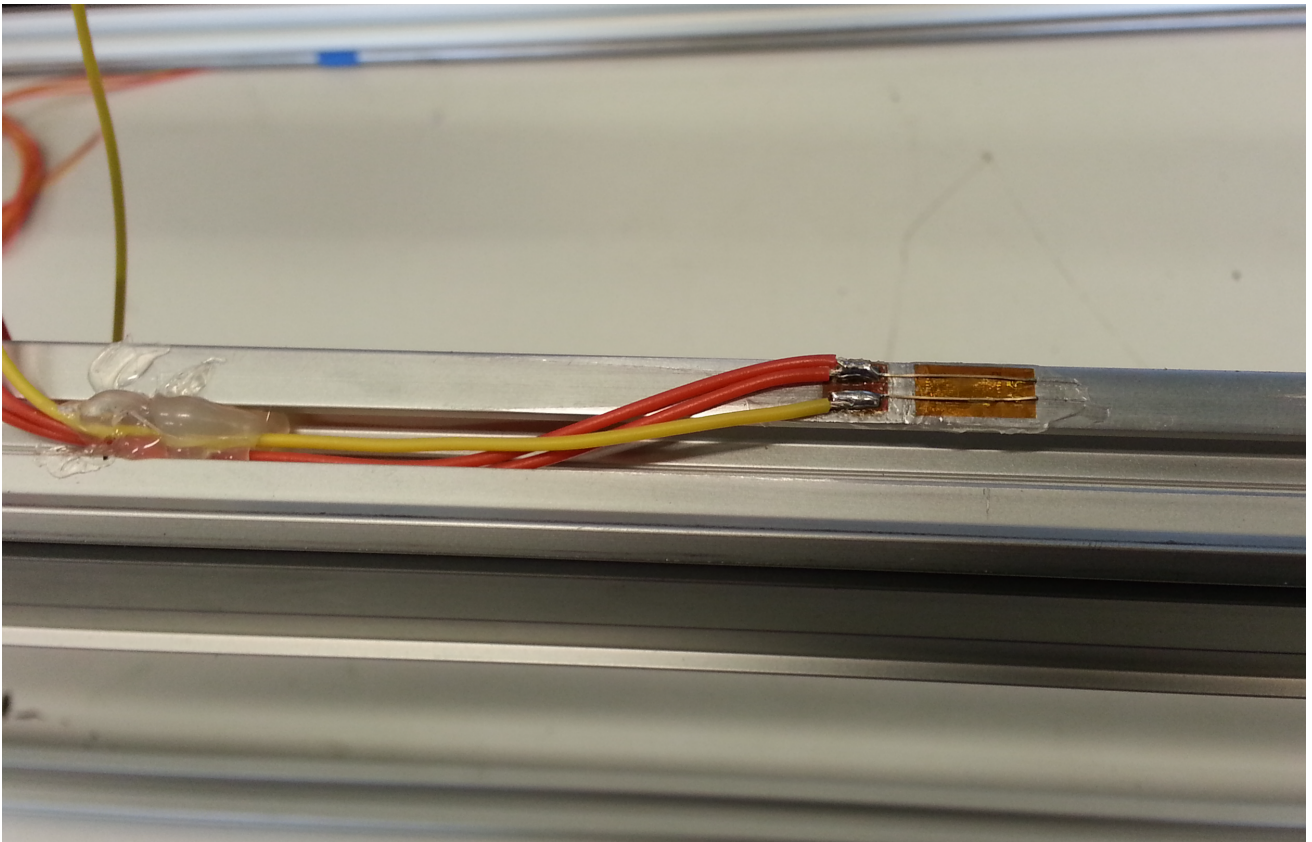
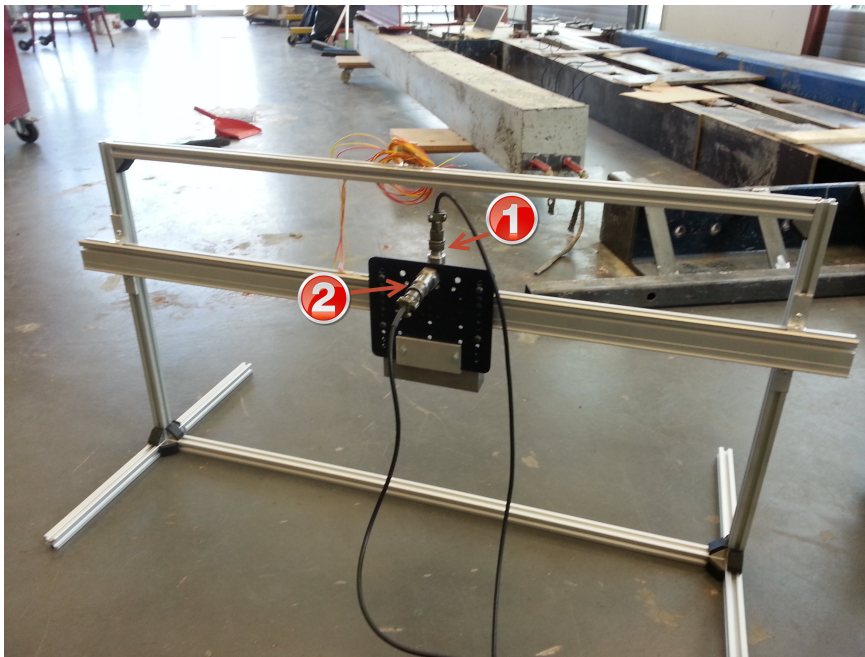


Figure 16: Strain gauge in place

2.5.2 Accelerometers

Accelerometers from Monitran were used to find the natural frequencies of both beams being tested. Data sheets for them are found in the appendix 5.4. There is a M6 hole on the bottom of the sensor so that it can be screwed tightly onto whatever is being tested. A bolt was fixed directly on top of the beams, midspan and the sensor screwed on it. This sensor would detect motion in the up and down directions, no.1 in Figure 17a. Another one was bolted to the plate which moves front and back along the beams. This sensor would detect motion from side to side, no. 2 in Figure 17a.



(a) Placement of accelerometers.

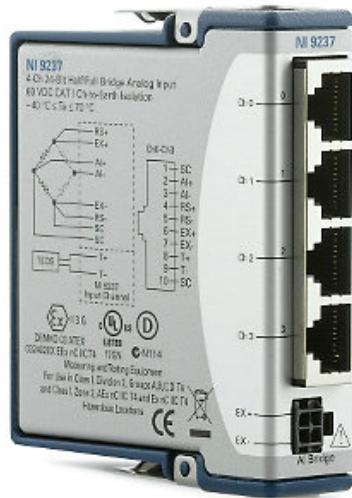


(b) Monitran accelerometer used.

Figure 17: Look and placement of sensors.

2.5.3 Data acquisition units

Data acquisition units (DAQ) from National Instruments (NI) were used to record signals from all sensors during the experiments. A modular carrier NI USB-9162 from NI was connected to a PC and depending on which test was being performed, either the signal module NI 9233 as seen in Figure 18a when doing vibration testing or the bridge module NI 9237 as seen in Figure 18b, when doing deflection testing. Two programs were used to analyse the data from the sensors, both of them were programmed in LabVIEW and have been used before by students and faculty on similar studies.



(a) NI 9237 Bridge module [30].



(b) NI 9233 dynamic signal acquisition module [31].

Figure 18: National Instruments signal processing models.

2.5.4 LabVIEW programs

Explanation of the LabVIEW interface for strain experiments with strain gauges.

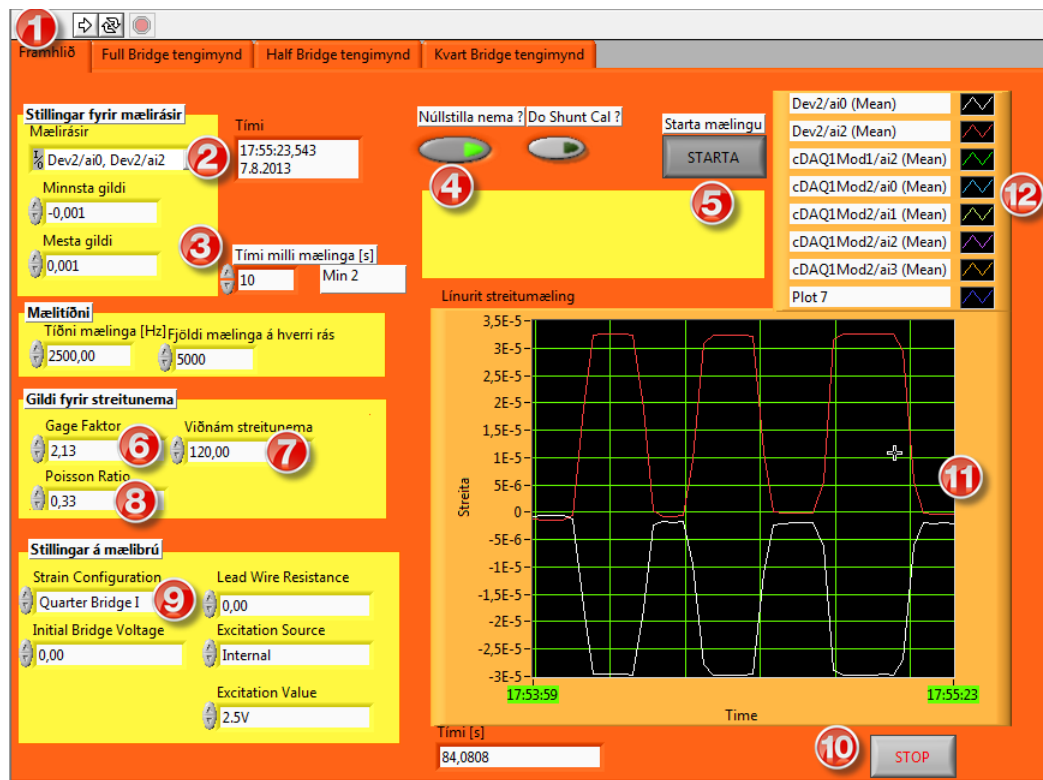


Figure 19: LabVIEW user interface for strain experiments

1. On/Off and reset, 2. Sensor input selection, 3. Time between taken measurements, 4. Reset strain gauge, 5. Start measurement, 6. Gauge factor, 7. Strain gauge resistance, 8. Poisson ratio, 9. Bridge type selection, 10. Stop measurement, 11. Plot area, 12. Plot legend

Explanation of the LabVIEW interface for the natural frequencies experiments with accelerometers.

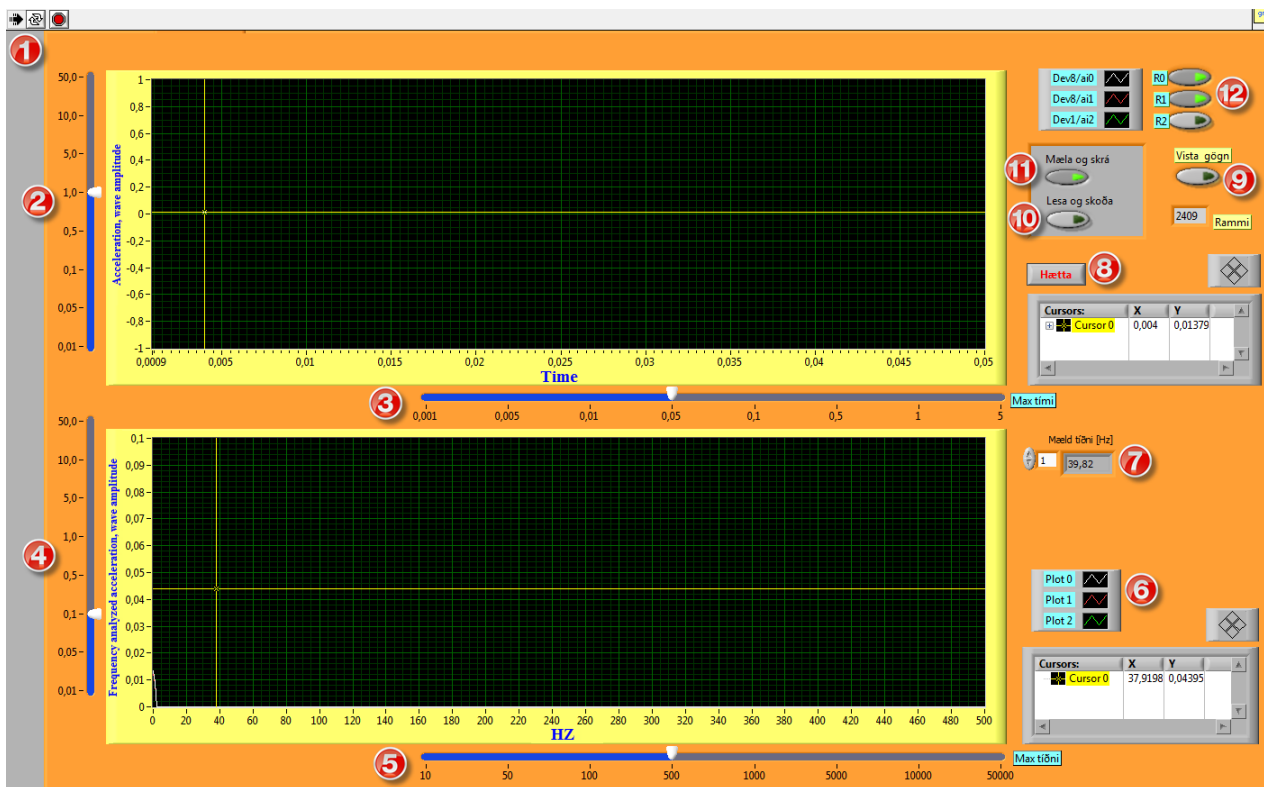


Figure 20: LabVIEW user interface for natural frequency experiments

1. On/Off and reset, 2. Scaling for y-axis, 3. Scaling for x-axis, 4. Scaling for y-axis, 5. Scaling for x-axis, 6. Plot legend, 7. Read out for cross-hair on plot, 8. Quit current data collecting, 9. Save data in real time, 10. Observe data, 11. Measure and log data, 12. Sensor input selection

2.6 Requirements

The minimum requirements for a 1 meter long linear system to be used in a large scale 3D printer is that the deflection of the system with a force of around 20 newtons N , which is the approximate weight of 3 Greg's Tilt Extruder's is less than 0.33 mm which is the upper layer height for a FDM printer, see Section 1.1 . In the case of both systems being tested, deflect less than this number, the one with the less deflection is the one that will be used. This result will be reached by using either the Makerslide or OpenRail systems.

3 Results and Discussion

3.1 Results

The results for each beam being tested are presented in Tables 6, 7, 8. The load was made from scrap material from the RU workshop, which was 21.68 N and therefore 7.7% more than is stated in Section 2.6, this is acceptable because the weight of extruder's that are available to buy range from 3 N up to 7 N . A quick explanation, δ_{max} stands for the maximum deflection that the beam will bend from it's rest position in millimetres $[mm]$ from the applied load of 21.68 [N] , f_n stands for the natural frequency for the given beam in Hertz $[Hz]$. There are shown calculated results from using standard structural references [8], [7]. In the first column titled "Studies", these calculations are available in appendix 5.5, 5.6 and 5.7. The second column titled "Strain exp." is where data from the experiments with strain gauges was used and calculated from them these results, go to appendix 5.11, 5.12 and 5.13. Third and final column, marked "Simulation" are the simulated results from SolidWorks Simulation Pro 2013. Settings and information on how the simulations were conducted can be found in appendix 5.14.

3.1.1 Makerslide

Results for the Makerslide linear system were the following. To see complete calculations see appendix 5.5 for Studies and 5.11 for Strain exp. Solidworks simulation settings can be found in appendix 5.14

Table 6: Results of Makerslide experiments and calculations

Makerslide			
	Studies	Strain exp.	Simulation
δ_{max}	0.12 mm	0.11 mm	0.15 mm
f_n	44.05 Hz	47 Hz	83.26 Hz

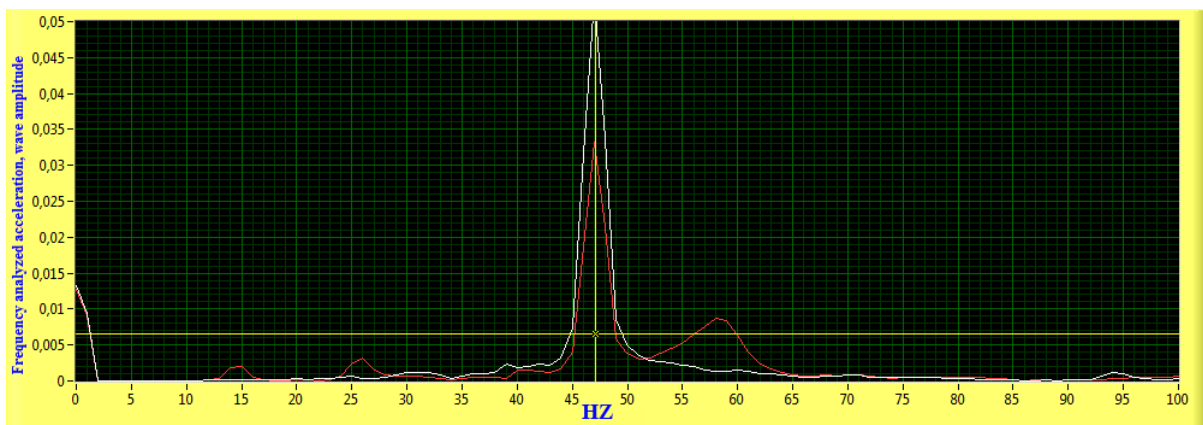


Figure 21: Screen shot from LabVIEW from a Natural frequency experiment

By my reading, the natural frequency of the Makerslide beam is very close to 47 Hz in Figure 21. Compared to the Studies column in Table 6 we have a increase of 6.7% and a 89% increase from Simulation. Similarly if we compare the deflection we get a 8,3% lower value in the experiment and a 25% increase in the simulation. The only thing that stands out is the unexpected high natural frequency from Solidworks Simulation, 89% is quiet a bit compared to the deflection from the same simulation is only off by 25%.

The high natural frequency from Solidworks Simulation was concerning, so another experiment was done. This time the beam was clamped on to a giant structural test machine in the Civil engineering lab in RU, which in the context of this experiment would be considered a rigid structure. By doing this any doubt of the test frame that was built to hold the specimens while conduct the previous experiments was interfering with earlier results. Results from this new experiment agreed with previous ones and therefore the conclusion was reached that the other experiments were valid and the reason for the high natural frequency Solidworks Simulation was giving are still undetermined. More investigation into that would probably tell us why that is, but first guess would be that Solidworks is not handling the end fastening connections properly.

3.1.2 OpenRail

Results for the OpenRail linear system were the following. To see complete calculations see appendix 5.6 for Studies and 5.12 for Strain exp. Solidworks simulation settings can be found in appendix 5.14

Table 7: Results of Open Rail experiments and calculations

OpenRail			
	Studies	Strain exp.	Simulation
δ_{max}	0.65 mm	0.55 mm	0.55 mm
f_n	19.54 Hz	21.2 Hz	80.80 Hz

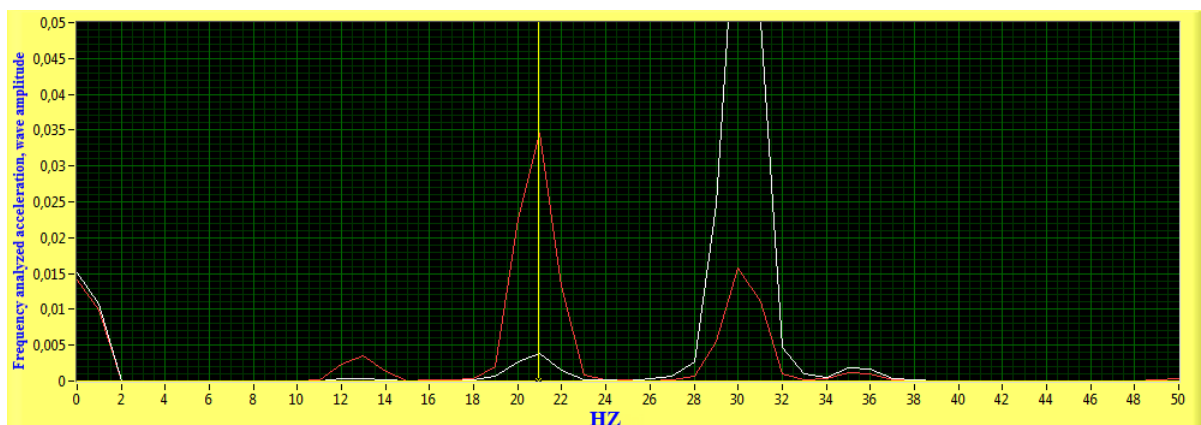


Figure 22: Screen shot from LabVIEW from a Natural frequency experiment

The natural frequency of the OpenRail beam is close to $21.2 Hz$ in Figure 22. Compared to Studies results in Table 7 we have a increase of 8.5% , from Strain exp. but a 313% increase from Simulation. Comparing the deflection the same way, we have a lower value of 15.4% from the experiment and the simulation. Similarly as in the results for the Makerslide beam in Section 3.1.1 the only thing that stands out is the unexpected high natural frequency from Solidworks Simulation, as we can see the displacement from the Strain exp. and Simulation is almost identical. Reasons for natural frequencies being higher are suspected to be the same as stated in Section 3.1.1.

3.1.3 Stainless steel rod

Results for the Stainless steel rod linear system were the following. Complete calculations can be seen in appendix 5.7 for Studies and 5.13 for Strain exp. Solidworks Simulation settings can be found in appendix 5.14

Table 8: Results of Stainless steel 8 mm rod experiment and calculations

Stainless steel rod			
	Studies	Strain exp.	Simulation
δ_{max}	12.96 mm	10.18 mm	3.89 mm
fn	4.37 Hz	5.2 Hz	8.69 Hz

After looking at the captured screen shots of this experiment a decision was made to not include any of them in this report because none of them were clear enough. It could be read from them with lots of zooming and careful observation that the natural frequency was close to $5.2 Hz$. Compared with the Studies results from Table 8, this gives a 19% increase in natural frequency but a 98% increase compared to the simulation results. As for the deflection, comparing to the Studies column, the difference was 21.46% lower from the experiment and 70% lower compared with the simulation results. Again reasons for strange simulation numbers is believed to be related to end fastening problems in Solidworks Simulation 2013.

3.2 Discussion

Strain gauges are a remarkable thing, I never imagined that they were as sensitive as the turned out to be. I noticed the first test I did with Indriði as you can see in Figure 23 that the strain gauges were affected by the small gust of wind that followed when someone walked by the test frame. The graph should be close to symmetric about the x-axis but the wind effects are causing it to go further away from it.

Seeing that the report and experiments part of the project took as long as it turned out to, I will be forced to change the scope of the project, see Future work in Section 4.1. The natural frequency of the Kysian stepper motor is around 5 Hz [32], at a maximum print speed of 250 mm/s . I chose this speed because I had to pick something, the speed varies greatly between the part your making, filament used, software used, structure of printer to name a few. Lowering the speed would also lower the frequency, even though that would be done the natural frequency of the stainless steel rod of 4.37 is uncomfortably close to that of the stepper motor of 5 Hz . We are pretty safe of not hitting the natural frequency of both the Open Rail at 19.54 Hz and Makerslide at 44.05 Hz . This excludes the 8 mm stainless steel rod from being used in a large scale printer. Now looking at deflection, the already excluded rod had 12.96 mm , Open Rail had 0.65 mm and is therefore excluded also. The Makerslide on the other hand had 0.12 mm in deflection. So the clear choice out of the three is the Makerslide with the highest natural frequency of 44.05 Hz and a maximum displacement of 0.12 mm and therefore meets both requirements stated in 2.6.

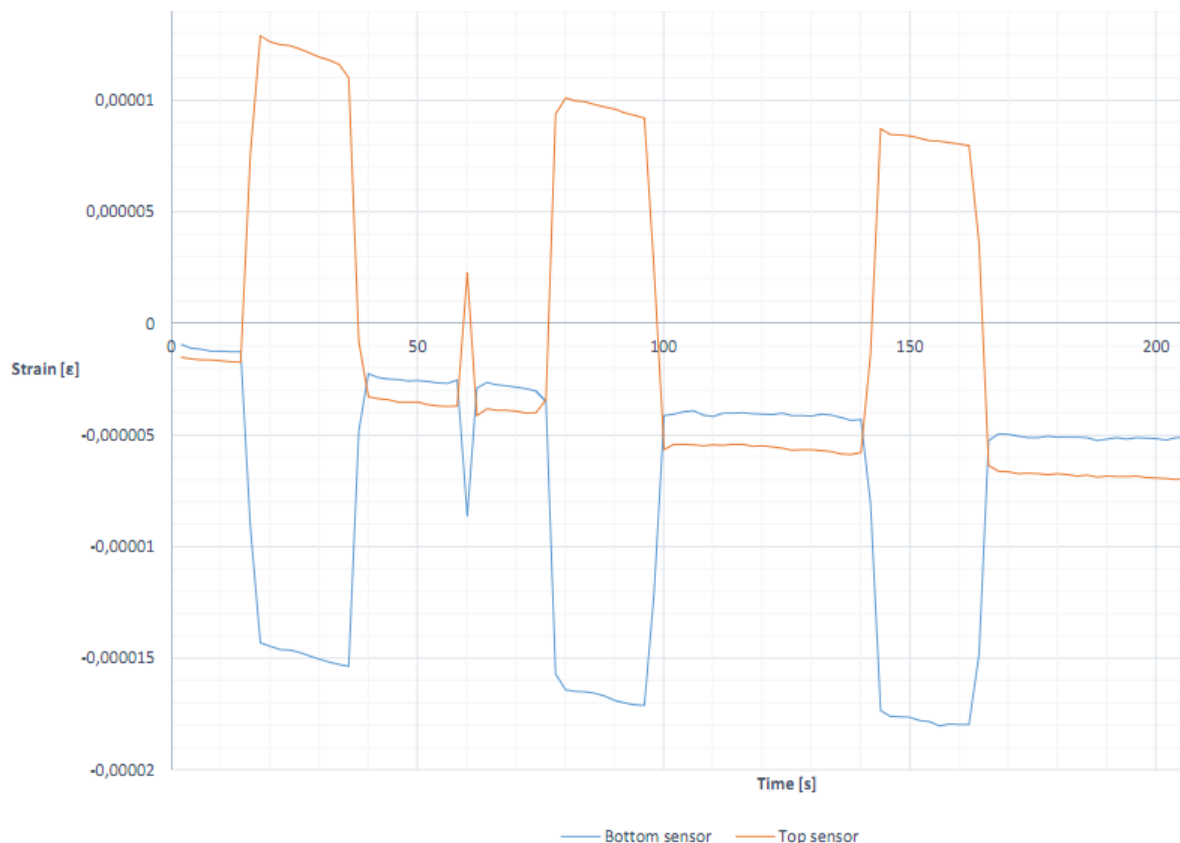


Figure 23: Strain gage reading from when someone walking by experiment in progress

4 Conclusion

We know that the deflection in a FDM 3D printer can not be more than 0.33 mm for it to be useful, see Section 2.6. In common practice for a $200 \times 200\text{ mm}$ print bed, 8 mm stainless steel rod is acceptable because it does not deflect that much on such a small span. This can be seen by looking at printed objects from existing Reprap printers [3]. As the size of the printer grows in length the rods will have problems with deflection. I plan to have the print bed on my first printer close to $400 \times 400\text{ mm}$ if that works out well I might take on an even bigger one. That's why I experimented with 1 meter long beams of all 3 systems, so that I could expand and still have useful data to build upon.

When we look at the data from the experiments in Section 3 we can see that the Makerslide has the least deflection out of the three at only 0.12 mm , next was the Open rail system with 0.65 mm and last was the stainless steel rod with 12.96 mm . This is from being subjected to a 21.14 N load, simulating the weight of approximately 3 Greg's Tilt Extruder's for multi color prints. The natural frequency of the stepper motor is around 5 Hz . The natural frequency of the Makerslide is 44.05 Hz and 19.54 Hz for the Open Rail, the 8 mm stainless steel rod had a natural frequency of 4.37 Hz this is too close to the 5 Hz of the stepper motor for it to be used in a large scale printer, but also its deflection is 12.96 mm which is an amusing 3827% over the requirement of maximum deflection would not exceed 0.33 mm . Open Rail's deflection, 0.65 mm is 97% over the maximum but Makerslide's deflection was only 0.12 mm and therefore it is the one that will be used for constructing the large scale 3D printer

4.1 Future work

Would include finishing the detailed assembly instructions so that other people could build a printer for themselves with little knowledge of mechanics and programming. Also would be interesting to build an even bigger one with a print bed along the lines of $1 \times 1\text{ m}$, but that would require more experiments, bigger stepper motors and controllers among other things.

In that category falls the assembly instructions that were supposed to detail every aspect of putting the printer together in an IKEA sort of way, using of course the newly crowned winning Makerslide linear system instead of the normal 8 mm Stainless steel rods.

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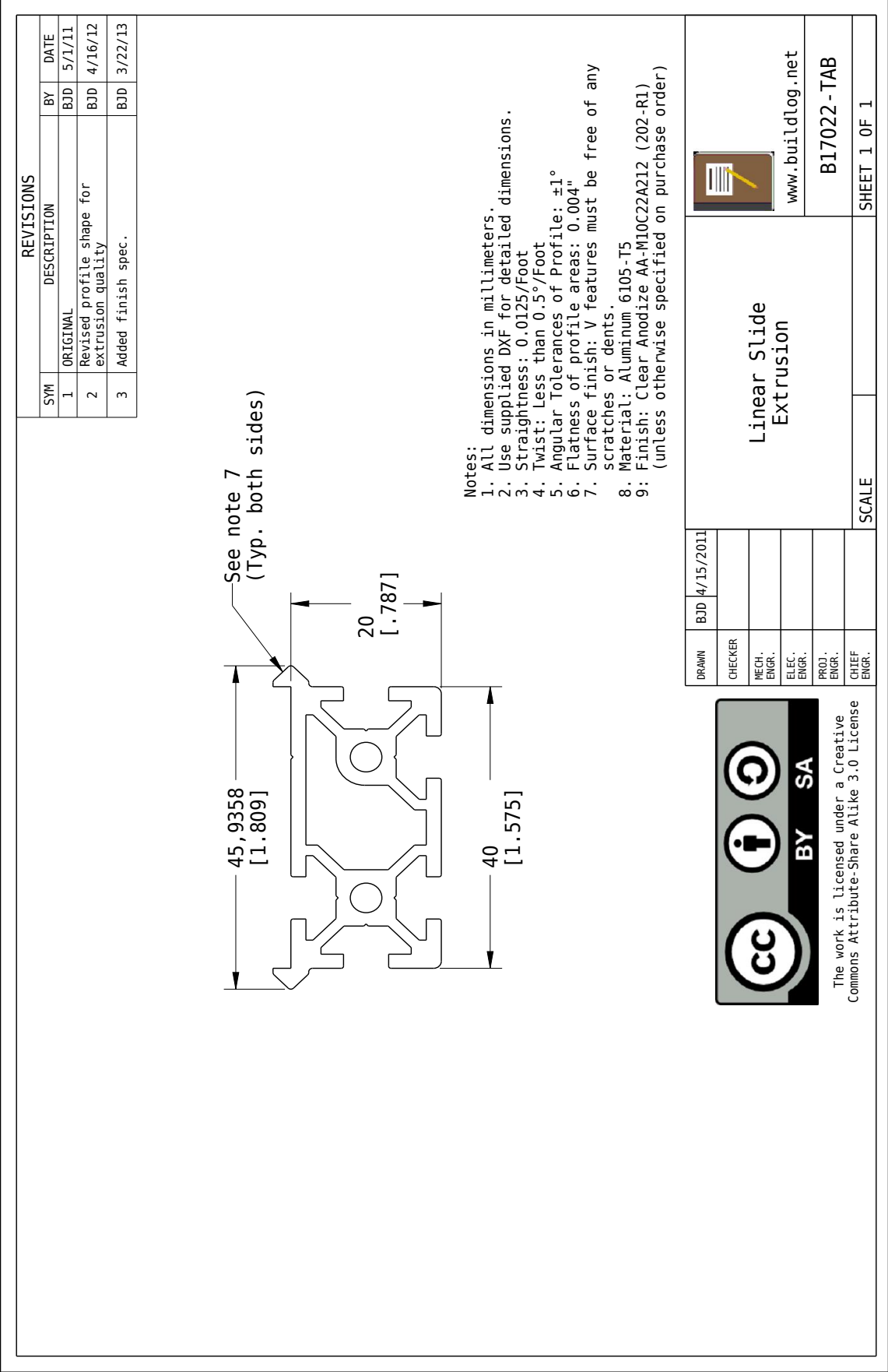
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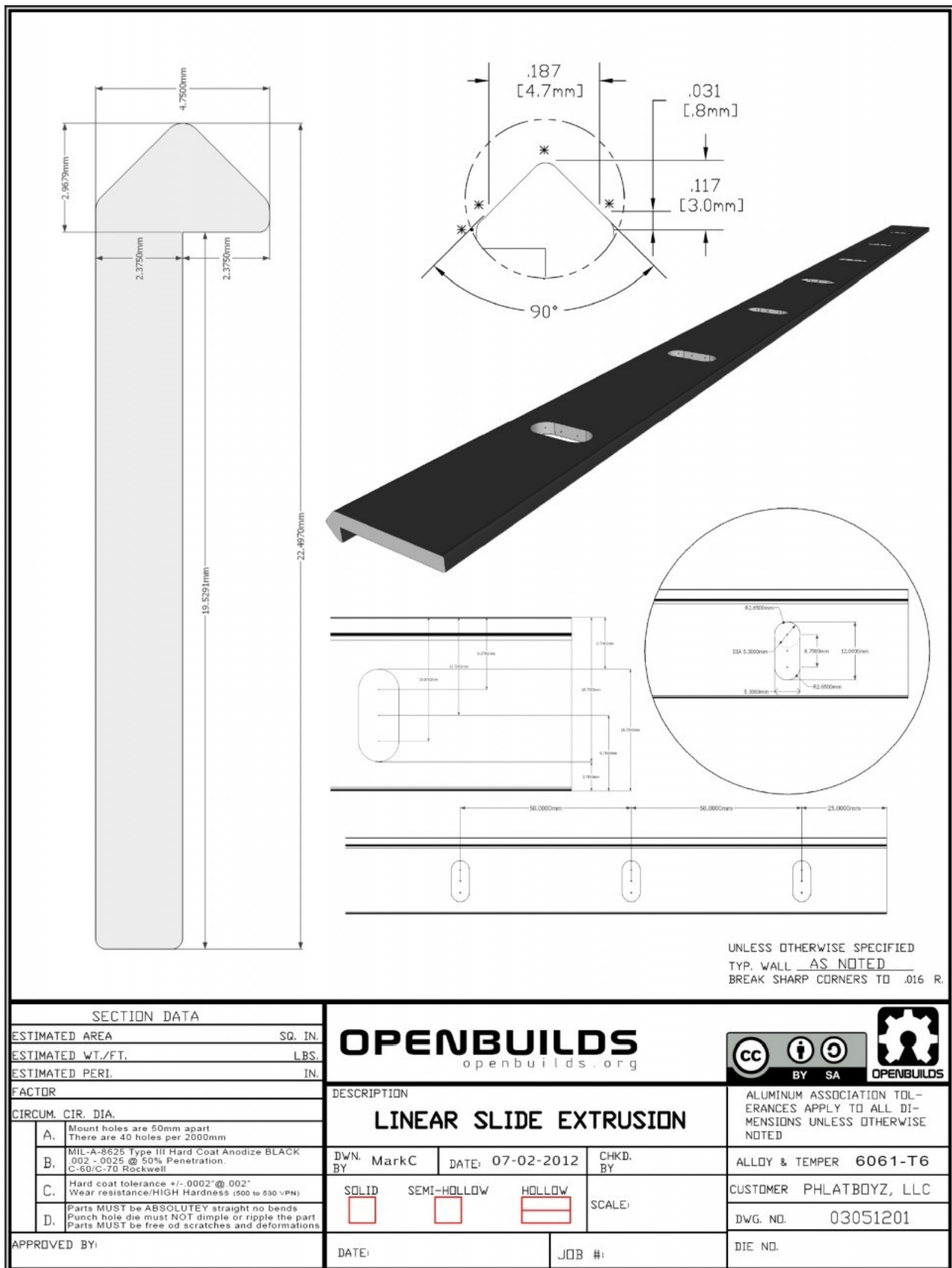
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5 Appendix

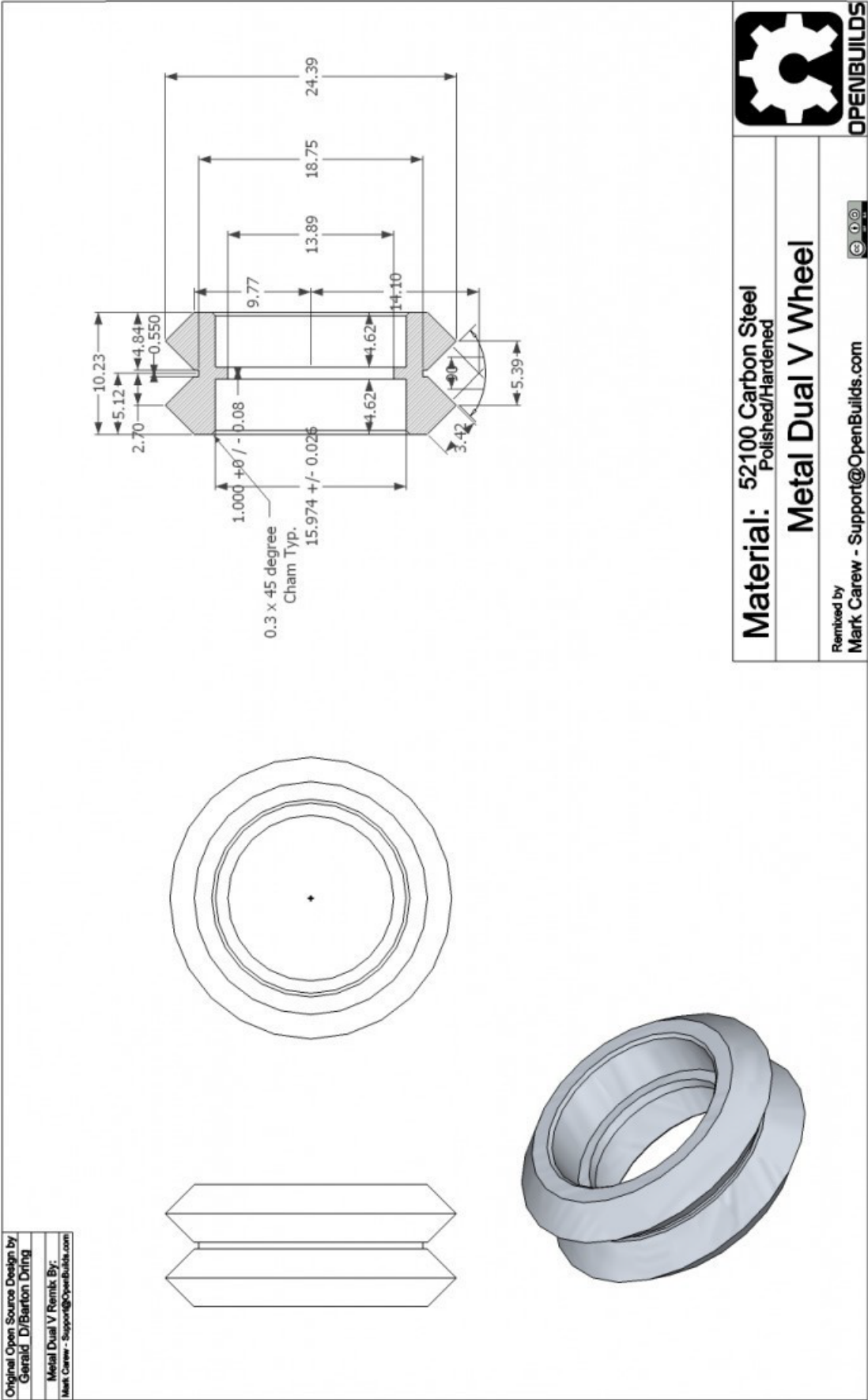
5.1 Makerslide drawing



5.2 OpenRail drawing



5.3 V-wheel drawing



5.4 Monitran accelerometer data sheets



MTN/1100 Series

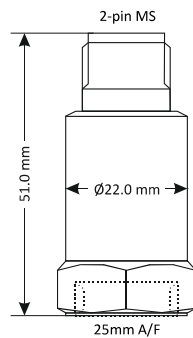
General purpose industrial accelerometer

General purpose top-entry constant current accelerometer with isolated AC output. Made from robust stainless steel throughout for long term vibration analysis in harsh environments. Sealed to IP67 and includes 2-pin C5015 military style connector. Available with a wide range of mountings.

MTN/1100



Dimensions



Applications

- Data collection
- Heavy industry
- Paper machinery

Technical

Standard sensitivity 100mV/g $\pm 10\%$ nominal @ 80Hz

Frequency response 2Hz to 10kHz $\pm 5\%$ (-3dB @ 0.8Hz)

Mounted base resonance 18kHz (nominal)

Isolation Base isolated

Dynamic range $\pm 80g$

Transverse sensitivity Less than 5%

Electrical noise 0.1mg max

Current range 0.5 to 8mA

Temperature range -55 to 140°C

Bias voltage 12V DC (nominal)

Case material Stainless steel

Mating connector MTN/MH002

Maximum cable length 1000m

Mounting torque 8Nm

Weight 110g (nominal)

Sealing IP67

Monitran Ltd | Monitor House | 33 Hazlemere Road | Penn | Bucks | UK | HP10 8AD
Telephone +44 (0)1494 816569 | E-mail info@monitran.com | Website www.monitran.com



We reserve the right to alter specifications without prior notice.

DS0200.1 Page 1 of 2

5.5 Makerslide hand calculations

Makerslide deflection and frequency calculations

The Makerslide is made from 6105-T5 aluminium and has the following characteristics.

Modulus of elasticity $E := 70 \text{ GPa}$

Moment of inertia $I_y := 61012.91 \text{ mm}^4$

Length of beam $L := 1 \text{ m}$

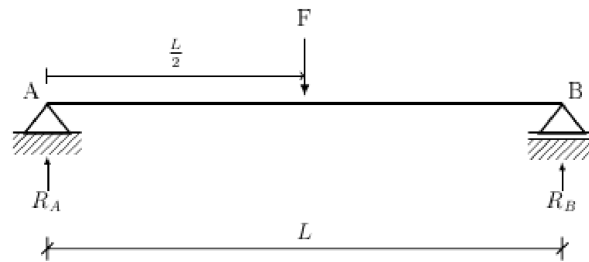
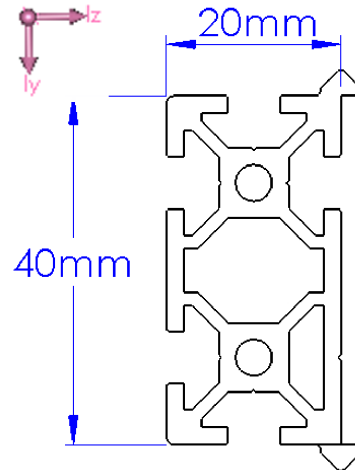
Mass of the Block $mass := 2.210 \text{ kg}$

Gravity $g := 9.81 \frac{\text{m}}{\text{s}^2}$

Force of block $F := mass \cdot g = 21.68 \text{ N}$

Beam weight $W_{Beam} := 0.745 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 7.308 \text{ N}$



The Resultant forces are according to Equation (1)

$$R_A := \frac{F}{2} \quad R_B := R_A \quad R_B = 10.84 \text{ N}$$

With Equation (2) the maximum moment is at $L/2$ and is

$$M_{Beam} := \frac{(F \cdot L)}{4} \quad M_{Beam} = 5.42 \text{ N} \cdot \text{m}$$

Maximum deflection occurs also at $L/2$ and is according to Equation (3)

$$\delta_{maxBeam} := \frac{(F \cdot L^3)}{48 \cdot E \cdot I_y} \quad \delta_{maxBeam} = 0.106 \text{ mm}$$

Now to take into account the weight of the beam and find the force F_q with Equation (4)

$$F_q := \frac{W_{Beam} \cdot g}{L} = 7.308 \frac{\text{N}}{\text{m}}$$

Makerslide deflection and frequency calculations

Using Equation (7) we can find the maximum deflection due to the weight of the beam

$$\delta_{maxUnif} := \frac{5 \cdot F_q \cdot L^4}{384 \cdot E \cdot I_y} = 0.022 \text{ mm}$$

Now either adding both deflection terms up or using Equation (8) gives the same result

$$\delta := \frac{L^3}{E \cdot I_y} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F}{48} \right) = 0.128 \text{ mm} \quad \delta_{maxBeam} + \delta_{maxUnif} = 0.128 \text{ mm}$$

Natural frequency

The spring equivalent constant K_{eq} is found with Equation (16)

$$K_{eq} := \frac{(48 \cdot E \cdot I_y)}{L^3} = 205.003 \frac{kN}{m}$$

The natural frequency is given by Equation (15)

$$\omega := \sqrt{\frac{K_{eq}}{mass}} = 304.568 \frac{rad}{s}$$

Which we can convert to Hz with Equation (17)

$$f := \frac{\omega}{2 \pi} = 48.474 \text{ Hz}$$

To find the period from peak to peak we use Equation (18)

$$T := \frac{1}{f} = 0.021 \text{ s}$$

In order to get the natural frequency of the whole system, we must use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta}}}{(2 \pi)} = 44.054 \text{ Hz}$$

5.6 OpenRail hand calculations

OpenRail deflection and frequency calculations

The OpenRail is made from 6061-T6 aluminium and has the following characteristics.

Modulus of elasticity $E := 70 \text{ GPa}$

Moment of inertia $I := 12012.02 \text{ mm}^4$

Length of beam $L := 1 \text{ m}$

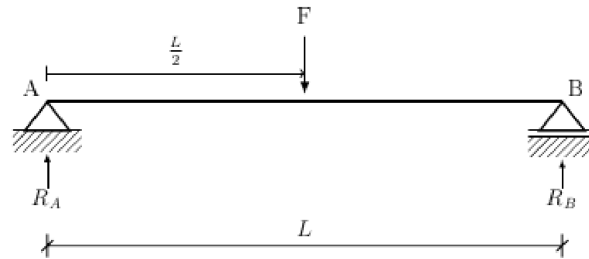
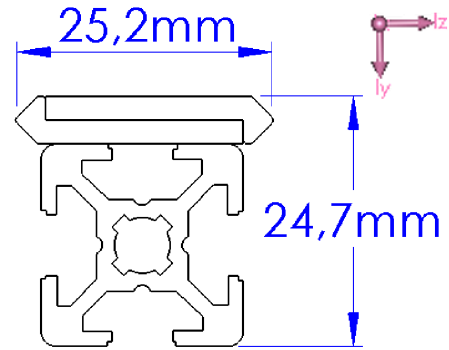
Mass of the Block $mass := 2.210 \text{ kg}$

Gravity $g := 9.81 \frac{\text{m}}{\text{s}^2}$

Force of block $F := mass \cdot g = 21.68 \text{ N}$

Beam weight $W_{Beam} := 0.745 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 7.308 \text{ N}$



The Resultant forces are according to Equation (1)

$$R_A := \frac{F}{2} \quad R_B := R_A \quad R_B = 10.84 \text{ N}$$

With Equation (2) the maximum moment is at $L/2$ and is

$$M_{Beam} := \frac{(F \cdot L)}{4} \quad M_{Beam} = 5.42 \text{ N} \cdot \text{m}$$

Maximum deflection occurs also at $L/2$ and is according to Equation (3)

$$\delta_{maxBeam} := \frac{(F \cdot L^3)}{48 \cdot E \cdot I} \quad \delta_{maxBeam} = 0.537 \text{ mm}$$

Now to take into account the weight of the beam and find the force F_q with Equation (4)

$$F_q := \frac{W_{Beam} \cdot g}{L} = 7.308 \frac{\text{N}}{\text{m}}$$

OpenRail deflection and frequency calculations

Using Equation (7) we can find the maximum deflection due to the weight of the beam

$$\delta_{maxUnif} := \frac{5 \cdot F_q \cdot L^4}{384 \cdot E \cdot I} = 0.113 \text{ mm}$$

Now either adding both deflection terms up or using Equation (8) gives the same result

$$\delta := \frac{L^3}{E \cdot I} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F}{48} \right) = 0.65 \text{ mm} \quad \delta_{maxBeam} + \delta_{maxUnif} = 0.65 \text{ mm}$$

Natural frequency

The spring equivalent constant K_{eq} is found with Equation (16)

$$K_{eq} := \frac{(48 \cdot E \cdot I)}{L^3} = 40.36 \frac{kN}{m}$$

The natural frequency is given by Equation (15)

$$\omega := \sqrt{\frac{K_{eq}}{mass}} = 135.139 \frac{rad}{s}$$

Which we can convert to Hz with Equation (17)

$$f := \frac{\omega}{2 \pi} = 21.508 \text{ Hz}$$

To find the period from peak to peak we use Equation (18)

$$T := \frac{1}{f} = 0.046 \text{ s}$$

In order to get the natural frequency of the whole system, we must use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta}}}{(2 \pi)} = 19.547 \text{ Hz}$$

5.7 Steel rod hand calculations

Steel rod deflection and frequency calculations

The steel rod is made from 304 H9 Stainless steel and has the following characteristics.

Modulus of elasticity $E := 193 \text{ GPa}$

Moment of inertia $I_y := 201.06 \text{ mm}^4$

Length of beam $L := 1 \text{ m}$

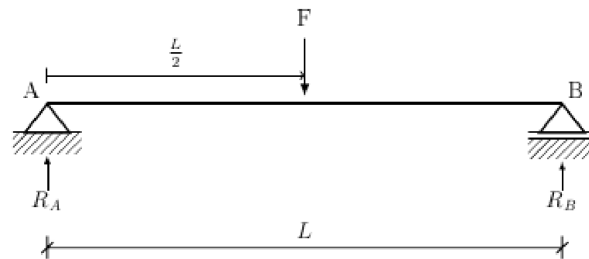
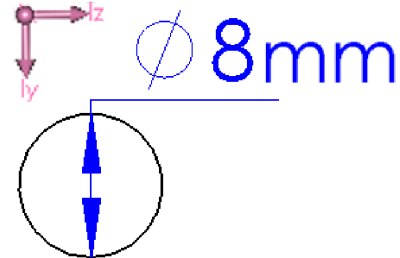
Mass of the Block $mass := 2.210 \text{ kg}$

Gravity $g := 9.81 \frac{\text{m}}{\text{s}^2}$

Force of block $F := mass \cdot g = 21.68 \text{ N}$

Beam weight $W_{Beam} := 0.412 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 4.042 \text{ N}$



The Resultant forces are according to Equation (1)

$$R_A := \frac{F}{2} \quad R_B := R_A \quad R_B = 10.84 \text{ N}$$

With Equation (2) the maximum moment is at $L/2$ and is

$$M_{Beam} := \frac{(F \cdot L)}{4} \quad M_{Beam} = 5.42 \text{ N} \cdot \text{m}$$

Maximum deflection occurs also at $L/2$ and is according to Equation (3)

$$\delta_{maxBeam} := \frac{(F \cdot L^3)}{48 \cdot E \cdot I_y} \quad \delta_{maxBeam} = 11.64 \text{ mm}$$

Now to take into account the weight of the beam and find the force F_q with Equation (4)

$$F_q := \frac{W_{Beam} \cdot g}{L} = 4.042 \frac{\text{N}}{\text{m}}$$

Steel rod deflection and frequency calculations

Using Equation (7) we can find the max deflection due to the weight of the beam

$$\delta_{maxUnif} := \frac{5 \cdot F_q \cdot L^4}{384 \cdot E \cdot I_y} = 1.356 \text{ mm}$$

Now either adding both deflection terms up or using Equation (8) gives the same result

$$\delta := \frac{L^3}{E \cdot I_y} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F}{48} \right) = 12.996 \text{ mm} \quad \delta_{maxBeam} + \delta_{maxUnif} = 12.996 \text{ mm}$$

Natural frequency

The spring equivalent constant K_{eq} is found with Equation (16)

$$K_{eq} := \frac{(48 \cdot E \cdot I_y)}{L^3} = 1.863 \frac{kN}{m}$$

The natural frequency is given by Equation (15)

$$\omega := \sqrt{\frac{K_{eq}}{mass}} = 29.031 \frac{rad}{s}$$

Which we can convert to Hz with Equation (17)

$$f := \frac{\omega}{2 \pi} = 4.62 \text{ Hz}$$

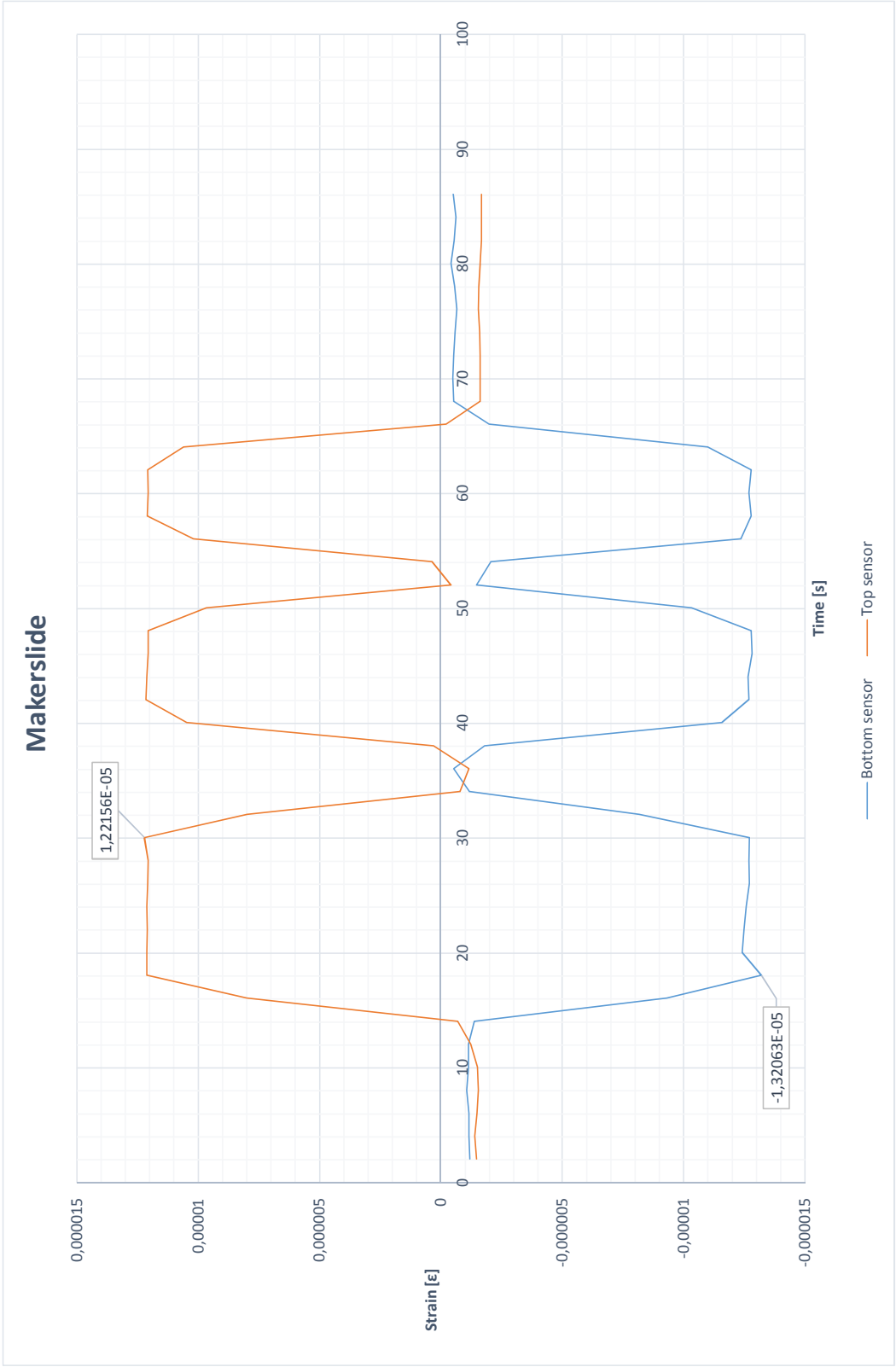
To find the period from peak to peak we use Equation (18)

$$T := \frac{1}{f} = 0.216 \text{ s}$$

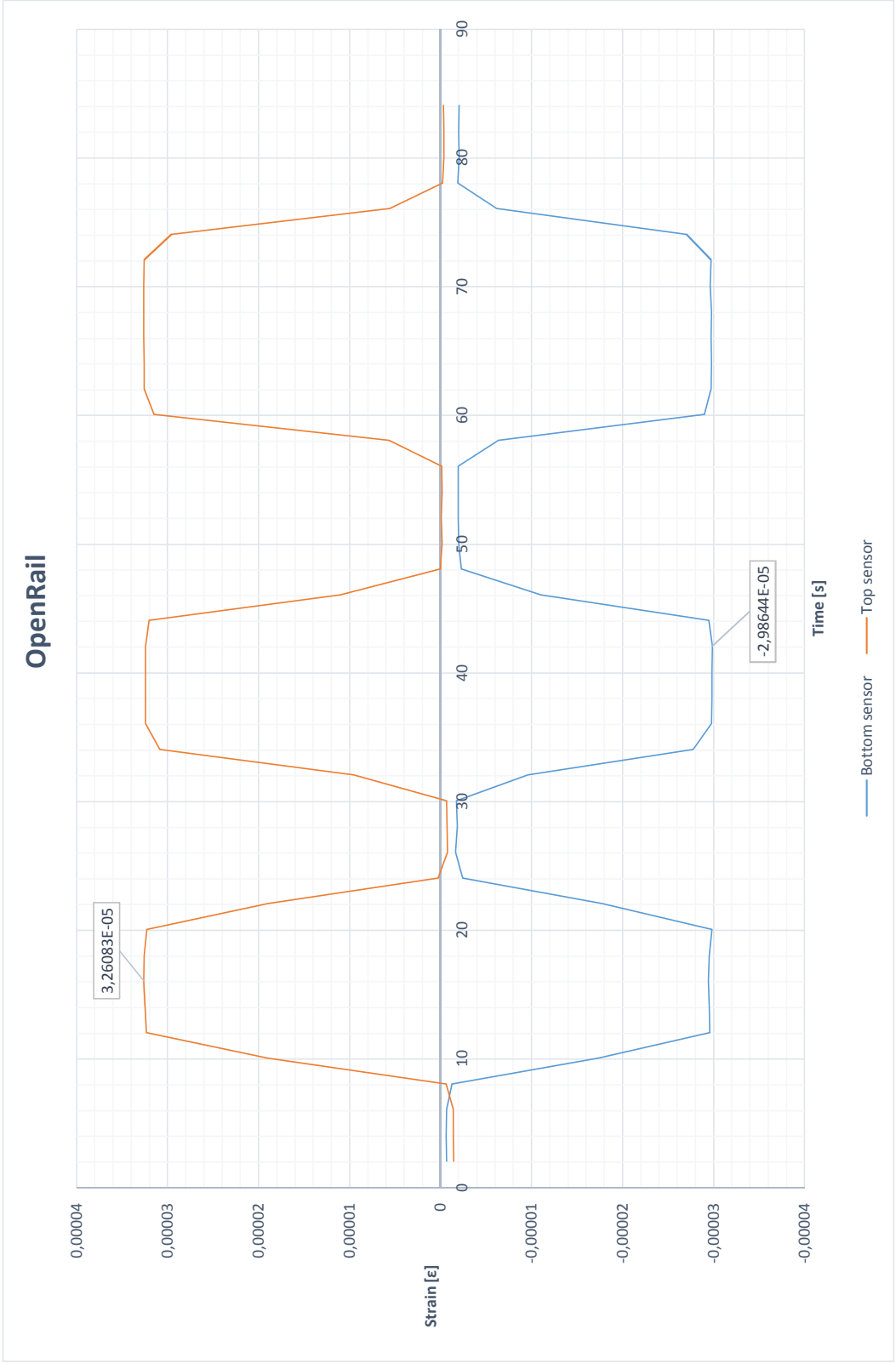
In order to get the natural frequency of the whole system, we must use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta}}}{(2 \pi)} = 4.373 \text{ Hz}$$

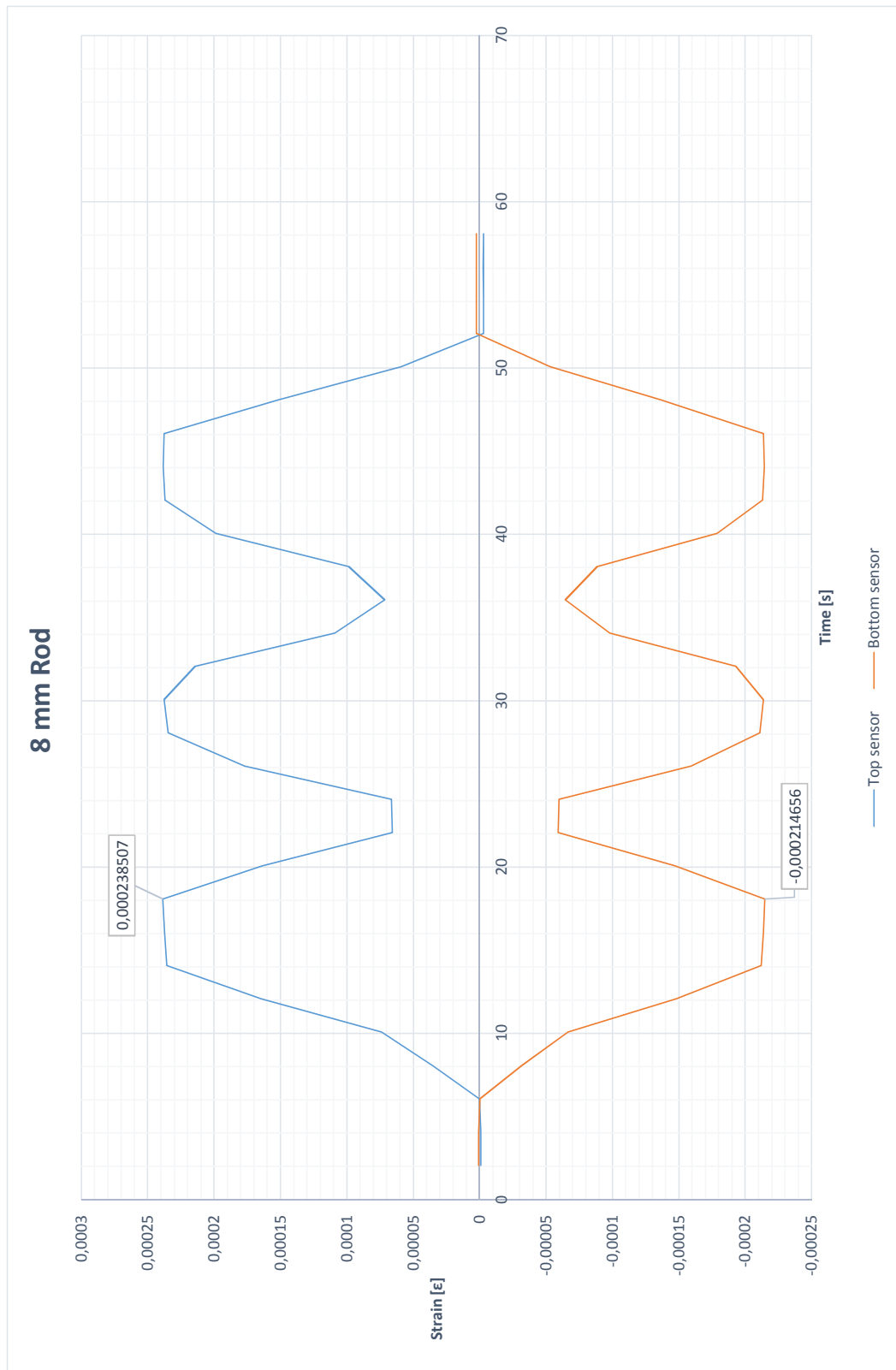
5.8 Strain data from Makerslide experiment



5.9 Strain data from OpenRail experiment



5.10 Strain data from 8 mm rod experiment

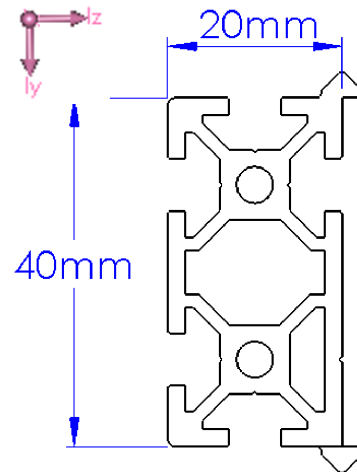


5.11 Hand calculations with strain data for Makerslide

Makerslide calculations from data

The Makerslide is made from 6105-T5 aluminium and has the following characteristics.

Modulus of elasticity	$E := 70 \text{ GPa}$
Moment of inertia	$I_y := 61012.91 \text{ mm}^4$
Length of beam	$L := 1 \text{ m}$
Mass of the Block	$mass := 2.210 \text{ kg}$
Gravity	$g := 9.81 \frac{\text{m}}{\text{s}^2}$
Force of block	$F_{Block} := mass \cdot g = 21.68 \text{ N}$



Length from neutral axis of beam to top of beam $c := 20 \text{ mm}$

Beam weight $W_{Beam} := 0.895 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 8.78 \text{ N}$

Uniform load F_q working on the beam because of it's own weight we use Equation (4)

$$F_q := \frac{W_{Beam} \cdot g}{1 \text{ m}} = 8.78 \frac{\text{N}}{\text{m}}$$

Then use Equation (6) to find the maximum moment of the beam from force F_q

$$M_{maxUnif} := \frac{F_q \cdot L^2}{8} = 1.097 \text{ N} \cdot \text{m}$$

Then we can insert our strain results from experiments

$$\varepsilon := 0.000013$$

Use Equation (10) to get the stress at work

$$\sigma := \varepsilon \cdot E = 0.91 \text{ MPa}$$

Makerslide calculations from data

Now to calculate the maximum moment according to our data with Equation (11)

$$M_{maxDATA} := \frac{\sigma \cdot I_y}{\frac{c}{2}} = 5.552 \text{ } N \cdot m$$

Using Equation (12) gives us the moment acting on the beam

$$M_{Beam} := M_{maxDATA} - M_{maxUnif} = 4.455 \text{ } N \cdot m$$

Then using Equation (12) we can find the force acting on the beam

$$F_{Block} := \frac{4 \cdot M_{Beam}}{L} = 17.819 \text{ } N$$

Finally using Equation (14) to get the maximum deflection of the beam according to the data

$$\delta_{maxDATA} := \frac{L^3}{E \cdot I_y} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F_{Block}}{48} \right) = 0.114 \text{ } mm$$

To get the natural frequency of the whole system we use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta_{maxDATA}}}}{(2 \pi)} = 46.752 \text{ } Hz$$

Using the graph produced by the accelerometer, we can also see what the max deflection is according to that data by solving Equation (19) for delta max

$$f_{nDATA2} := 46.8 \text{ } Hz$$

$$\delta_{maxDATA2} := \frac{g}{4 \cdot (f_{nDATA2})^2 \cdot (2 \cdot \pi)^2} = 0.028 \text{ } mm$$

5.12 Hand calculations with strain data for OpenRail

OpenRail calculations from data

The OpenRail is made from 6061-T6 aluminium and has the following characteristics.

Modulus of elasticity $E := 70 \text{ GPa}$

Moment of inertia $I_y := 12012.02 \text{ mm}^4$

Length of beam $L := 1 \text{ m}$

Mass of the Block $mass := 2.210 \text{ kg}$

Gravity $g := 9.81 \frac{\text{m}}{\text{s}^2}$

Force of block $F_{Block} := mass \cdot g = 21.68 \text{ N}$

Length from neutral axis of beam to top of beam $c := 10 \text{ mm}$

Beam weight $W_{Beam} := 0.745 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 7.308 \text{ N}$

Uniform load F_q working on the beam because of it's own weight we use Equation (4)

$$F_q := \frac{W_{Beam} \cdot 1}{1} = 7.308 \frac{\text{N}}{\text{m}}$$

Then use Equation (6) to find the maximum moment of the beam from force F_q

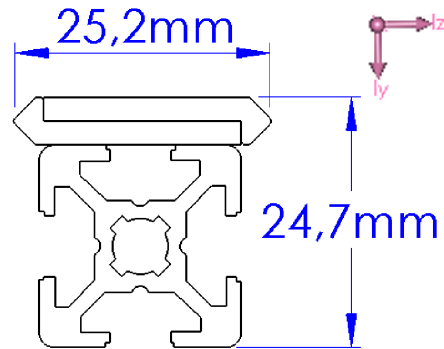
$$M_{maxUnif} := \frac{F_q \cdot L^2}{8} = 0.914 \text{ N} \cdot \text{m}$$

Then we can insert our strain results from experiments

$$\varepsilon := 0.000032$$

Use Equation (10) to get the stress at work

$$\sigma := \varepsilon \cdot E = 2.24 \text{ MPa}$$



OpenRail calculations from data

Now to calculate the maximum moment according to our data with Equation (11)

$$M_{maxDATA} := \frac{\sigma \cdot I_y}{\frac{c}{2}} = 5.381 \text{ } N \cdot m$$

Using Equation (12) gives us the moment acting on the beam

$$M_{Beam} := M_{maxDATA} - M_{maxUnif} = 4.468 \text{ } N \cdot m$$

Then using Equation (12) we can find the force acting on the beam

$$F_{Block} := \frac{4 \cdot M_{Beam}}{L} = 17.871 \text{ } N$$

Finally using Equation (14) to get the maximum deflection of the beam according to the data

$$\delta_{maxDATA} := \frac{L^3}{E \cdot I_y} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F_{Block}}{48} \right) = 0.556 \text{ } mm$$

To get the natural frequency of the whole system we use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta_{maxDATA}}}}{(2 \pi)} = 21.141 \text{ } Hz$$

5.13 Hand calculations with strain data for steel rod

Steel rod calculations from data

The steel rod is made from 304 H9 Stainless steel and has the following characteristics.

Modulus of elasticity $E := 193 \text{ GPa}$

Moment of inertia $I_y := 201.06 \text{ mm}^4$

Length of beam $L := 1 \text{ m}$

Mass of the Block $mass := 2.210 \text{ kg}$

Gravity $g := 9.81 \frac{\text{m}}{\text{s}^2}$

Force of block $F_{Block} := mass \cdot g = 21.68 \text{ N}$

Length from neutral axis of beam to top of beam $c := 4 \text{ mm}$

Beam weight $W_{Beam} := 0.412 \text{ kg}$

Force of beam $F_{Beam} := W_{Beam} \cdot g = 4.042 \text{ N}$

Uniform load F_q working on the beam because of its own weight we use Equation (4)

$$F_q := \frac{W_{Beam} \cdot 1}{1} = 4.042 \frac{\text{N}}{\text{m}}$$

Then use Equation (6) to find the maximum moment of the beam from force F_q

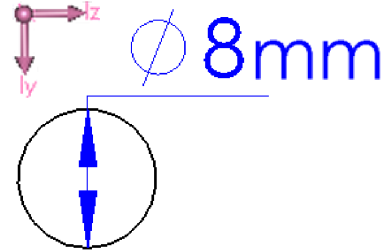
$$M_{maxUnif} := \frac{F_q \cdot L^2}{8} = 0.505 \text{ N} \cdot \text{m}$$

Then we can insert our strain results from experiments

$$\varepsilon := 0.000238$$

Use Equation (10) to get the stress at work

$$\sigma := \varepsilon \cdot E = 45.934 \text{ MPa}$$



Steel rod calculations from data

Now to calculate the maximum moment according to our data with Equation (11)

$$M_{maxDATA} := \frac{\sigma \cdot I_y}{\frac{c}{2}} = 4.618 \text{ } \mathbf{N \cdot m}$$

Using Equation (12) gives us the moment acting on the beam

$$M_{Beam} := M_{maxDATA} - M_{maxUnif} = 4.113 \text{ } \mathbf{N \cdot m}$$

Then using Equation (12) we can find the force acting on the beam

$$F_{Block} := \frac{4 \cdot M_{Beam}}{L} = 16.45 \text{ } \mathbf{N}$$

Finally using Equation (14) to get the max deflection of the beam according to the data

$$\delta_{maxDATA} := \frac{L^3}{E \cdot I_y} \cdot \left(\frac{(5 \cdot F_{Beam})}{384} + \frac{F_{Block}}{48} \right) = 10.188 \text{ } \mathbf{mm}$$

To get the natural frequency of the whole system we use Equation (19)

$$f_n := \frac{\sqrt{\frac{g}{\delta_{maxDATA}}}}{(2 \pi)} = 4.939 \text{ } \mathbf{Hz}$$

5.14 Solidworks Simulation Pro settings

All simulations were run with the same settings, fixtures and loads. A split line was inserted, that was the same size as the block that was used during experiments. This is where a load of 21.68 N was applied. Gravity worked on the whole beam. One end of the beam was fixed in all 3 directions but allowed rotational translation. The other end was fixed the same way except allowing movement along the length of the beam. Temperature was set to 25°C and meshing was set to maximum. Nodes were 61.011, Elements were 32.223 and Degrees of freedom were 496.485.

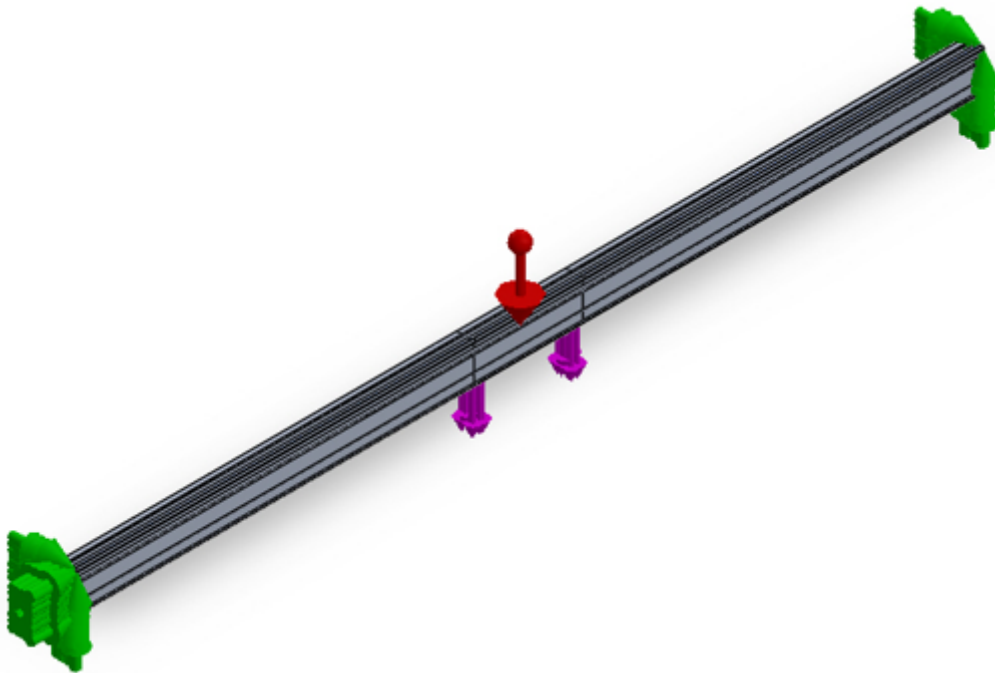


Figure 24: Fixtures and loads



Figure 25: Mesh set to max

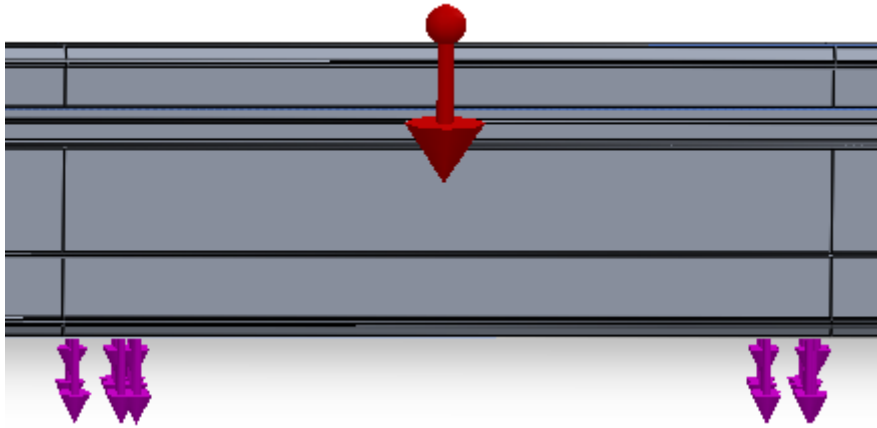


Figure 26: Split line inserted in the middle of the beams

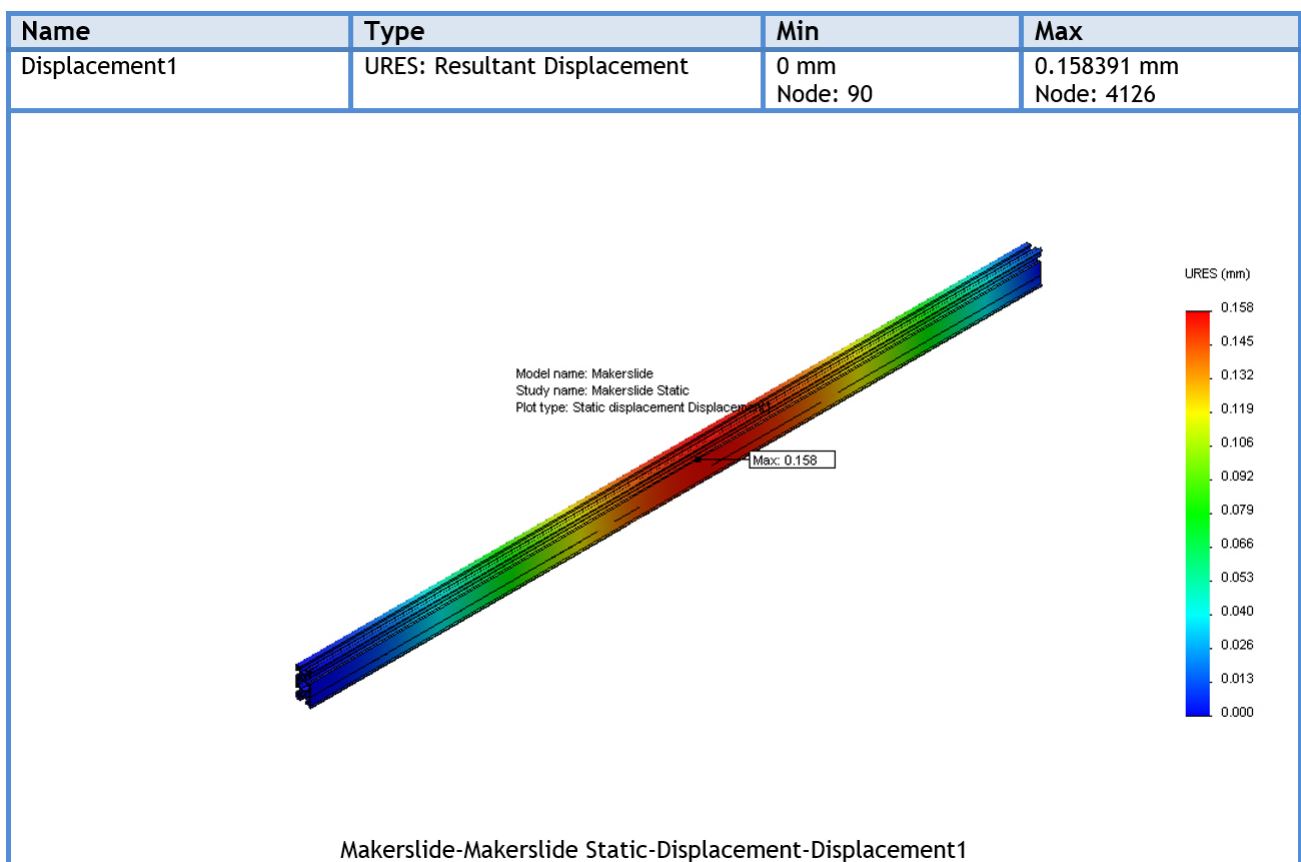


Figure 27: Result sample

5.15 Test frame drawing

6

5.16 Solidworks Model

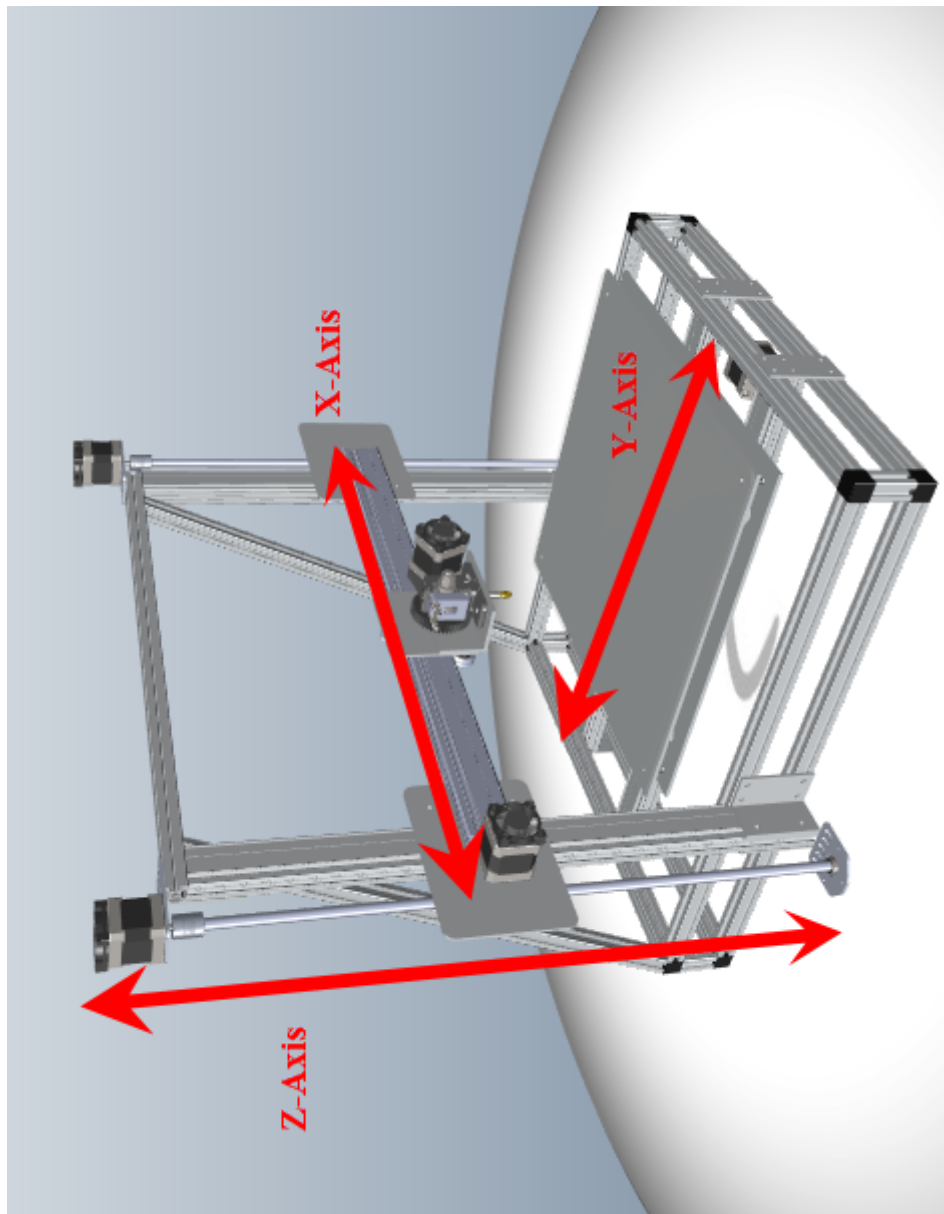


Figure 28: Axis movement on new printer

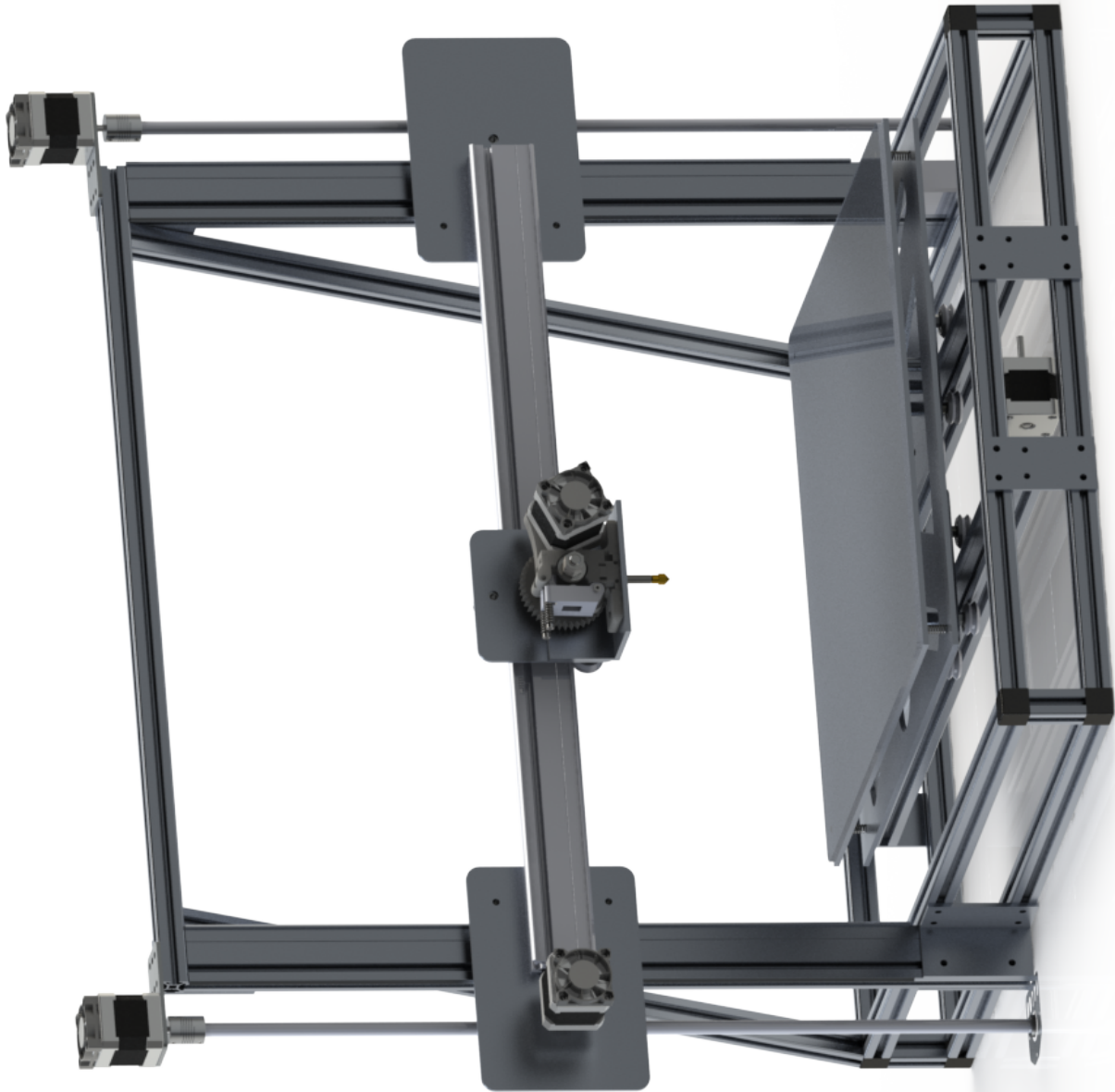


Figure 29: Solidworks assembly of the modelled parts so far