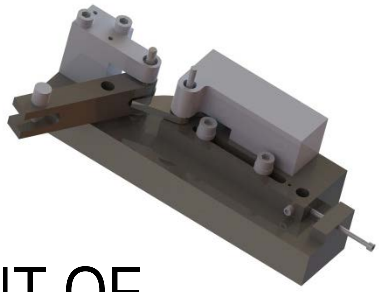

INTERNSHIP ASSIGNMENT

INVESTIGATION AND IMPROVEMENT OF A FRICTION-BASED JOINT LOCK

(AS PART OF THE MYOPRO PROJECT)

20 FEBRUARY- 6 JULY 2012

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Summary

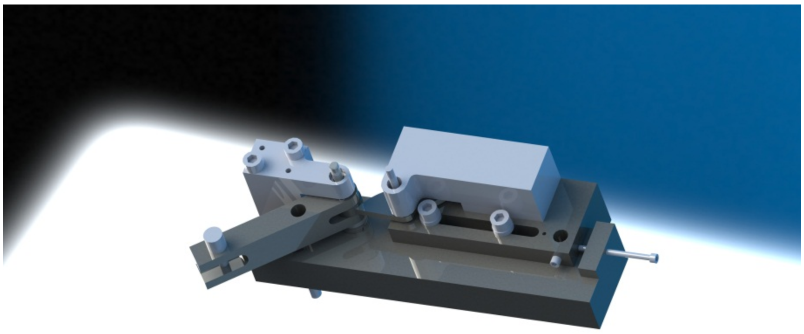


Figure 1 The joint lock test rig

In a consortium called the ‘MYOPRO project’ the University of Twente develops a new type of myo-electric prosthetic. As part of the project under actuated fingers is developed, fingers who can be controlled by pulleys and wires and joint locks.

This report is about how a joint lock works, how a joint lock test rig is develop, research on the joint lock and how a joint locks can be improved.

The idea of a under actuated finger is that with one motor a whole hand can be actuated. The system works with wires and pulleys. The problem is that it is not possible to control each joint separately. That is why a joint lock is developed, to lock each joint and control the fingers motion (Peerdeman, 2012) (pieterse, 2012).

Pieterse and Peerdeman, design concepts of the joint lock, did experimental tests on different concepts of the joint lock and chose test a joint lock system, the pawl-drum system.

The pawl-drum system is a drum is connected to one phalange and the pawl sits in the other phalange. When locking is needed the pawl is pushed to the drum and a torque is applied on the drum and the system locks.

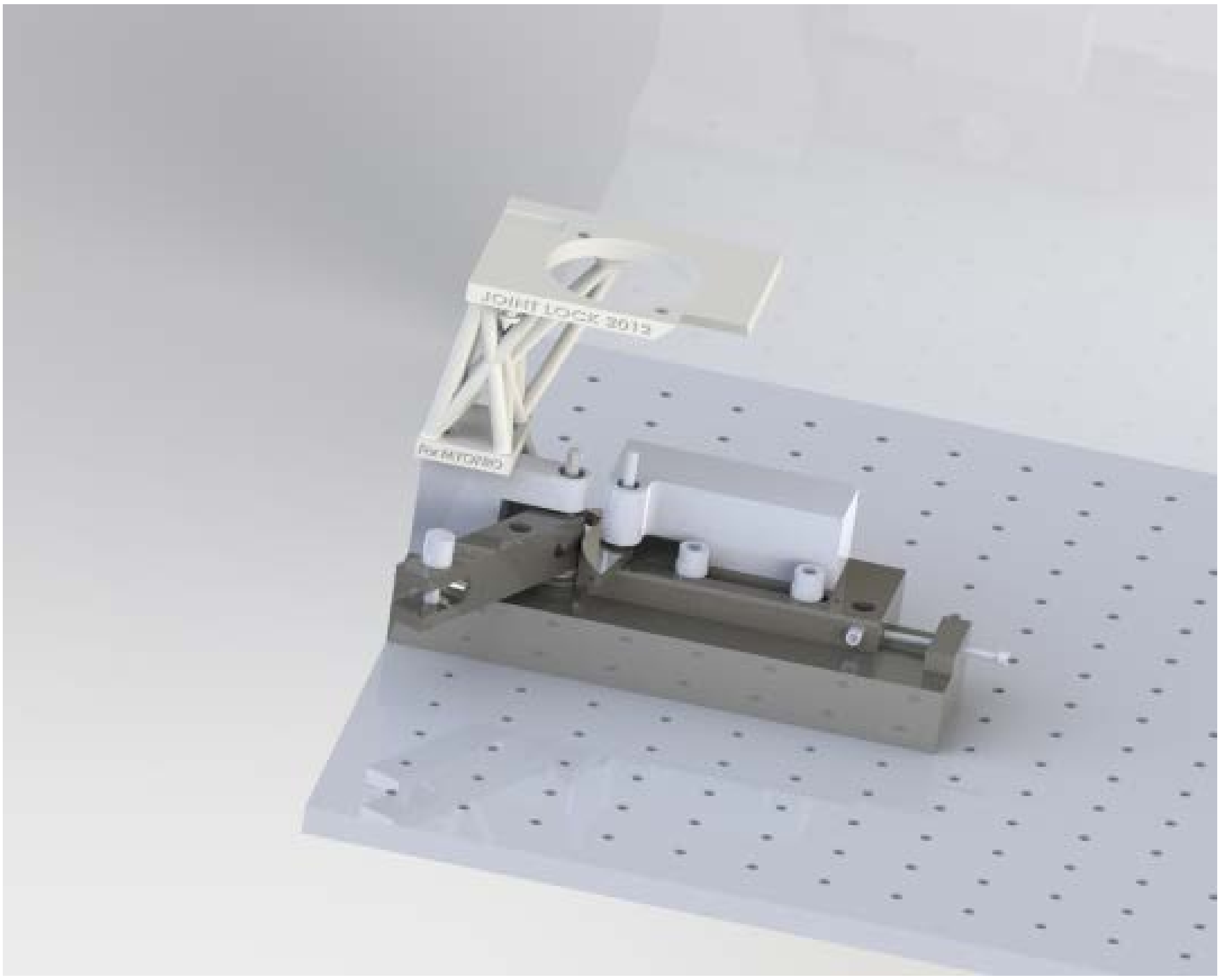
A test rig is design and build for testing the joint lock. The lock is tested on the forces in the system, the friction between the pawl and the drum and the influence from dimension of the system. Also a model is made to calculate theoretical what the influences is from the dimension on the forces and the friction of the system. As result of that, the contact angle between the drum and the pawl, the radius of the drum, and contact line between the drum and pawl are very important for the forces and the needed friction in the system.

Several ideas for improving the lock are developed. Such as a spiral surface for the pawl, coatings and other ideas.

Research is done in the test rig especially on friction between the drum and the pawl. Some tested coatings for extra friction found promising: a coating from rubber, and a coating with silicon carbide particles. But further research is needed on these coatings.

Also a several redesigns of the lock are proposed.

Preface/Voorwoord (English/Dutch)



Beste lezer,

Met dit verslag wil ik u een idee geven hoe de joint lock blokkering werkt, het onderzoek gegaan is en hoe het verbeterd kan worden. U leest de werking, hoe een test installatie gebouwd is, hoe testen gedaan zijn zowel in een model als in de test installatie.
Graag zou ik Dannis Brouwer en Bart Peerdeman bedanken voor hun begeleiding. Ook wil ik Edsko Hekman en Sarthak Misra voor hun inbreng bedanken. Dankzij deze mensen hun tijd, goede discussies, ondersteuning en energie heb ik afgelopen 5 maanden veel geleerd en naar ik hoop een resultaat afgeleverd te hebben waar men wat mee verder kan.

Arnoud Stapelbroek

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1. Introduction

1.1 Position of the project

The myoelectric prosthetic hands who are used by amputees today have many problems. They are too heavy, lack of functionality, no good control and problems with energy supply. That is why many of the users do not used them. To solve these problems a consortium:
Reference: (Atkins DJ) (B. Peerdeman, July 2011) (pieterse, 2012)

Figure 2 The concortium of MYOPRO

1.2 The basic idea

In **Figure 3** are the sub systems what are needed to control and get a myoelectric hand. We are focusing on the mechanics.

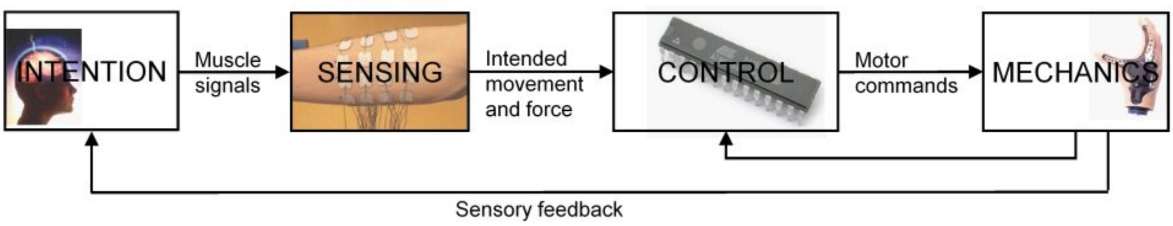


Figure 3 Stages of a myoelectric hand

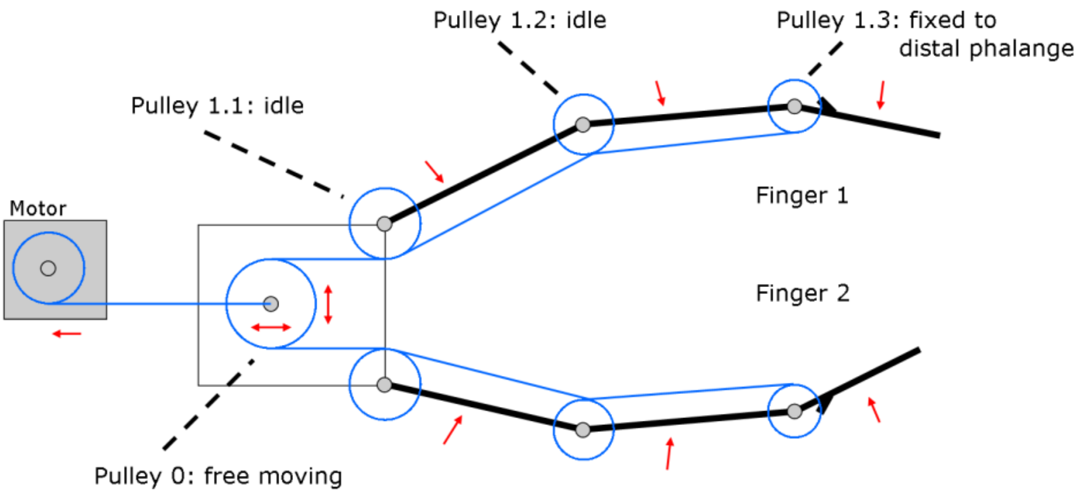


Figure 4 An under actuated 2-fingered hand. A tendon attached to the motor pulls the free moving pulley, located in the box. A second tendon is attached to pulley 1.3, runs over idle pulleys 1.2 and 1.1, over the free moving pulley 0, over idle pulleys 2 Picture and description from Pieterse.

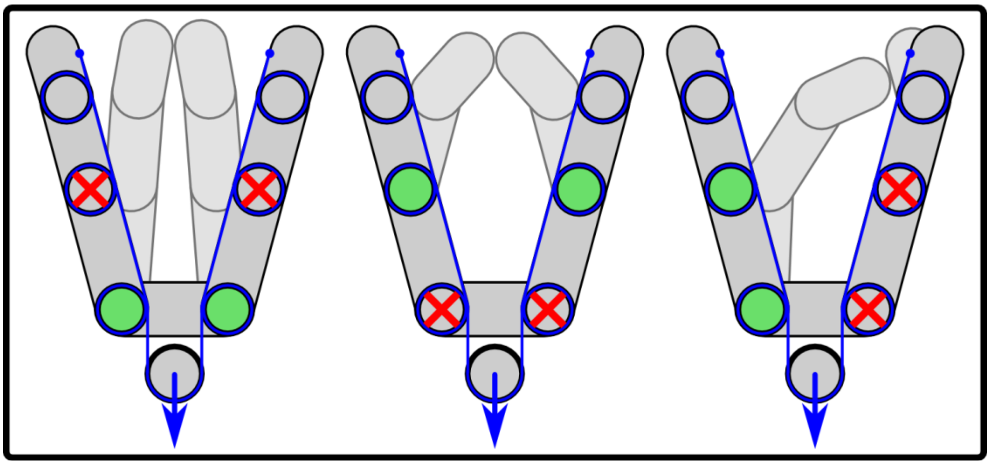


Figure 5 what happens when a joint is locket

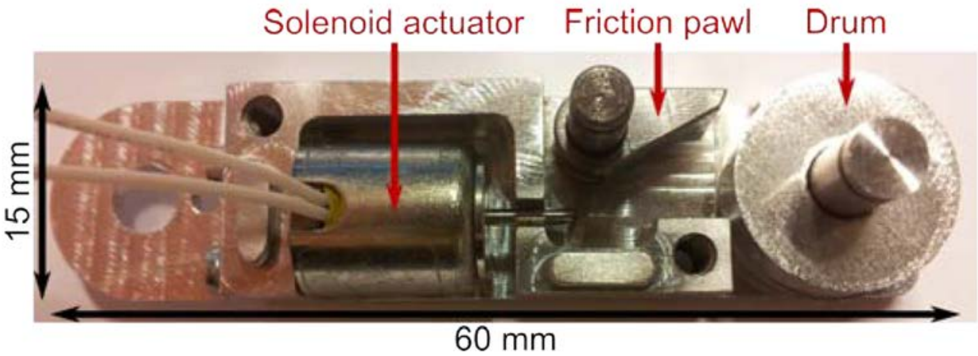
The basic idea is that one motor can actuated the entire hand (a underactuated system). The motor pulls a wire connected to a free moving pulley creating a gripping motion of all the fingers (Figure 4). The drawback of this type of motion is all the fingers move together. You want to control each joint individual to success fully pick up or grab an item. A way to control these joints is to lock them. This enables the fingers to be partly locked and pulled in a way such a desired position can be attended (like Figure 5).

If another way of using the lock is to use is in the other way. When the item is in the grasp of the hand all the joints can be locked and the item can be lifted. In this state large forces can be applied without the hand losing grip. This neither requires a powerful motor or a constantly running motor witch saves energy. And because a smaller motor and battery, cuts the amount of weight and space required. Currently is the not a part of the system.

Existing systems work with a motor in each of the knuckles the drive the motion of each joint, this system weighs more and is less functional than the proposed system.

This project is based on work done by Pieterse (pieterse, 2012). He designed new joint locks and did experimental research on the joint locks. One lock system is selected for further development.

This concept (see Figure 6) is based on friction and has an unlimited resolution. In the concept when locking is needed the pawl is pushed by a solenoid actuator against the drum with a low force. When a torque is applied, the pawl stops the drum turning (something like a wedge). Then the joint is locked. The whole idea is that the locks are self-locking so no force or energy is needed to keep the joint locked. See chapter 1.3 for how the system works.



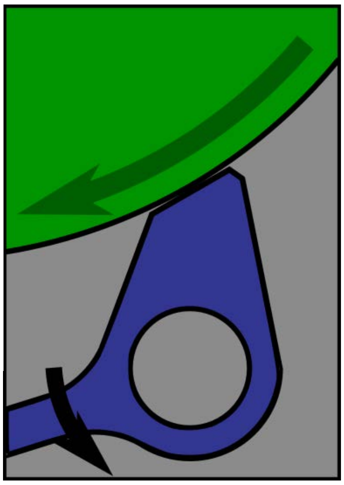


Figure 6 Prototype of the FA joint lock concept with solenoid actuation. (Peerdeman, 2012)

1.3 Analysis how the forces work in the system

One of the project goals is to see how the locking system works. After analysing the system and the way forces are affected within the mechanism. Next is an chronological description of the forces. For more reference see **Figure 11** (**Figure 11** is split in **Figure 10**, **Figure 9**, and **Figure 8**). In **Figure 11** the black circle is the drum and the black line is the pawl. They make contact with each other in D and the E and G are the points for where the shafts for the pawl and drum are.

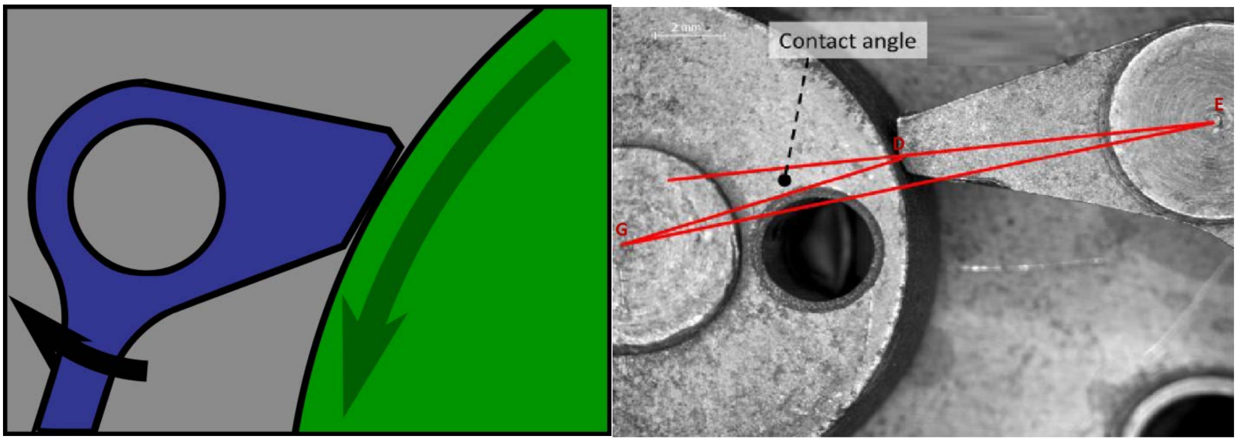


Figure 7 The pawl and drum translated to vls

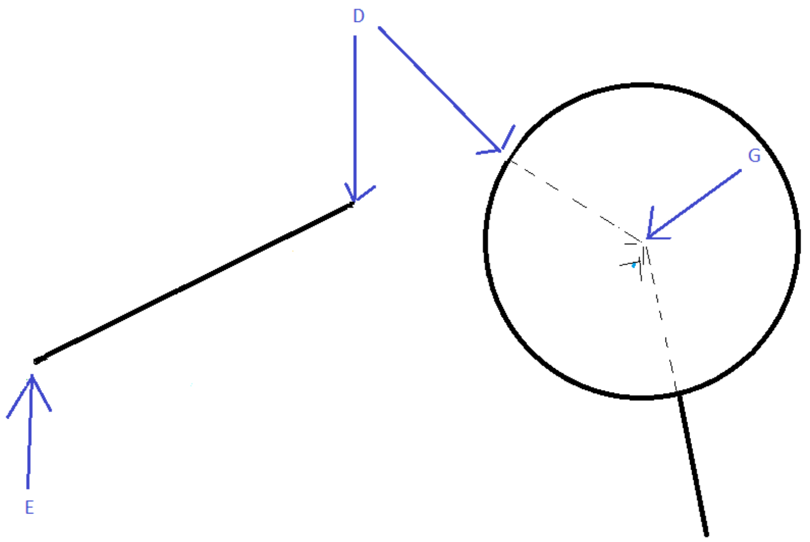


Figure 8 Points

D is the contact point between the drum, pawl G is the shaft of the drum and E is the shaft of E.

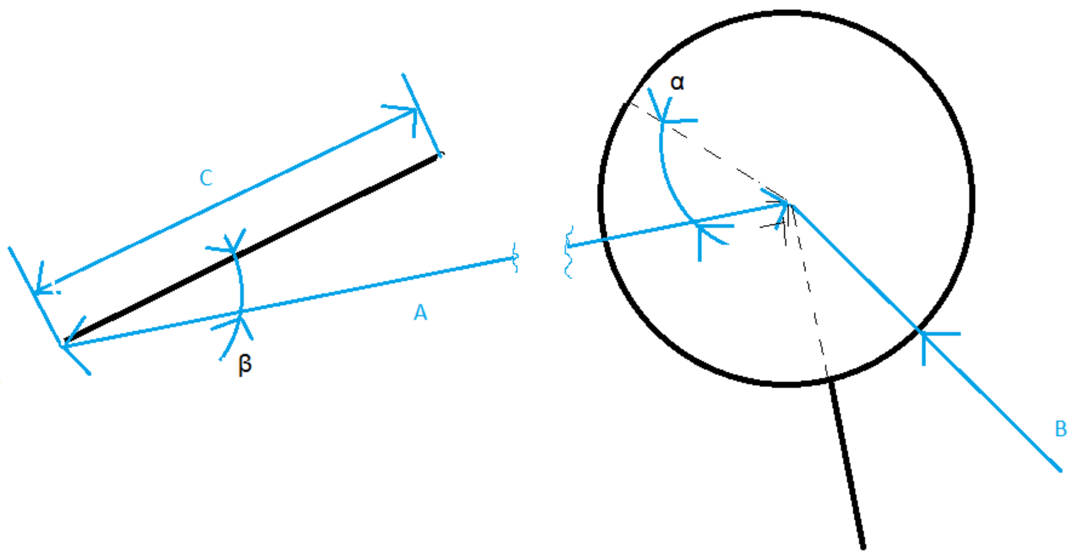


Figure 9 The distances in the system

C is the length of the pawl, distance B is the radius of the drum ,distance A is the distance between the shafts af the drum and pawl

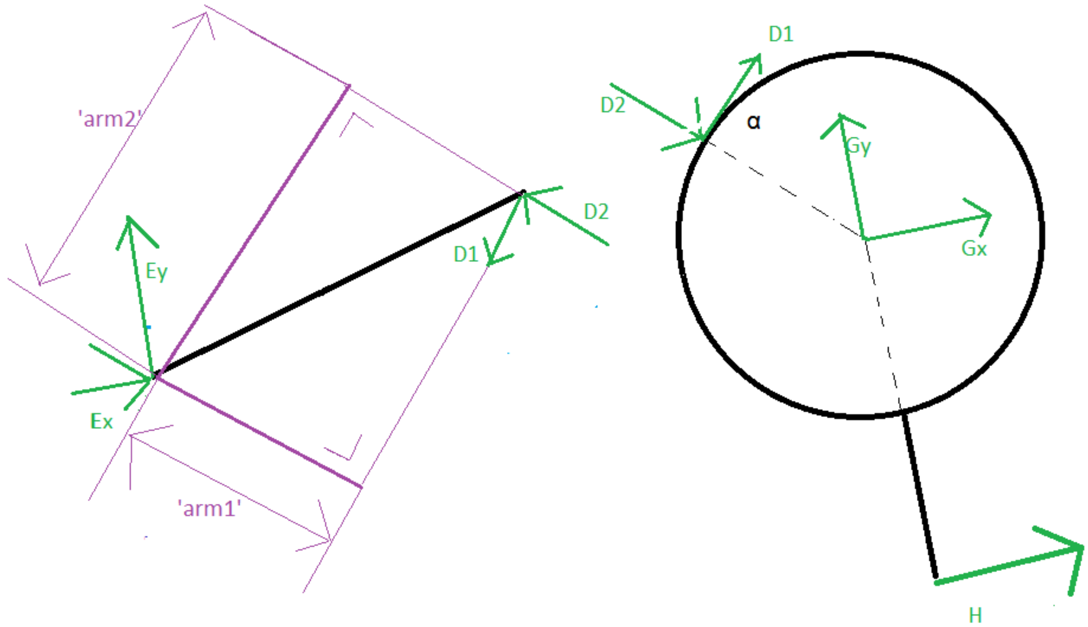


Figure 10 forces in the system,

The forces are green and purple is the "arm1 and arm2". The forces D are the contact forces, G are the forces in the shaft of the drum, E are the forces in the shaft of the pawl and H a force on the drum that creates a torque.

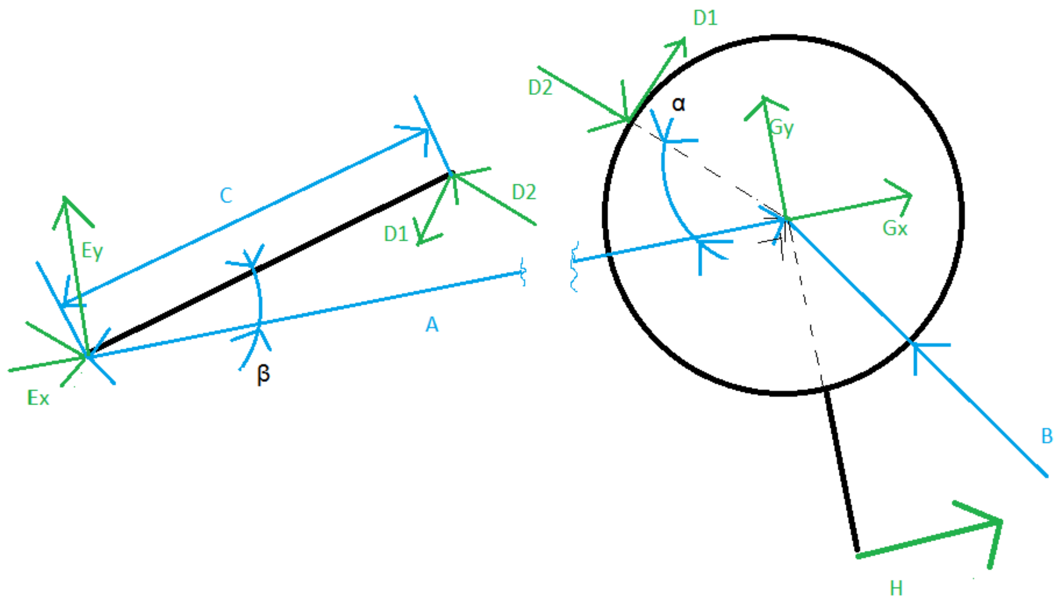


Figure 11 figure VLS from Pawl and Drum

By dynamical analysing the calculation model (chapter Calculation model10.5) and the vls the locking process is analysed.
The locking process:

1. Small normal force D2 applied by the pawl by putting the pawl against the drum.
   a. Assuming there is a normal force that is greater than 0 (this in contrast with the momentum that starts at 0)
2. Small moment on drum because by H causes torque around G
   a. Assuming that the torque (through H around G) starts at 0 and increases to the desired torque*.
3. Friction transfer (D1) from de drum to the pawl
4. Through momentum theorem ("D1*arm1 =D2*arm2" around E) Larger normal force(D2).
5. Moment increases by H around G
   a. On the momentum when the moment increases to quickly then maybe to high contact forces
6. By friction transfer the D1 increases
7. Through momentum theorem (around E) larger normal force (D2)

   a. The system does not slip, assuming “D1=μ*D2” and “D1*arm1=D2*arm2”, then has to “arm2/arm1< μ”

8. Point 5,6 and 7 keep repeating until the contact tension rises to much and “D1= μ*D2” does not apply anymore or the system breaks down (different friction-resistant’s)
   a. By suspending the system when under stress, the forces will change the geometry and hereby the size of the forces.

*In the case when a directly a large torque is used (by H around G) the system will slip however there is still a frictional force being applied that will create a normal force until there is no more slip.

Summary: The position of point A, G and D in respects to one another are very crucial.
In chapter **4 The lock system** is a description of how critical these positions are.

2. Problem analyses

2.1 The problem

Due to several reasons the experimental test result from the existing test rig are unreliable. We want to know how the system exactly works. What makes it self-locking. How high are forces in the system. How does the spiral really works. Where comes the compliance in the system from. What is needed to get a lock reliable working.

2.2 Assignment

The assignment consist of 4 main points.

1. Design and build a test rig for evaluating current and future joint lock components.
2. Design and manufacture several new pawls and drums with increased friction, using different coatings and materials.
3. Take thorough measurements with all pawl/drum combinations, and note any effects of misalignment, deformation, and coating/material of the joint lock components.
4. Provide a recommendations for the joint lock.

2.3 Set of requirements of the hand

The overall requirements of the hand where set in (pieterse, 2012).

1. The hand is able to generate cylindrical and key grasp forces of 70 Newton and a tripod grasp force of 20 Newton.
2. Each finger contains 3 degrees of freedom for flexion and extension: in the DIP, PIP and MCP joints.
3. An under actuated differential system is able to actuate the flexion of finger joints.
4. Springs will allow the extension of the finger joints.
5. Total flexion of all fingers can be done within 1 second.
6. The range of motions for the flexion of the joints is from 0° to 90°.
7. For each joint the rotation can be measured.
8. The fingers have anthropomorphic dimensions.
9. The total weight of the hand is less than 500 grams.

2.4 Set of requirements of the lock

The overall requirements of the lock where set in (pieterse, 2012).

1) Withstand 2.0 Newton meter of torque, 200 Newton of axial force and 20 Newton of radial force.

2) A torque overload protection mechanism is included, which protects the important parts for torques above 6.0 Newton meter.
3) Locking is possible in both directions.
4) Locking and releasing can be done with a single actuator.
5) The lock should have an angular tolerance under 3˚. *
6) Maximum dimensions are comparable with MCP joint dimensions: breadth (medial – lateral direction) 25 mm and depth (palmar – dorsal direction) 30 mm.[14]
7) Favourable dimensions are comparable with PIP joint dimensions: breadth (medial – lateral direction) 18 mm and depth (palmar – dorsal direction) 17 mm.[14]
8) The lock can be operated from -10˚ C to + 40˚ C.

* Note: It is not favourable that the lock has a large angular tolerance, since this can be demonstrated in a displacement of the fingertip. The fingertip may vary for 5 mm max as a result of angular tolerance in a single joint. For the MCP joint, this comes to a maximum angular tolerance of 3°.

3. Test rig

To solve the questions of the lock a test rig needs to be built. The goal was to build a flexible multi usable rig to answer questions from now and in the future.

3.1 Set of requirements test rig

This are the lock test rig requirements:

- Compatible with old lock parts:
  - Drum (15 mm)
  - Large pawls FA4, FA5 and FA6
  - Small pawls (the pawl used in the test setup of Peerdeman)
- Same relative position of drum and pawl
- Bearings:
  - Well aligned
    Preferably the same bearings
  - Different types (roller of plain) of bearings can put in
- High stiffness of the system
- Good visibility at point of contact between the drum and pawl
- Better way of measuring with high accuracy:
  - The distance between the shafts of the drum and pawl
  - The torque on the drum
  - The angle of the arm
  - The angle of the pawl
- Testing how critical certain dimensions are
  - Angle of the drum and pawl (in z-axle/vertical)
  - The distance between drum and pawl
- Dismountable
  - Bearings
  - Bushings
  - Pawls

3.2 The concepts

Different concepts where created and reviewed. A short description is made of the concepts.

To fulfil the requirements a solution have to be found for the next sub systems:

- Shaft assembly drum
- Shaft assembly pawl
- A system for the changeable distance of the shafts
- Change of contact angle
- Measurement of force and angles.

First several design solutions for the main design sub systems are integrated in proto 1, proto1.5, proto 2 and proto 3 to see how they work out. After a review The best solutions are chosen and combined to proto 4. Then proto 4 is reviewed then 5 developed and also reviewed. Proto 6 then made and evaluated to get a producible rig. Proto 7 is created en with a small adjustments produced.
In the next paragraphs the main design differences are explained.

3.2.1 Proto 1

With two different sizes of bearings at the drum shaft. and fully demountable

Pros:
+ Ease of manufacture.
+ Bearings are well placed
+Is a stiff construction
+Fully demountable

Cons:
-Two different bearings

3.2.2 Proto 1.5

Based on proto1 only with two same bearings at the drum shaft (the lower bearing is fitted in a socked)

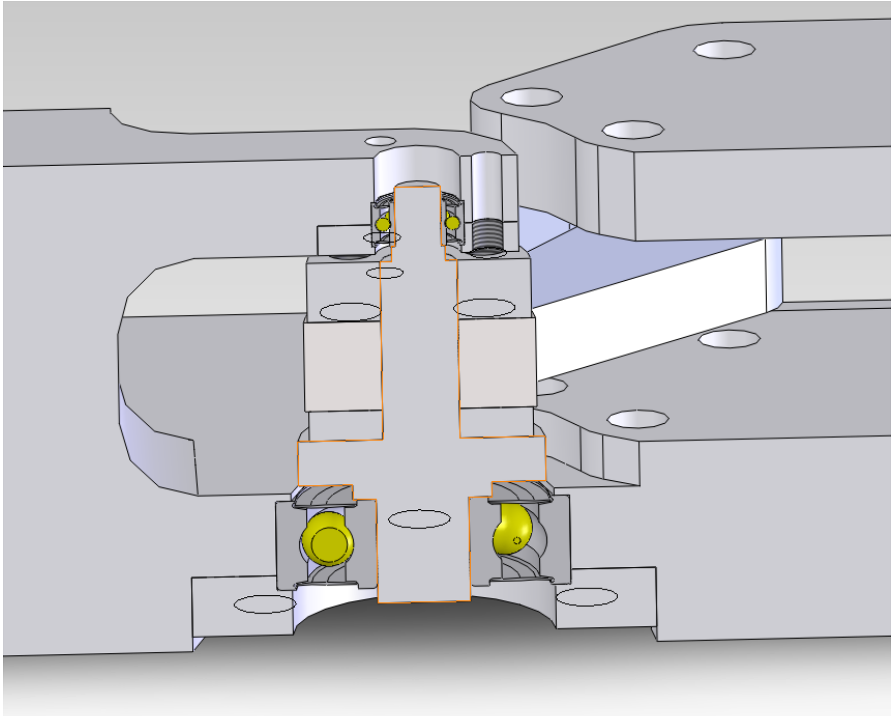


Figure 12 shaft assembly of the shaft of the drum in proto 1.5

Pros:
+ Ease of manufacture.
+same bearings

Cons:
-good alignment of the bearings is difficult

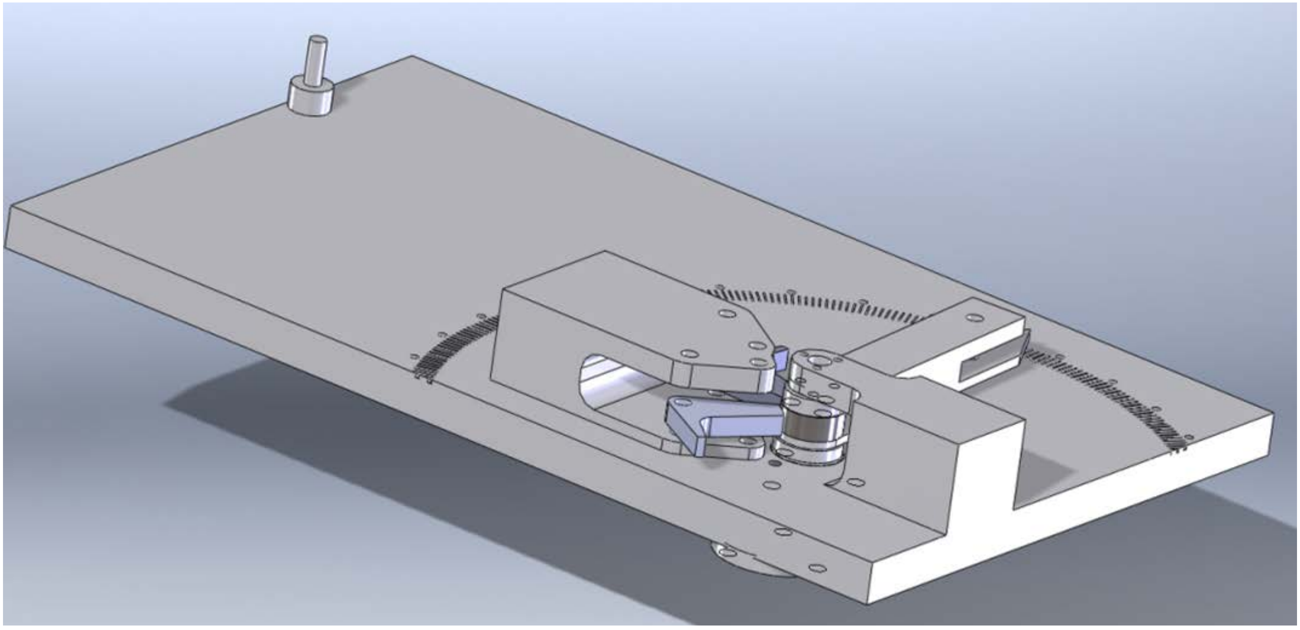


Figure 13 proto 1.5

3.2.3 Proto 2

Proto2 with two same bearings at the drum shaft

Pros:
+good alignment of the bearings
+stiff construction
+ Ease of manufacture.

Cons:
- not fully demountable

3.2.4 Proto 3

The shaft of the drum is longer
Options in this concept are:
Adjustment system is made for bearing alignment (to test angle between pawl and drum on z-axle) and for distance change between pal and drum

Pros:
+same bearings
+ Bearings are well placed
+Is a stiff construction
+fully demountable
+fully demountable
+Adjustment system for distance between pal and drum
+Adjustment system for bearing alignment (to test angle between pawl and drum on z-axle)

Cons:
- More difficult to manufacture
- Difficult view on contact point

3.2.5 Proto 4

After a review:
Proto 4 is an upgrade version op proto 1.5. It is chosen from the 5 concepts for further development. The thrust bearing is replaced by a Teflon bus and the screws for holding the bearings are removed.

Pros:
+same bearings
+ Bearings are well placed
+Is a stiff construction
+fully demountable

Cons:
-a Teflon bearing gives more friction
-the base is too big

3.2.6 Proto 5

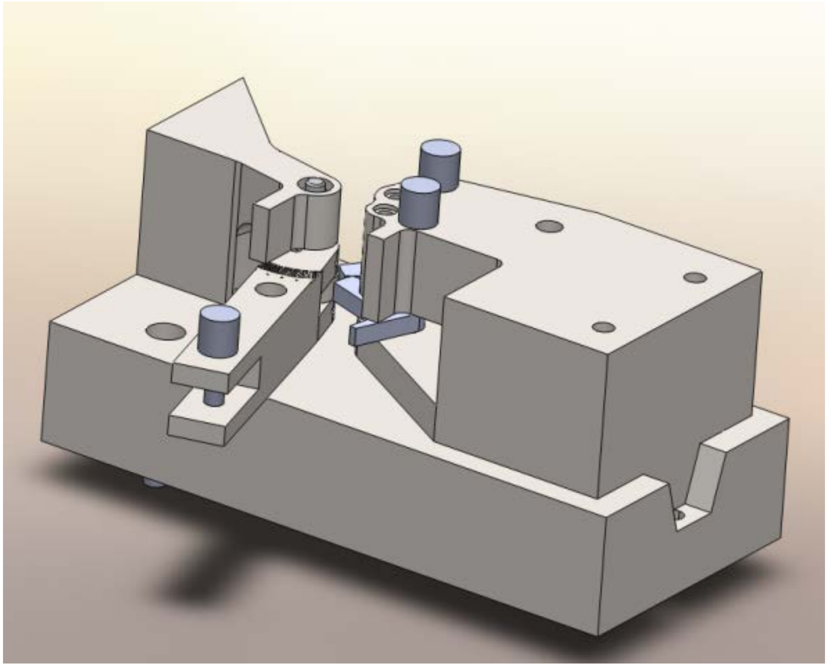


Figure 14 proto 5

Upgrade from version 4 the holder for the pawls is moveable and could be put under an angle.

Pros:
+same bearings
+ Bearings are well placed
+Is a stiff construction
+fully demountable
+Adjustment system for bearing alignment (to test angle between pawl and drum on z-axle)
+Adjustment system for distance between pal and drum

Cons:
-It is possible that the holder get stuck in the slot in the base.

3.2.7 Proto 6

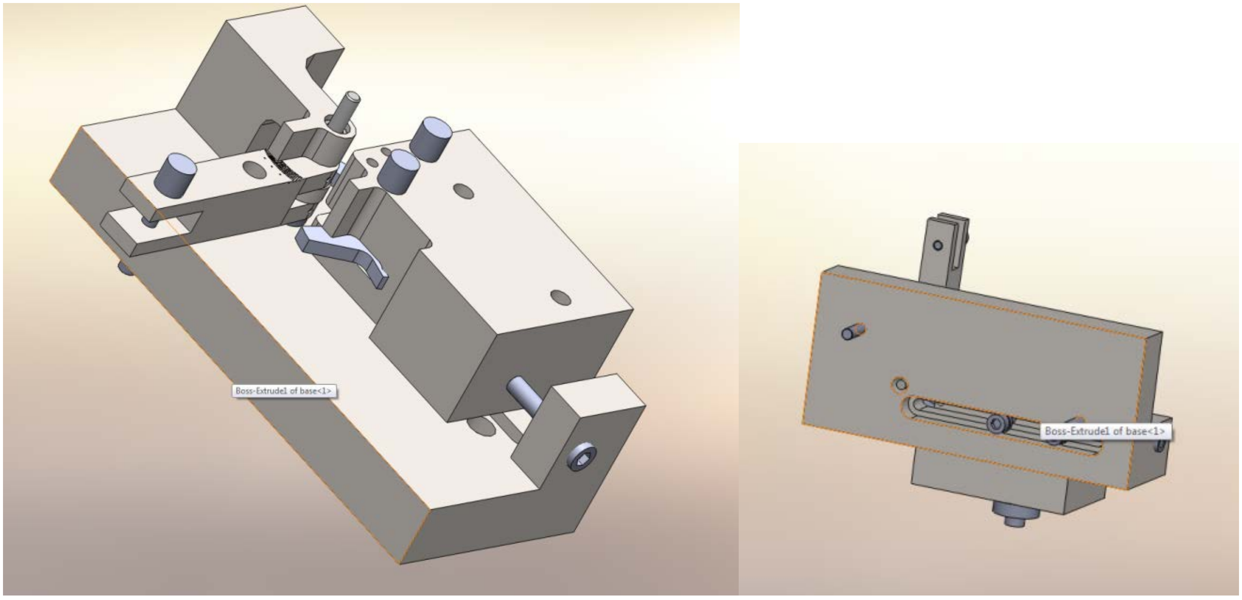


Figure 15 proto 6

Proto 6 has another system of moving the holder

Pros:
+same bearings
+ Bearings are well placed
+Is a stiff construction
+fully demountable
+Adjustment system for bearing alignment (to test angle between pawl and drum on z-axle)
+Adjustment system for distance between pal and drum
+The holder can't get stuck

Cons:
-the displacement of the holder happens by unbolting screws under the system.
-it is expensive/difficult to make it from one piece.

3.2.8 Proto7

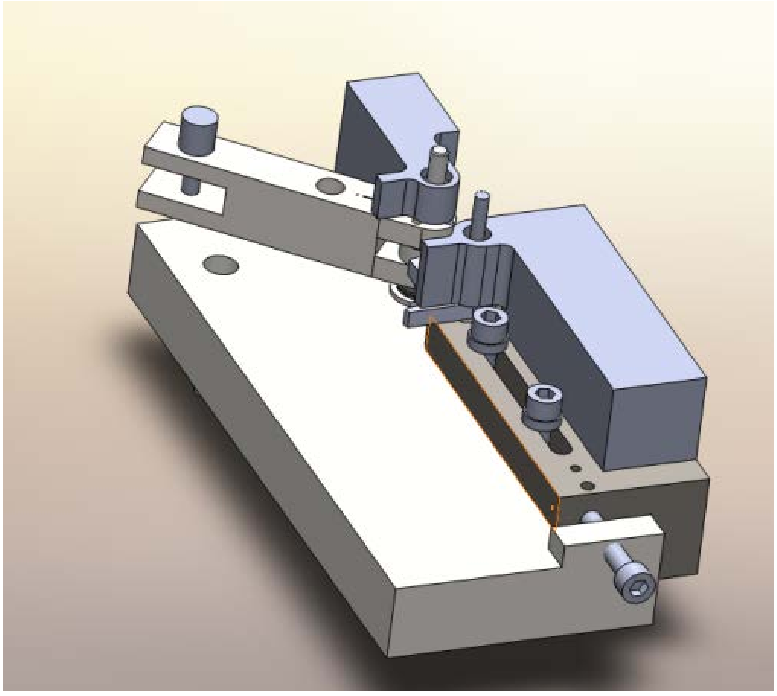


Figure 16 proto 7

The holder and base are split and held together by screws and location pins. The screws who are holding the holder and the base can be unscrewed on the up side. The shaft of the pawl has been fitted with bearings.

Pros:
+same bearings
+ Bearings are well placed
+Is a stiff construction
+fully demountable
+Adjustment system for bearing alignment (to test angle between pawl and drum on z-axle)
+Adjustment system for distance between pal and drum
+The holder can't get stuck

Cons:
-the displacement of the holder happens by unbolting screws under the system.

3.3 Final design

The final design is a tuned version of the proto 7. In chapter **10.2** the 2d drawings are showed.

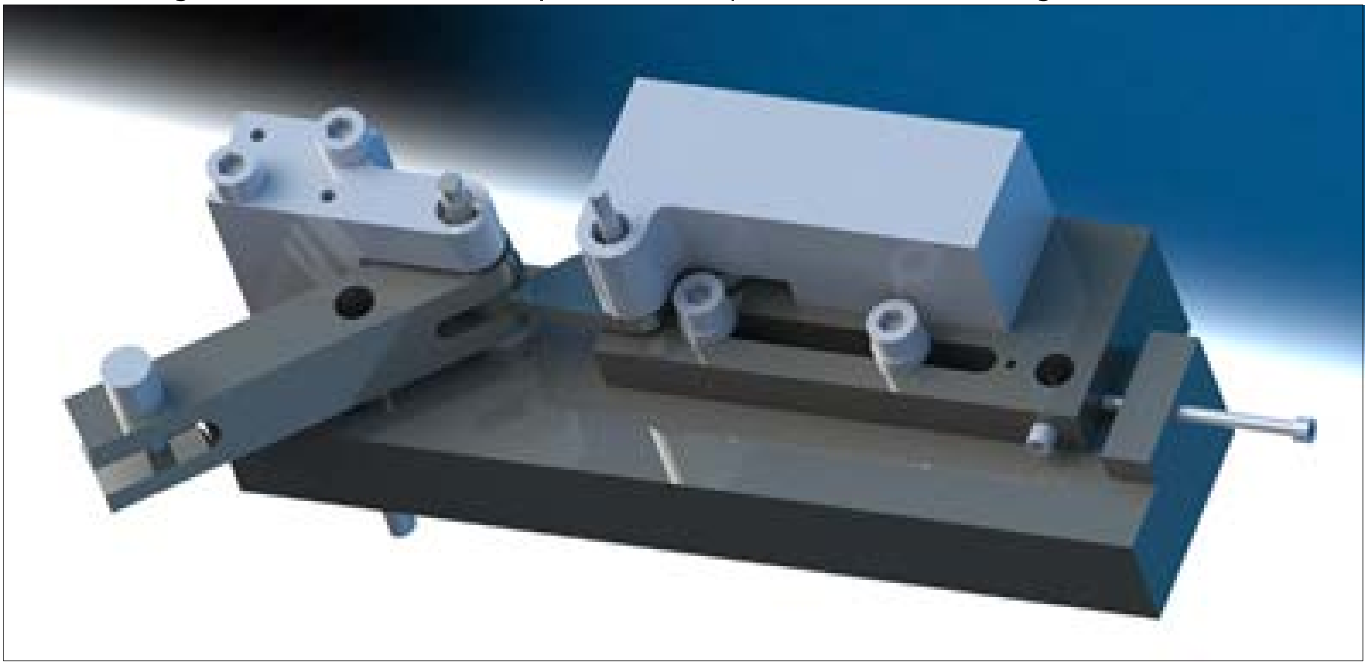


Figure 17 test rig final design

This design was developed for easy manufacture, low production cost, high accuracy, flexibility and this design allows reliable, accurate and diverse testing.

This design fits in perfectly in this stage of the project due to its low costs and easy of testing. Mainly because the project is still in its infancy.

The design can be easily manufactured with low production cost due to relative simplicity of each part. This also means it can be simply produced without many clamping. One of the design features of this design is the high tolerances that can be applied in an easy way. This translates in a high accuracy at a relative low cost. Due to the fact that the design is over dimensioned means that it reliable, stiff (more accurate) and can handle a lot of stress beyond its requirements. Furthermore this design offers a greater range of tests available.

3.3.1 Functionality finial design

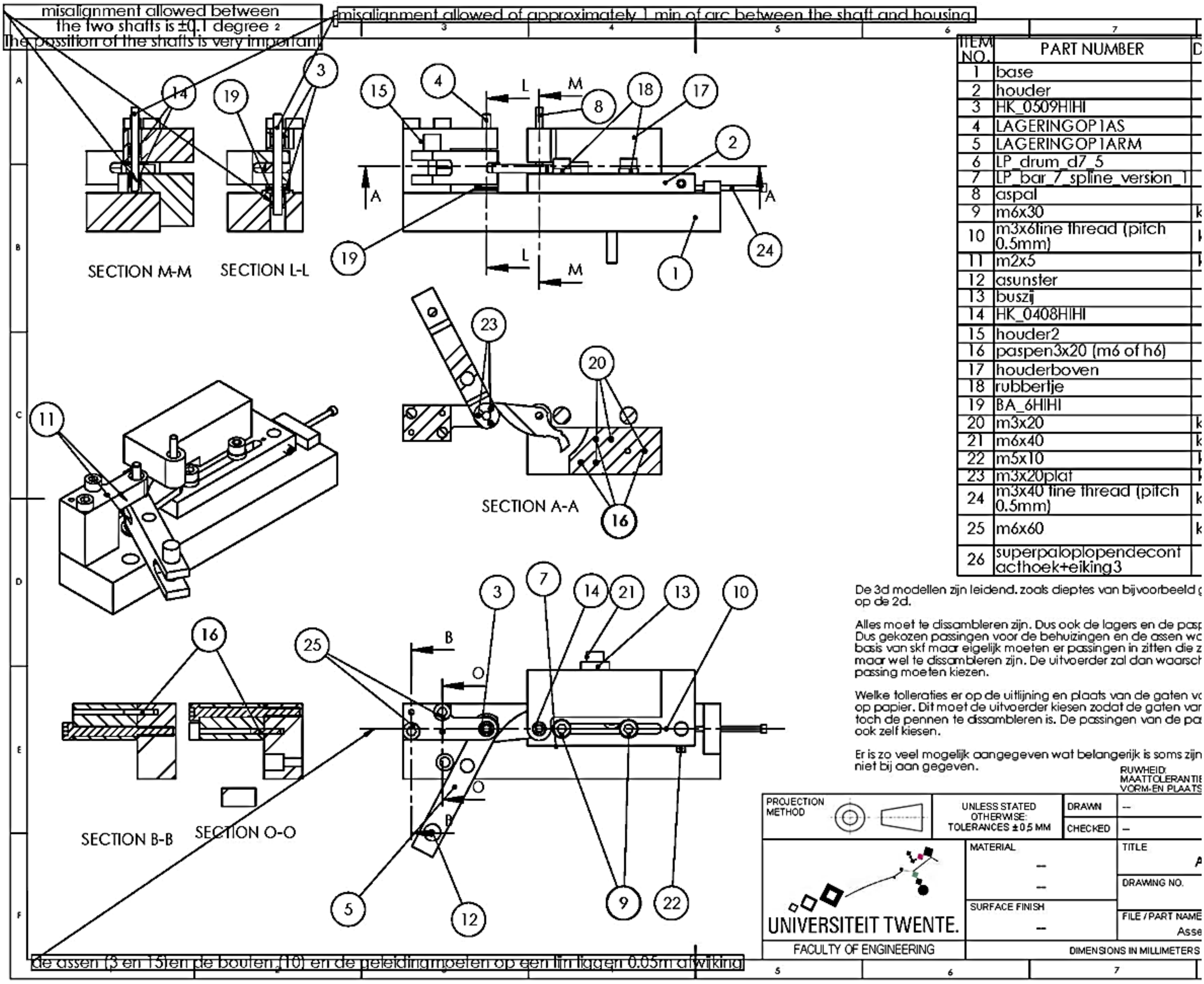


Figure 18 part of assembly drawing test rig

In the system it is possible to use all the pawl who are made for the joint lock and all the different drums. The measurement of the distance between the shaft can be done by a micro meter on the top of the two shafts. Measurement of the angle of the drum and pawl (in z-axle/vertical) can be done by putting a dial Indicator in houder2(fasted by bolt 22). The angle can be adjusted by bolt 10 or by putting plates from known thickness under 'houder2'(number 2).
The angle change of the 'arm'(number 5) can measured by Hall effect meter.
The torque of the drum can be measured by attaching a spring balance to shaft 12.

The bearings can be easily disassembled and assembled because the holes go all the way through. The distance of the shafts can adjusted by unbolting bolts 9 and 21and precisely adjusted by bolt 24.

The ‘houders’ (number, 2, 17 and 15) can be dissembled by unbolting 25 and 20 and then screw down the bolts for pushing the parts from another, see arrows in **Figure 19**.

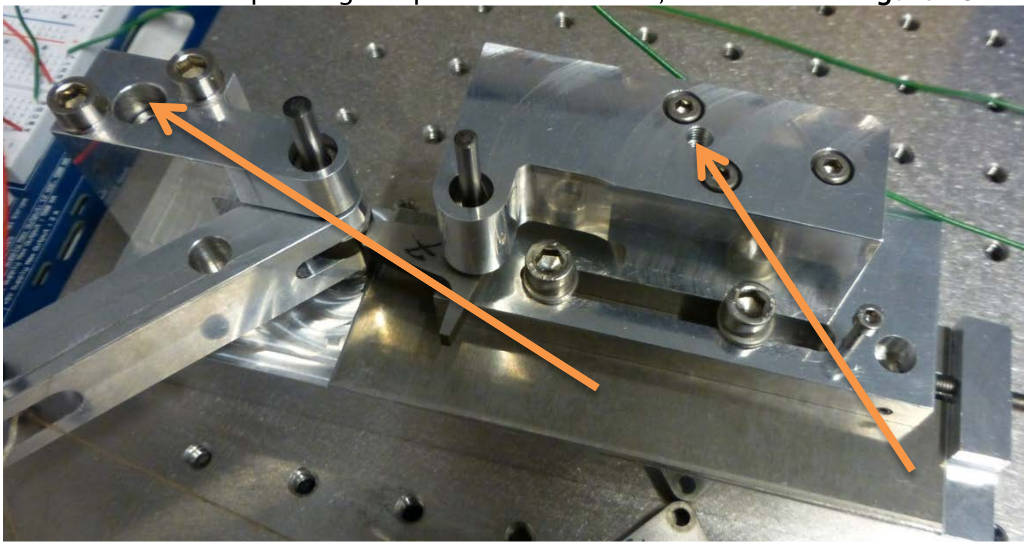


Figure 19 arrows to bolt for dissembling 'houders'

3.3.2 Changes in the production model

The angle of the guidance of base and the houder2 was not 90 degrees. It is repaired with a sheet of aluminium (**Figure 20**).
Bolts for disassembly of the ‘houder’-‘base’ and ‘houder2boven’- ‘houder2’ where not designed but added later by Parts and Tools.
The hole of the upper bearing (part number 13) is bigger, instead of 9mm R6 (for a shaft R6 a tolerance between -16 and -25 micron) the hole is 9,054 mm that is 0.060 mm to big. 8,984 mm is the outer dimensions of the bearing.

Figure 20 the arrow to the repair sheet

3.3.3 Production specs

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	base	maken	1
2	houder	maken	1
3	HK_0509HIHI	lager skf	2
4	LAGERINGOP1AS	maken	1
5	LAGERINGOP1ARM	maken	1
6	LP_drum_d7_5	bestaand	1
7	LP_bar_7_spline_version_1	bestaand	1
8	aspal	maken	1
9	m6x30	koop Din912	4
10	m3x6fine thread (pitch 0.5mm)	koopDin913	1
11	m2x5	koopDin913	2
12	asunster	maken	1
13	buszij	maken?	1
14	HK_0408HIHI	lager skf	2
15	houder2	maken	1
16	paspen3x20 (m6 of h6)	koop	5
17	houderboven	maken	1
18	rubbertje	koop	2
19	BA_6HIHI	lager skf	1
20	m3x20	koop Din912	3
21	m6x40	koop Din912	1
22	m5x10	koopDin913	1
23	m3x20plat	koopDin913	3
24	m3x40 fine thread (pitch 0.5mm)	koop Din912	1
25	m6x60	koop Din912	2
26	superpaloplopendecontacthoek+eiking3	bestaand	1

Figure 21 part list test rig

The parts are mad by Parts and Tools in Enschede.
Part number 1,2,5,15,17 are made from aluminium ansi aw6082.
Part number 4,8 and 12 are shaft made of 100cr6 (wstnr:1,3505) hardened 60+/-2 hrc by din6325 (iso 8734A).

For the used tolerances and 2d drawings see chapter: **10.2 2d drawings test rig**

3.4 Design bracket for support camera

For the test rig is a bracket needed for support of the camera. The camera measured the contact angle of the drum and pawl. This is done by making a picture when the pawl makes contact with the drum. Di picture is corrected on the lens and correct on perspective and then the angles is measured with the position of the shafts and contact point.

3.4.1 The requirements for the bracket:

The requirements for the bracket are:

- The bracket is design for the camera Panasonic Lumix DMC-TZ30.
- Every time the bracket is used the camera is at the same place in relation to the test rig. (stiff construction)
  - o The system has to be stiff, the maximum displacement of (theoretical in Solidworks simulations) 0.40 mm
- The camera has to be 10 cm above the contact point on the line between the two shafts.
- The bracket has to be design for a weight of 500 gram (camera is total 206 gram)
  - o In case of using 3d print material. The maximum tensile strength is 10 times lower than specified maximum tensile strength from the supplier. (because the material has a lot of creep)

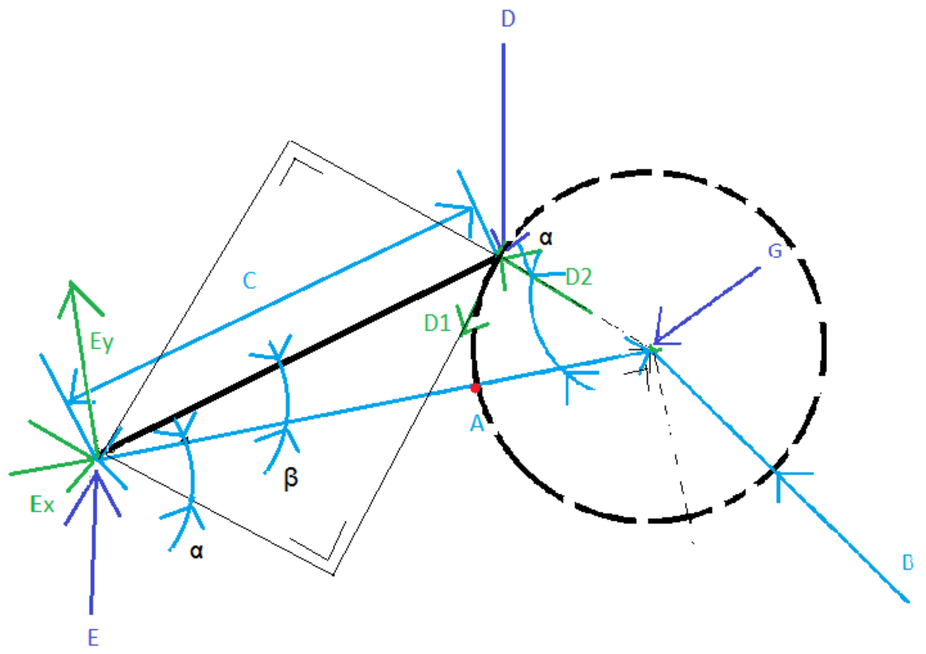


Figure 22 the lens of the camera is exactly aliened above the red dot

3.4.2 Concepts

First concept is to build a bracket from lab system material.
Pros :A stiff construction possible.
Cons: It is difficult the acquire the building material and get parts made.

Second concept is to 3d print the bracket.

Pros : The shapes are unlimited, An relative fast production time
Cons: The material has on a long term a lot of creep.

The design was created by first drawing with the right dimensions the bracket of the camera, on 'houder' and the bracket on the system board. There were different solutions to get things supported. Supporting by truss structure, by pipe construction and a shape that follows from the lower to the upper shape of the construction.
The first 3d print concept was with the shape that follows from the lower to the upper shape of the construction, the construction didn't work with Solidworks and 3d print. Second 3d print concept (**Figure 23**) is a combination of truss and pipe also the design is hollow with ribs. The final design (**Figure 24**) is only with truss structure and reinforced bracket for the camera and the design is solid. The final design has to be solid and without the pipe because of the use support martial of the 3d printer.

3.4.3 The final design

The final design is calculated thru with Solidworks simulation. As showed in **Figure 25** the model is fixture on the bottom and a force (5 newton) is added on the top surface of the camera holder. The results are an maximum stress of Misses of 0.8 n/mm^2 and a displacement of 0.280mm. that are acceptable results

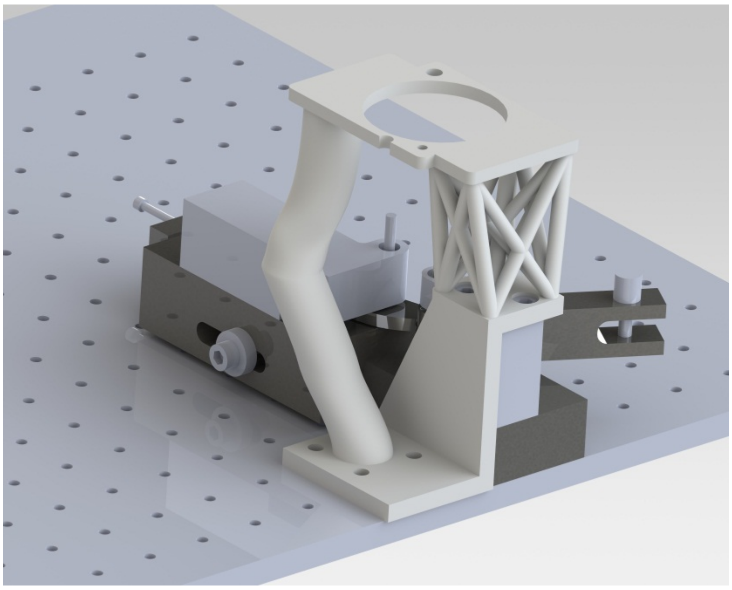


Figure 23 second concept bracket

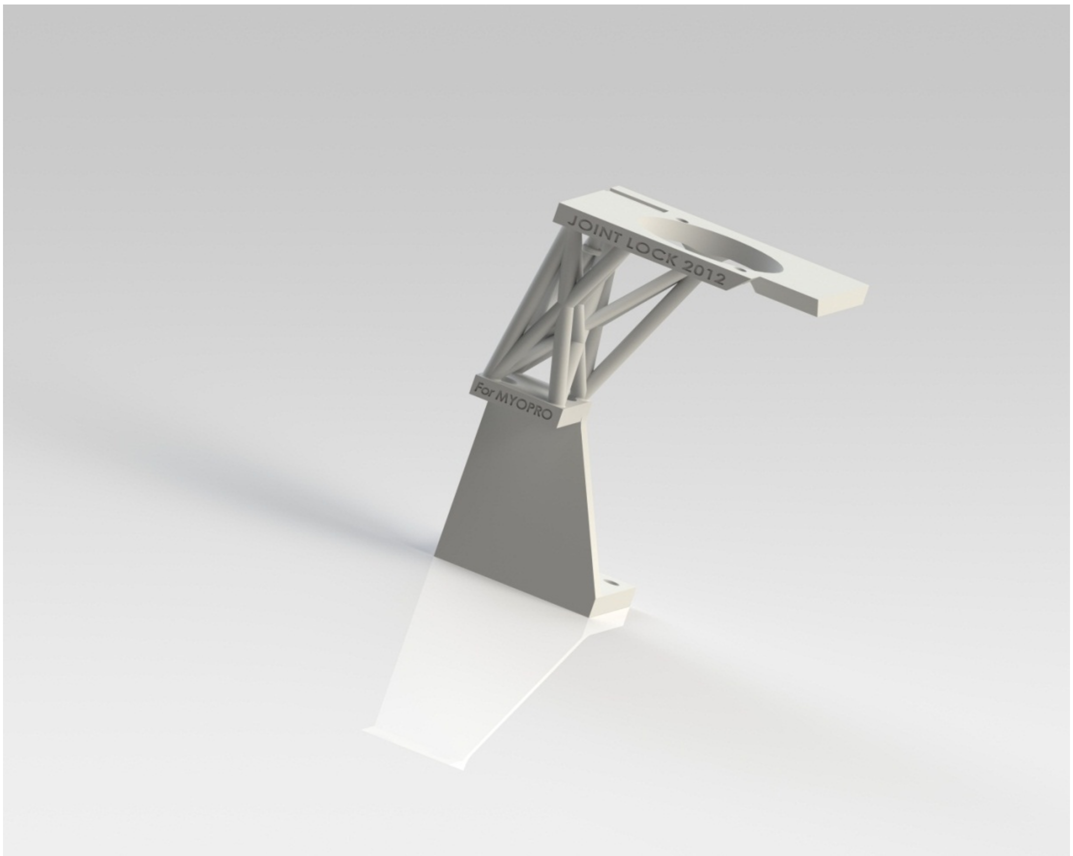


Figure 24 the final second concept

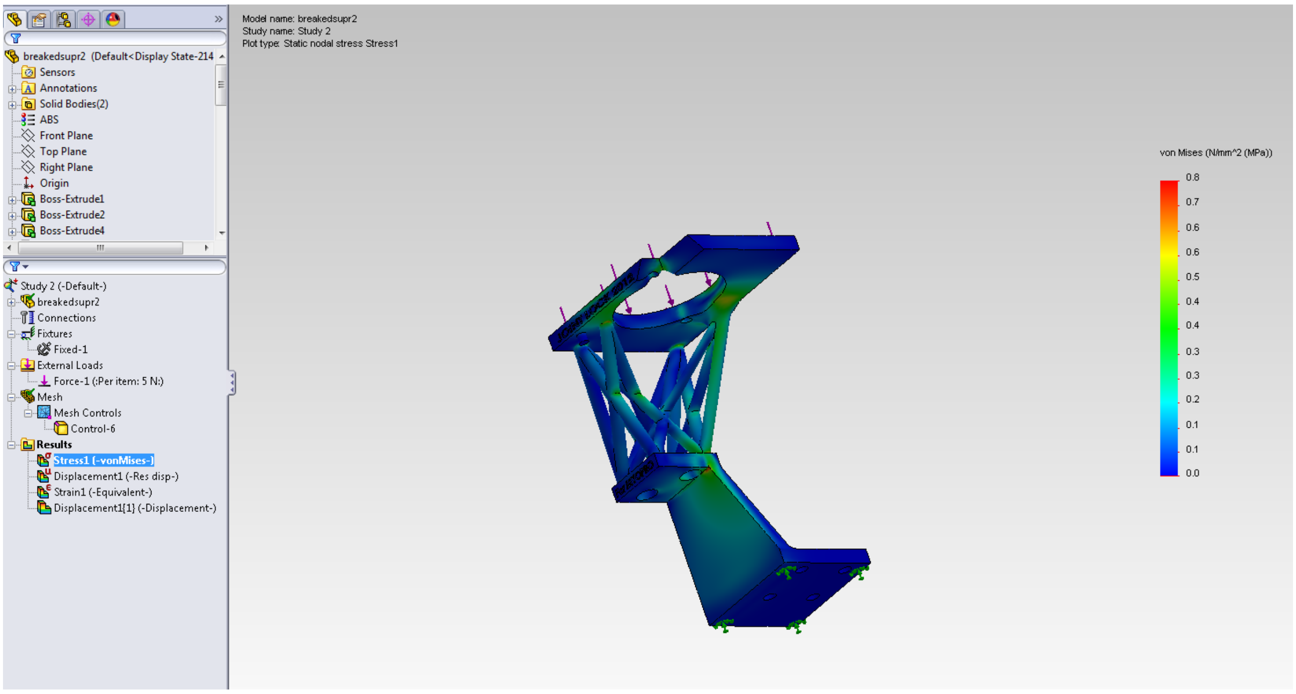


Figure 25 the stress in camera bracket

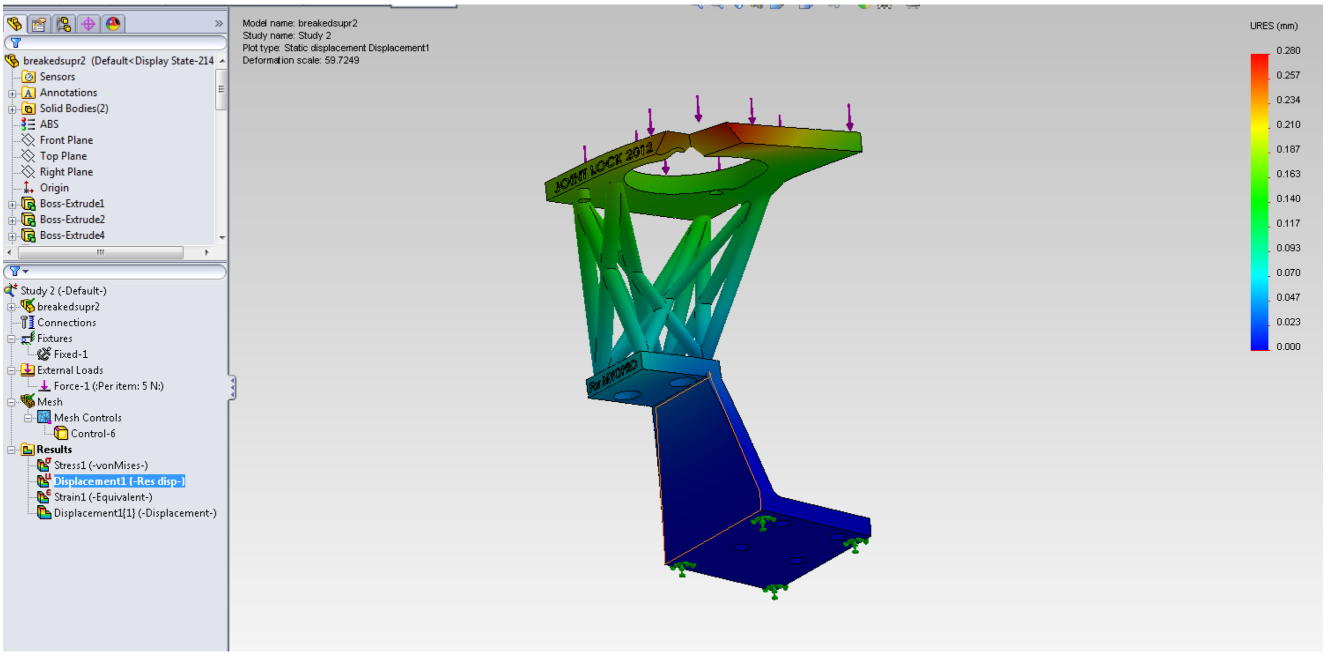


Figure 26 displacement in camera bracket

4. The lock system

In this chapter is describes the different parts of the lock system and the results of the model behind the lock.

4.1.1 Calculation model

A calculation model was needed to calculate static the forces, needed friction and contact tensions in the system for different situations. The model is made in excel. (The explanations of the formulas are in the model called 'calcuationmodeljoinlock2.2' and also in chapter: **10.5 Calculation model**)

reactie	hoe berekent/ uitleg			in mm or N	in mm or N	in m or n or Nm
Constanten:				ingetypt	gerekent	gebruikt
μ		=	tan(D1/D2)	0,120	0,09	0,120000
arm H		=		100,00		0,100000
hoeken nieuw						
A		=		30,27	30,27	0,030270
B		=		7,50		0,007500
Hoek pal		=		7,00		7,000000
γ		=			173,00	173,000000
β		=	sinus regel		1,73	1,730342
α		=			5,27	5,269658
C		=			22,8121	0,022812
radius spiraal		=	bij benadering		37,50	0,037500
contact lengte		=		5,00		0,005000
v1		=	poisson ratio pawl	0,30		0,300000
v2		=	poisson ratio bus	0,30		0,300000
e1		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
e2		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
			Reduce rol radius		0,01	0,006250
			reduced Young's modulus		230769230769,23	230769230769,231000
Krachten						
H		=		20,00		20,000000
Momei	moment veroorzaakt door H om G	=	H*"arm H"		2,00	2,000000
D1	krachten plaatje bus	=	"moment H"/B		266,67	266,666667
D2	krachten plaatje pal met de momenten stelling met het moment om E z	=	D1*((C*SIN(90-α-β)/(A*SIN(α)))		2171,83	2171,825714
Ex	krachten plaatje pal in krachten in x richting	=	D2*cos(α)+D1sin(α)		2187,14	2187,137977
Ey	krachten plaatje pal krachten in y richting	=	D1cos(α)-D2sin(α)		66,07	66,072019
Ex	via momenten om snijpunt Ey en D1 in krachten plaatje pal	=	D1*((A*(COS(α))-B)/(A*SIN(α))		2171,83	2171,825714
Ey	via momenten om snijpunt Ex en D1 in krachten plaatje pal	=	(D1*B)/A		66,07	66,072019
Ex	via hoek beta	=	Ey/TAN(β/180*PI())		2187,14	2187,137977
Ey	via hoek beta	=	Ex*TAN(β)		66,07	66,072019
E	krachten plaatje bus in krachten in x richting	=	(D1^2+D2^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2172,83	2172,830514
E	krachten plaatje bus krachten in y richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
β(controle))		=	tan^-1(Ey/Ex)		1,74	1,742534
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
Gx		=	-H-D1sin(α)-D2cos(α)		-2207,14	-2207,137977
Gy		=	-D1cos(α)+D2sin(α)		-66,07	-66,072019
G		=	(Gx^2+Gy^2)^0,5		2208,13	2208,126709
locking						
μ		=			0,12	0,120000
		=	D1/D2		0,12278	0,122785
Wmax		=			260,62	260,619086
is er zelf locking		=			0,00	0,000000
herts contact spanning(n/mm)		=	(((D2/"contact lengt")*: Reduced Young's modules")/(2*pi()*"reduced rol radius"))^0,5		1597,67	1597,667834
The elatic foudation model tension						
						0,000000
e3		=	elastic moduli rubber layer (n/m^2)	100000000,00		100000000,000000
h	thichnes rubber	=		0,10		0,000100
a	Halfcontact width		((3*D2*'radius spiraal'*h)/e3)^(1/3)		0,6252	0,000625
elastic foundation contact spanning(n/mm)		=	(e3/(2*'radius spiraal'*h))*((a^2)+('contact lengte'^2))		338,5443652	338,544365

Figure 27 Picture calculation model

4.1.2 Result calculation model

In the model is tested what the influence theoretical is from different dimension of the lock on the forces in the system.
We see that changing A or C only the reaction forces in E and G change in the x and y direction, the resulting force stays the same.
When the contact angle is changed with x2 the contact force D2 approximately x0.5 also the Herts contact stress reduces. That is seen in **Figure 73** and **Figure 29**. When the contact angle doubles the needed friction coefficient for self-locking becomes doubles too (see **Figure 30**).
In **Figure 74** and **Figure 31** we see that when distance B (the radius of the drums) changed with x2 the contact force D2 also approximately x0.5 also the Herts contact stress reduces.
When the contact length doubles the Herts contact stress is cut in half, see in **Figure 32.**

Conclusion with chancing distance B, the contact length and the contact angle can be influence greatly.

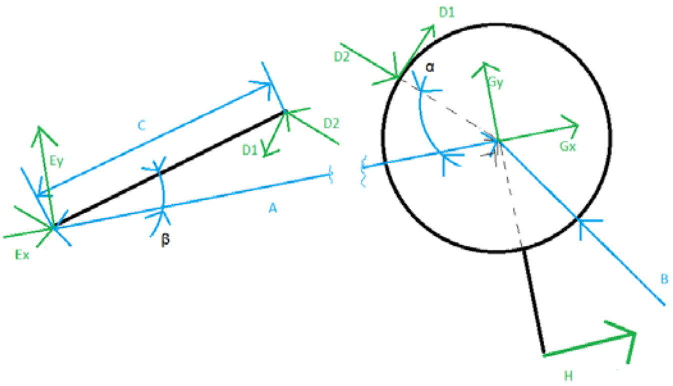


Figure 28 figure two VLS from Pawl and Drum

In **Figure 28** the black circle is the drum and the black line is the pawl. They make contact with each other in D and the E and G are the points for where the shafts for the pawl and drum are.

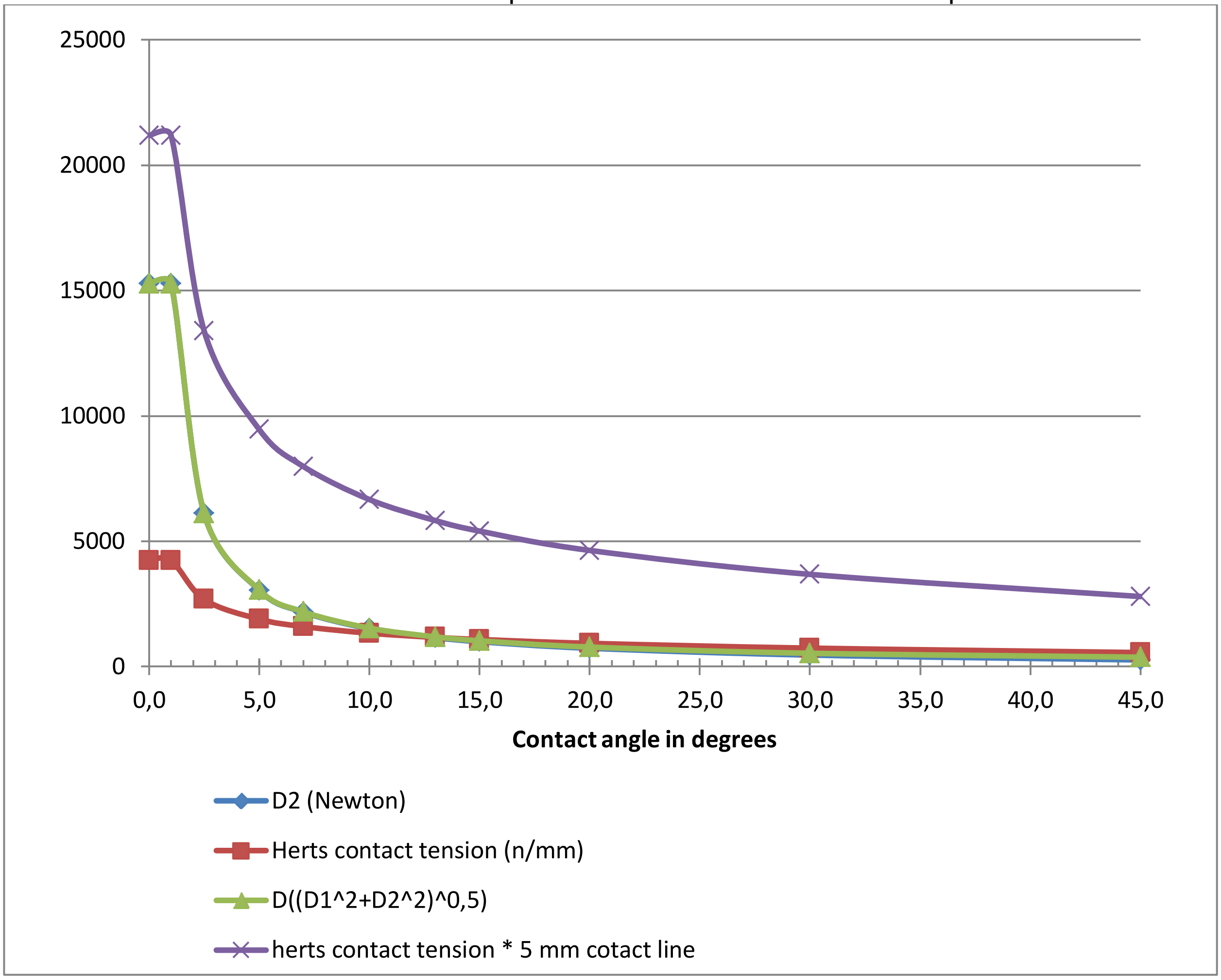


Figure 29 the effects of changing the contact angle on: the contact forces.

Contact angle (α) (degrees)	Needed friction coefficient for self-locking
0,0	0,02
1,0	0,02
2,5	0,04
5,0	0,09
7,0	0,12
10,0	0,18
13,0	0,23
15,0	0,27
20,0	0,36
30,0	0,58
45,0	1,00

Figure 30 table results needed friction coefficient changing contact angle

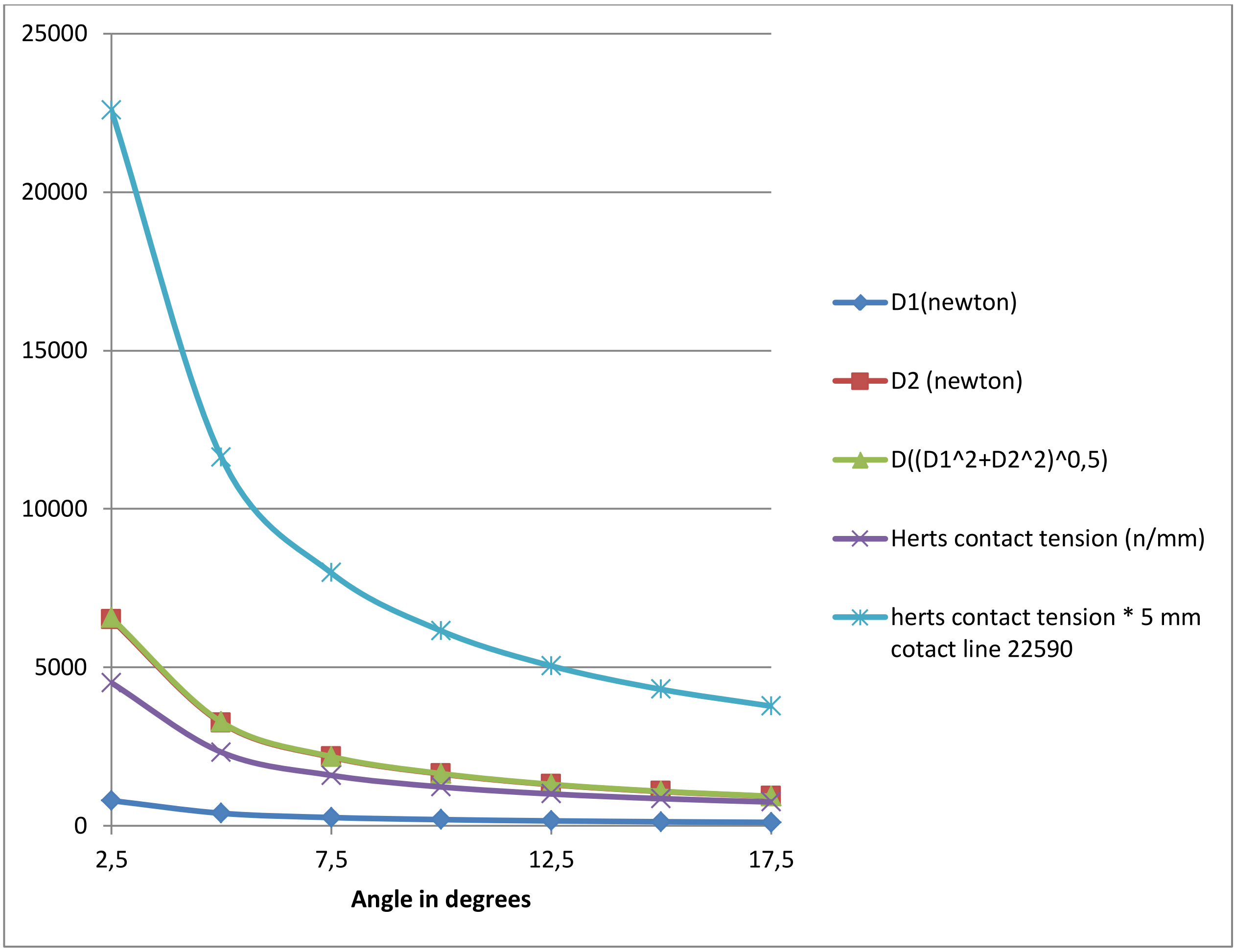


Figure 31 graph results changing B

Line contact length	Herts contact tension (n/mm)
2,5	2259
5	1597
7,5	1304
10	1129

Figure 32 table results changing line contact with A(mm)=30,27, B(mm)=7,5 and α(degrees)=7

Legend: Light grey are fixed input values and dark grey is change in put values and whit are the calculated values.

4.2 Analysing compliance in the system

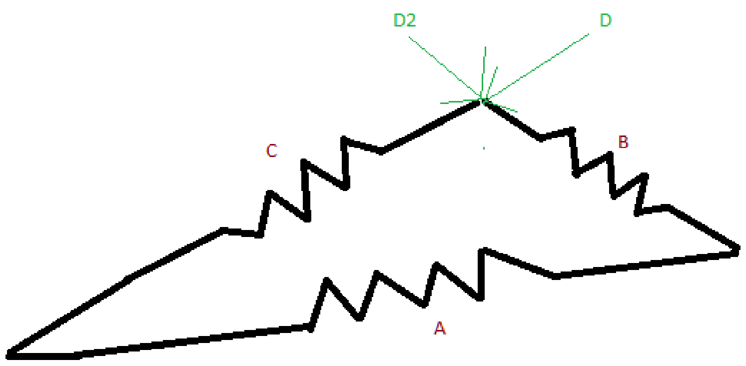


Figure 33 compliance

One of the mayor challenges in the system is compliance . The problem with compliance is that the dimension of the systems change and the forces so the place of the pawl on the drum change. This can be compensated by the spiral scape on the pawl. While the pawl compensates the displacement the drum and pawl are rolling to each other so while locking the phalange change in angle.
The test rig is made so stiff as possible and still angle change of the phalanges happens. Our expectation is that with respect to the stiffness of the rest of the system, the largest part of the compliances comes from the (needle) bearings. By using stiff plain bearings (from example brass) from high accuracy with very low play the biggest compliance problem would be solved.

4.3 Pawl

The pawl blocks the drum when the pawl has the right properties.

4.3.1.1 Why doesn't a flat contact surface work

In a basis the locking mechanism is built with a pawl with a flat contact surface. From testing performed by Pieterse it appeared unsuccessful. The answer on the question "why?" was found in a solid works analyses as described next.

A flat surface does not work because even a difference in A (distance between the axis) of 0,05 mm was enough to change the contact angle substantially. We tested a small pawl with a distance A 12.5 mm and fixed contact angle of 7 degrees(the dimensions of the pawl and the drum stays the same). When A changes to 12.480 mm the angle changes to 0 degrees instead of the desired 7 degrees. When A changes to 12.55 mm contact angle is 14.35 degrees. See illustration for details. We also see in the model that the bigger the pawl is des the lesser the influence of a small distance change of A is. We also see that the changes of the fixed contact angle of the pawl has no big influence.

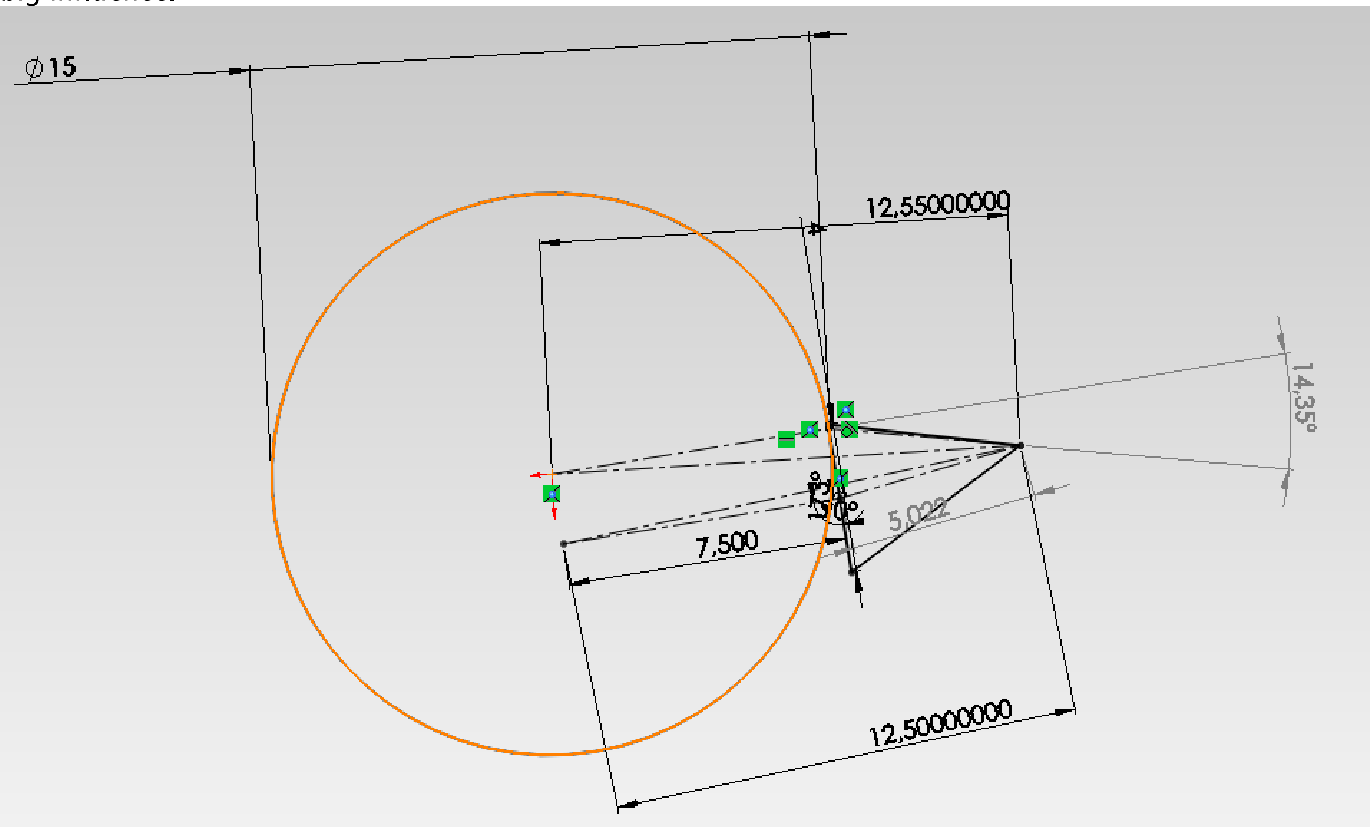


Figure 34 flat surface with A=12.55

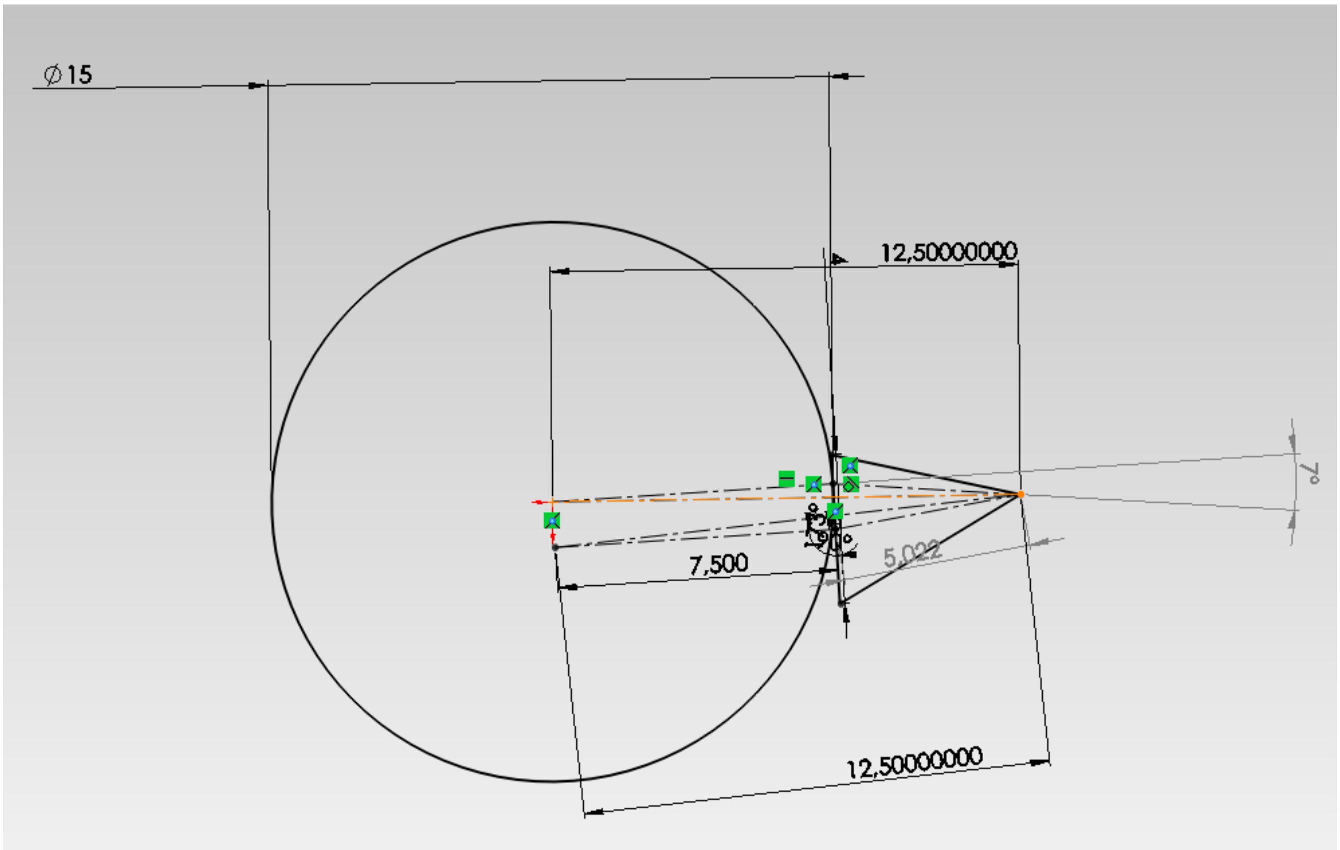


Figure 35 flat surface with A=12.50

4.3.2 Logarithmic Spiral solution

The idea of using a logarithmic spiral for the contact surface of the pawl is even if the distance of the shafts change or other distance deviation the spiral keeps the contact angle the same.

4.3.3 Principle of a logarithmic spiral

A explanation of the basics of a spiral.

The basic formula:
$r = a * \epsilon^{b*\theta}$

a =begin value of a spiral
b=is the steepness of a spiral radius
ϵ= Euler
r= is the distance from zero
θ= is the angle in radians

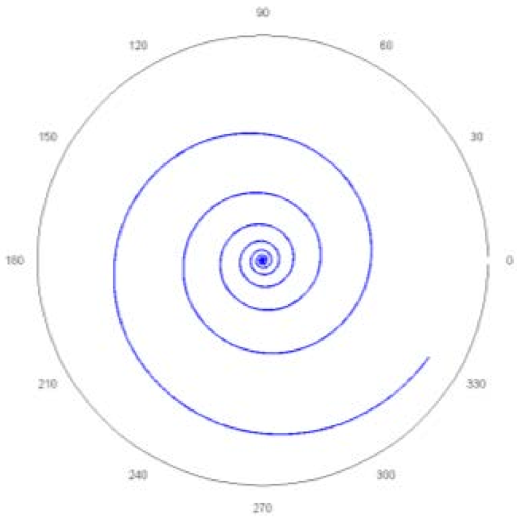


Figure 36 a spiral a = 0.1 and b = 0.1 with ϴ ranging from 0 and 100 radians

4.3.4 The previous pawls

Previous work is done on the pawls by Gert-Jan Pieterse (pieterse, 2012).

Name	
FA 1	The FA 1 model uses a drum with a 15 mm diameter and a push pawl with a contact angle of 18°. The contact area is flat. The pawl is 8.5 mm long and made with wire spark eroding. The other parts are made with milling and turning.
FA 2	The FA 2 model is similar to the FA 1 model and uses the same drum, but is using a contact angle of 24°. This would require a higher friction coefficient but will also give a lower normal force.
FA 3	The FA 3 model is made from the parts of the FA 1 model. The drum and push pawls are sand blasted for increased friction coefficient. The contact angle is reduced to between 6° and 10° to allow for an even lower friction coefficient.
FA 4 FA 5 FA 6	The FA 4 to FA 6 models are renewed designs, based on experiences with the FA 1 to FA 3 models. The drum is renewed but again with a 15 mm diameter and hardened. The push pawls are longer, 12.8 mm, which means that play in the mechanism will have less influence on the rotation of the pawls while engaged. The contact area is curved based on a logarithmic spiral. The contact angle is 7, 10 and 13 degrees. The production method is probably (it is not recorded) first milled then hardened then only the contact area is wire spark eroded.

Figure 37 Pawl information (copy from (pieterse, 2012), page 31 with added information)

With the FA 4 to FA 6 models are some problems. Which are not mentioned. There are some problems with the spiral contact area.

In a analyses of the 3d model which is used for the 7 degree contact angle pawl several things are discovered. When the real contact angles is measured we discovered that the contact angle isn't 7 degree but around 6 degrees(at θ is 10° the contact angle is 5.83 degrees and at θ is -6° the contact angle is 6.03dergeers). In the next paragraphs this is explained.

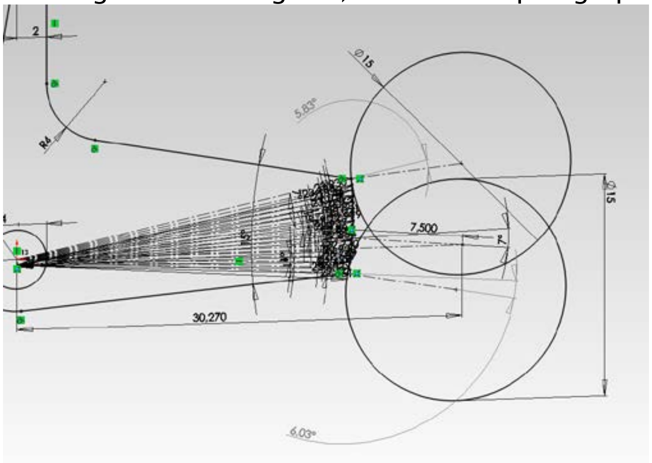


Figure 38 measurement of contact angles at a radius of 10 and -6

At first a point on as θ is 0 was drawn with the next formula: $r=a*e^{\wedge}(\alpha*\theta)$ (the formula is in radian) Pieterse assumed that because the contact angles are small $\tan(\alpha)=\alpha$. However for large angle the deviations from the real value are significant. So $\tan(\alpha)\neq\alpha$
Calculation example:

7 degrees:
$7 / 180 * \pi=0{,}122173$(radians)
$\tan(0{,}1221732)=0{,}122784$ (radians)

45 degrees:
$45 / 180 * \pi=0{,}7853981$(radians)
$\tan(0{,}7853981)=1$ (radians)

In a Solidworks model where the spiral is drawn by the formula $r=a*e^{\wedge}(\alpha*\theta)$ we see only because of that at $\theta=10°$ or $\theta=-6°$ a contact angle deviation of 0.2 degrees. The proposal Is to use the formula $r=a*e^{\wedge}(\tan(\alpha)*\theta)$ (the formula is in radian).

When the point at $\theta=0$ is drawn, we see by reconstruction that is it assumed that every $\triangle\theta=1°$ the distance C (the distance between the contact point and pawl shaft) will change 0.042 mm. That is not true. At $\theta=10°$ or $\theta=-6°$ we see a distance deviation of C of 0.1 mm.

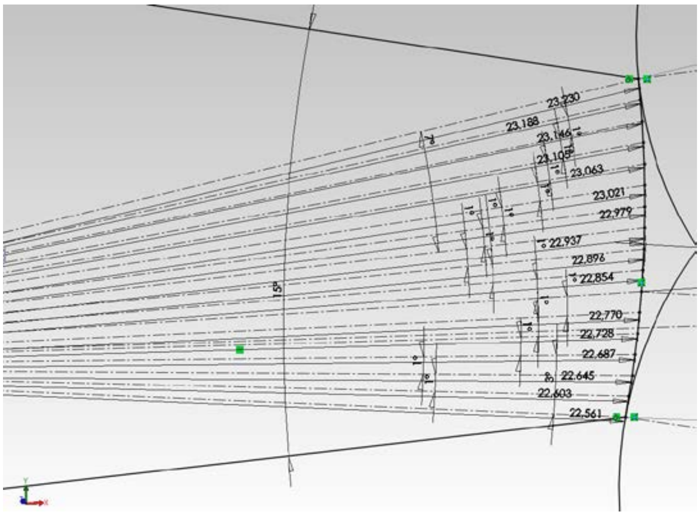


Figure 39 distance change of C in pawl FA 4

After that a circle is drawn approximately through the points which are at $\triangle\theta=1°$. It is approximated so it is a deviation.
All these 3d drawn flaws can be solved by entering directly in the 3d model. In Solidworks the function is called: Equation driven curve.

Other deviations are deviations in the wire spark machine and the hardening process of the pawls.

From measurements is in the 3d models of the big 7 degree pawl (FA 6) and the small 7 degree pawl(used in the test setup of Peerdeman (Peerdeman, 2012)) is known that the real contact angle has to be 5.5 degree for the small one and 5.5 degree for the big one, besides the production flaws of the pawls.

Conclusion
The deviations, between de concept of a constant contact angle and the made pawls FA 4, 5 and 6, are too big. The 7 degree pawls deviates more the 1.5 degree. That comes from wrong assumptions and a wrong way of drawing the shape and dimensions of the contact area of pawl in SolidWorks.

4.3.5 Super pawl solution

A pawl or several pawls is needed to test the different friction coefficients of the used drum or coating to get the friction coefficient higher.

Set of requirements.
-The cost of the solution relative to the possibilities so low as possible
-Get the pawl measure so precise as possible.
-An calibration possibilities on the pawl

The different found concepts.

- Different pawls with different with different constant contact angles
- A pawl where the length is adjustable.
- A pawl with the formula $r=a* e$ ^($\tan(d*\theta)*\theta$). One pawl for different contact angles.
- A pawl with a linear change of contact angles.

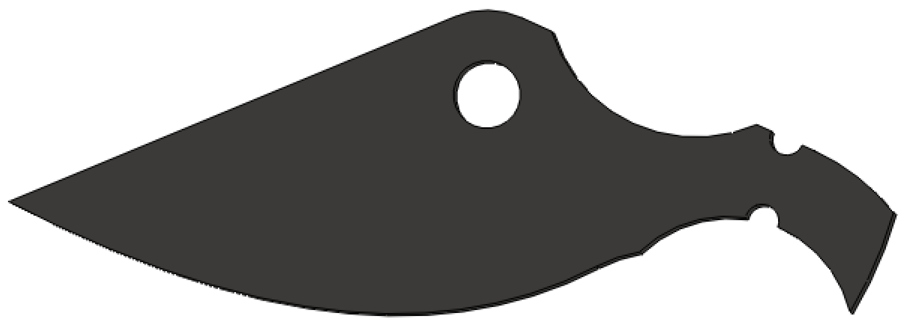


Figure 40 a pawl with linear contact angle change

The concept of a pawl with linear change of contact angle is chosen because the measurement of the contact angle(so the friction coefficient) can be taken very précises and it was cheapish solu-tion.

4.3.5.1 Explanation of the spiral of super pawl

For the pawl with a linear change of contact angle a formula is needed to describe the shape of the contact surface.
What is needed is θ*b= α(α is contact angle, and θ is the corner of the pawl). So a linear relation between θ and α. Linear relation is useful because then in the test the angle of the pawl can be measured therefor the contact corner(so the friction).

The formula:

Modify the now used formula ‘r=a* *e* ^(tan(α)*θ)’ to ‘r=a* *e* ^(tan(θ*b)*θ)’ isn’t possible because ‘r=a* *e* ^(tan(α)*θ)’ is an integral from formula 1 in Figure 41 (see: see chapter:**10.1** and **Figure 43)**

We replaced in formula 1 ‘Θ– α’ with ‘(1-d)*Θ’. We put it in Matlab and make an integral from it (Figure 44**).** The result of the integral is formula 2 in Figure 42. And that is the formula that is used for the contact surface shape.

$$\frac{1}{r}dr = -\frac{\left(\cos\theta + \sin\theta\tan\left(\frac{\pi}{2}+\theta-\alpha\right)\right)}{\sin\theta - \cos\theta\tan\left(\frac{\pi}{2}+\theta-\alpha\right)}d\theta$$

Figure 41 forumula 1

$$r = c_1\frac{\left(\tan\left(\frac{\pi}{2}-\theta(d-1)\right)^2+1\right)^{\frac{1}{2d}}}{\left(\sin(\theta)-\tan\left(\frac{\pi}{2}-\theta(d-1)\cos(\theta)\right)\right)^{\frac{1}{d}}}$$

Figure 42, formula 2, c1 is begin value r where Θ=0

II. CURVATURE CALCULATIONS FOR A CONSTANT ANGLE

In Fig. 2 a straight slot and a curved slot are depicted. For every angle θ the angle α must be constant. In Cartesian coordinates this results in the following relation:

$$\frac{dy}{dx} = \tan\left(\frac{\pi}{2}+\theta-\alpha\right)$$

A transformation to polar coordinates is achieved by

$$x = r(\theta)\cos\theta$$
$$y = r(\theta)\sin\theta$$

where $r(\theta)$ is a function of θ. Differentation of x and y with respect to θ and combining this with (1) results in the following differential equation:

$$\frac{r'\sin\theta + r\cos\theta}{r'\cos\theta - r\sin\theta} = \tan\left(\frac{\pi}{2}+\theta-\alpha\right)$$

where r' and r are functions of θ. Eq. (3) can be written as follows:

$$\frac{1}{r}dr = -\frac{\left(\cos\theta + \sin\theta\tan\left(\frac{\pi}{2}+\theta-\alpha\right)\right)}{\sin\theta - \cos\theta\tan\left(\frac{\pi}{2}+\theta-\alpha\right)}d\theta$$

After integrating both sides and some mathematical

Figure 43 see chapter: 10.1

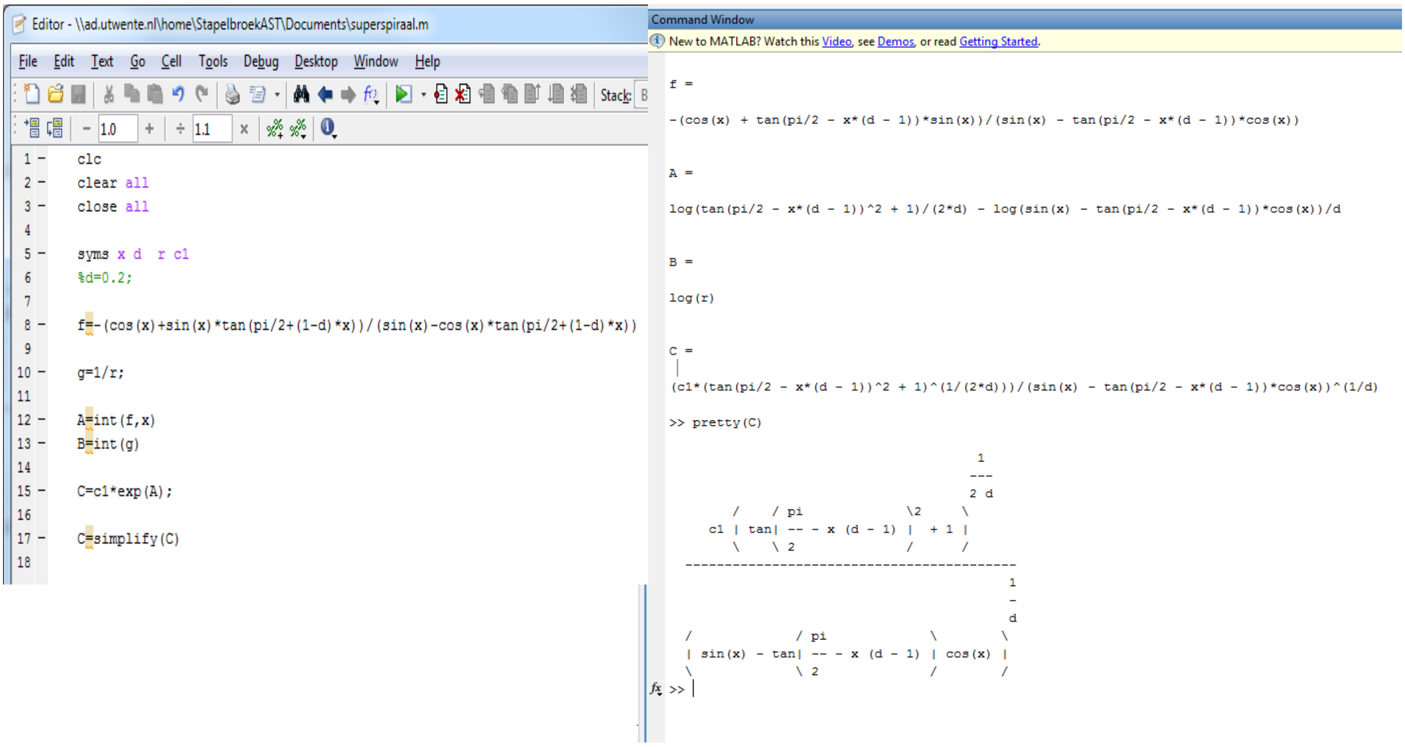


Figure 44 Integral in Matlab

4.3.5.2 Properties produced pawl

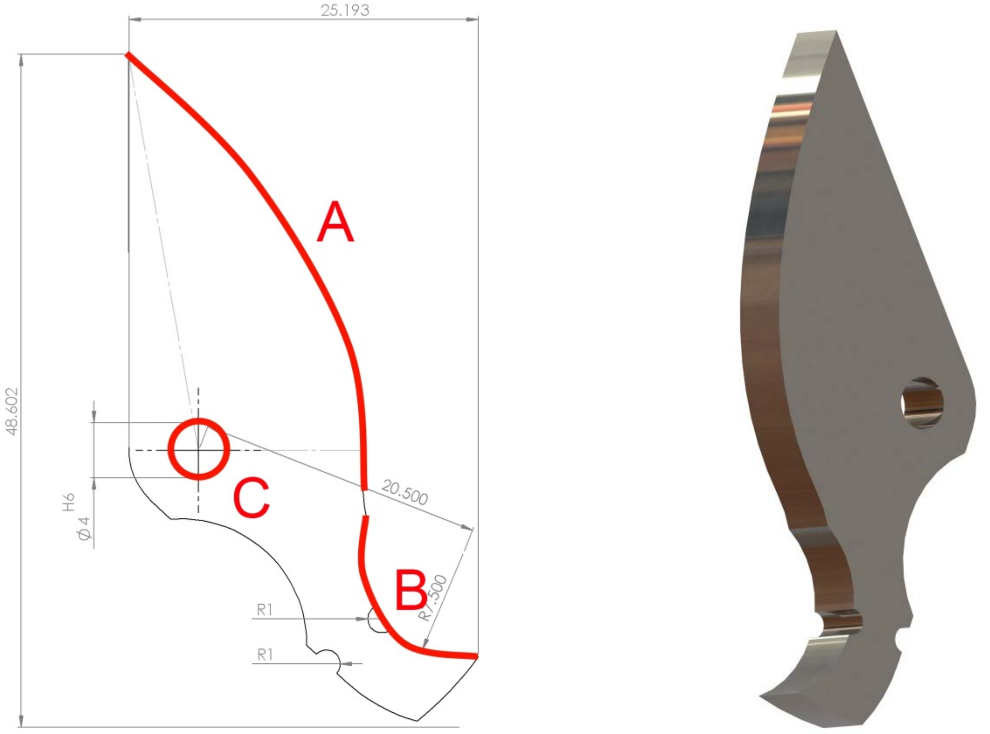


Figure 45 important surfaced from superpawl and rendered picture superpawl

The superpawl is made from hardened steel (1.2379) where the red surfaces (Figure 45) are wire sparked in one fixture.

The spiral is draw in Solidworks with equation driven curve in parametric mode. With the next input values.

x(t)=cos(t)*(12*(tan(pi/2 - t*(0.5- 1))^2 + 1)^(1/(2*0.5)))/(sin(t) - tan(pi/2 - t*(0.5 - 1))*cos(t))^(1/0.5)
y(t)= sin(t)*(12*(tan(pi/2 - t*(0.5- 1))^2 + 1)^(1/(2*0.5)))/(sin(t) - tan(pi/2 - t*(0.5 - 1))*cos(t))^(1/0.5)

t_1= -10/180*pi
t_2= 100/180*pi

There are deviations between the theoretical spiral and the reel one. There is a calculation flaw in solid words what has probably of deviation of 0.1 degree of the contact angle(Solidworks has an accuracy limit +/- 1e-6). Other flaw is the wire sparking flaw of ±0.003 mm.

De measurement of the contact angle can be measured thru making a picture of the pawl en the shafts to measure the angle in the picture, measure the angle between the pawl en the rig and measure the distance of the shafts .

4.4 Drum

A drum for this purpose has to be stiff and be able the transit the forces (up to 2.2 kN) on the 'arm' and the shaft. The full 2d drawing is in the attachment (chapter: **10.4**).

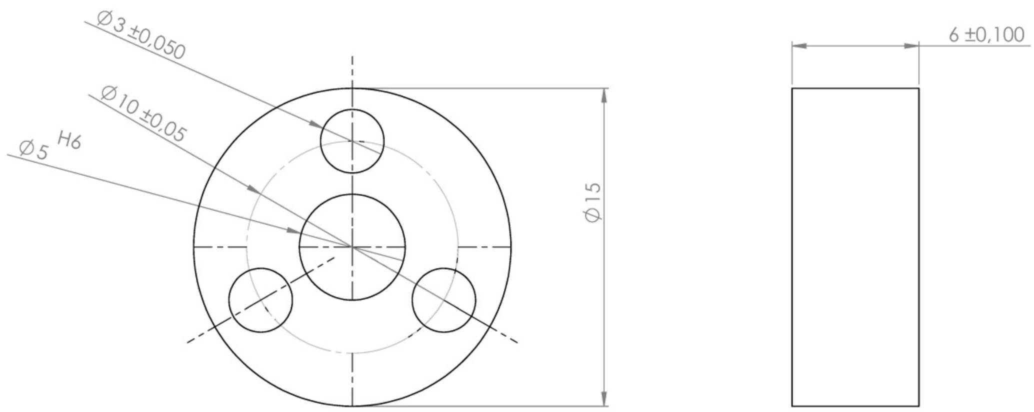


Figure 46 Drum

The versions of the drums. The first version is made of hardened steel and wire sparked. The second version is made for the fingers-test rig whit instead of three 3 mm holes, one 2mm. The drums used for the test are made of normal "automatenstaal (free machining steel)" with the dimensions of **Figure** 46, 30 are made.
Also is looked for higher friction to made the drum or the pawl from another material. Other metals have a higher friction that is also seen in the results seen in chapter 5.3. The differs between the friction coefficient are so low that we first look to coatings. Other materials like plastic could good give a higher coefficient but give also compliance . It is better to look for coating from plastics etc.

4.5 Bearings

The problem with the existing design is that they use ball bearings. The ball bearings are not able to withstand the forces from the locking joint. These fall a factor 10 short and are hereby unusable.

The only bearings who are able to withstand the force are needle rolling bearings, plastic slide bearings, composite slide bearing and metal slide bearing. The needle rolling bearings have less friction but have more tolerance and are bigger. The plastic slide bearings have more friction than a needle rolling bearing but less than metal bearings, less tolerance than needle bearings and spent under the force to must. Composite slide bearing are the same as plastic but are stiffer and more reliable. Metal slide bearings have the biggest tolerance, almost no compliance , the largest amount of friction and are the smallest in size.
For the test rig is chosen for needle bearing because it has no friction who can influence the measurements. But the rig is capable of having slide bearings. In the future the slide metal bear-

ings are most likely, due the size and high tolerances are these bearings probably the best option. This has to be tested in the future.

4.6 Coating

The goal is to get the friction coefficient higher to reduce forces and the wear in the system. The wishes is that the coating do not increase the compliance in the system. In general there are four kind of coatings available. For the pros and cons of the coatings see Figure 48.

Rubber type of coating
These increase friction, the contact surface(lowers the contact tension) and the compliance is higher.

Plastic type of coating
Types of plastic likes epoxy and polyester resin with particles (for example silicon carbide) who increase the friction. The coatings have a lot of compliance . Due to the lack of information and chance of success this type was not further investigated.

Metal coatings
These type of coatings use the metal as binder for particles like silicon carbide or diamonds which are friction increasing. This types of coating have law compliance.

Ceramic coatings
These types do not increase the friction coefficient enough and are very expensive. That are the reasons why the coatings are not chosen. If the chosen coatings do not work the ceramic coatings will be option. Coatings like the BALINIT coatings from Balzer or Ionbond. The ceramic TiN is maybe a solution. For these coatings the drum has to be made from hard material otherwise the coating will break. This types of coating have law compliance.

Friction exist from plowing and adhesion(see **Figure 47**). We want the adhesion component so high as possible and the plow component so low as possible. With adhesion there is deform elastic of the contact material an plow is from plastic of the contact material.

$\mu = \mu_a + \mu_p$
Total friction coefficient: μ
Adhesion coefficient of friction: μ_a
Plowing coefficient of friction: μ_p

Figure 47 Friction coefficient

	Rubber	Plastic	Plastic with particles	Metal	Metal with particles	Ceramic
Friction	+++	++	+++	+	+++	+
μ_a	+++	++	++	+	+	+

μ_p	-	-	++	+	+++	+
compliances	+++	++	++	-	-	-
sustainability	-	+	+	+	+	+++

Figure 48 table of pros and cons coatings.

4.6.1 Keratherm Gluey Soft Coating

A rubber type of coating.
Keratherm gluey soft coating is normally used for putting a phone on the dashboard of a car. Expect is that the forces are too high for the coating. The film is placed like sticky tape.

4.6.2 Plastic Dip Coating

A rubber type of coating
Plastic dip is normally used for increasing grip on handles of tools. And placed on the drum by dipping the drum in the can of coating

Manufactures number: 61001023

4.6.3 3m film

Plastic/rubber type of coating

The film didn't had on the carrier(the white stuff where the film sticks on) any 3M logo. Almost every 3M product has a 3M logo on it, even on the carrier. It is not known if it is the real stuff.
The film normal used for covering paint of cars. The film is placed like sticky tape.
The film is ordered at: carbodydesign_eu in Grażyna Kempa Poland. The official name: 3M SCOTCHGARD Paint Protection Film PU 8592 E Clear Bra 200 MICRONS (110849968757)

4.6.4 Nickel silicon carbide Coating

Metal coatings
"SIC-9-DURNI-DISP" AHC 9-12 µm size silicon carbide particles
ca. 7 µm thick Nickel layer as carrier of the particles. The decision is made for silicon carbide instead of diamonds because the silicon carbide particles are more of a sphere shape. It is expected that this coating increase the friction. Considering the alternatives this coating will be probably the best solution.

http://www.ahcbenelux.nl/nieuws/wrijvingverhogende-oppervlaktebehandeling-sic-9

5. Research

5.1 Introduction

This chapter is about a number of tests conducted on the new test rig. To answer our questions.

The test who are planned:

- Trial Tests Friction (superpawl) (done)
- Test Friction (superpawl) (done)
- Test old pawls (not done)
- Test vertical contact angle (not done)
- Test friction of the bearings (not done)
- Test influence of the bearings on the angle change of the 'arm' (not done)
- Test new pawls (not done)

5.1.1 Measurement errors

Test done with the super pawl with measurements taken on the shafts has a flaw between -0.08 and -0.025 mm.
The measurement of the flaw is taken with micrometer, from Steinmeyer 0-25 mm 0.01 DIN83./I, on the two shafts with the calibration circle (line B in: **Figure 49)** from the super pawl against the drum. The outer (shafts pushed out horizontal) measurement was 24.975 the inner (shafts pushed in horizontal) measurement was 24.92. Distance has to be 25.00 mm(20.5+5/2+4/2).

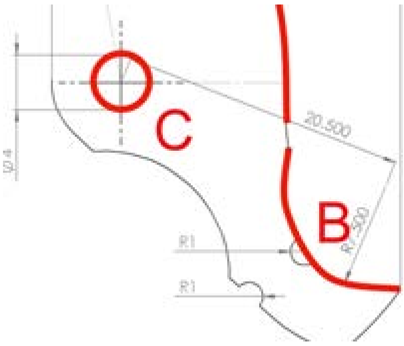


Figure 49 oak circle B

5.2 Trial Tests Friction (superpawl)

Some trial test are done to get an idea of how the several coating behave.

First with testing with the superpawl on the highest contact angle when the pawl gest grip then the next procedure is used:

1. Assembly drum and pawl combination.
2. Check for contamination in the test rig.
3. Goal is to test the frictions on 3 different point on the drum
4. Test steps
    a. Tighten light the horizontal m6(number 21 in the assembly drawing[1]) nut and the two vertical m6 nuts(number 9 in the assembly drawing[1]). So light that the 'houd-er'[1] still moves.
    b. Put the 'houder'[1] at the right distance (use micrometer to measure distance). Use if needed nut number 24 in the assembly drawing[1] to change the distance.
    c. Tighten hard nut number 21 in the assembly drawing[1]
    d. Tighten hard nuts number 9 in the assembly drawing[1]
    e. Un tighten nut number 24 in the assembly drawing[1]
    f. Start the measurement.
        i. distance between the shafts (4 and 8 in the assembly drawing[1])before the tests
        ii. picture directly when the pawl get grip on the drum
        iii. picture when the "arm" doesn't move any more. If the "arm" keeps moving make the picture when the unster is 8 newton If 8 newton's is not possible note the force maximum used.
        iv. Measure the angle change between picture 1 and 2.
        v. Write down if the noticeable things.
        vi. distance between the shafts (4 and 8 in the assembly drawing[1])after the tests
        vii. Measure in the picture the contact angle
        viii. Measure in the picture the angle of pawl
        ix. Measure the angle of the pawl
        x. Write down the forces when the
5. Results
    g. The reel contact angle of the in the different test, also an average and a remarkable points.
    h. The force in the system when the measurement are taken.
    i. Conclusion
        i. Difference between practice and theory.
        ii. Analysing where angle change from the 'arm' comes from.

5.2.1 3M film 8592

Measurement:
The 3M film is 0.20 mm thick
Distance between the shafts (measured from the outer side) 24.03 that is equal to a contact angle of 1.844 degrees (corrected whit the thickness of the film and corrected with the measurement flaw of 0.08 (see chapter: 5.1.1) measured in Solidworks model of superpawl(see: Figure 50)).

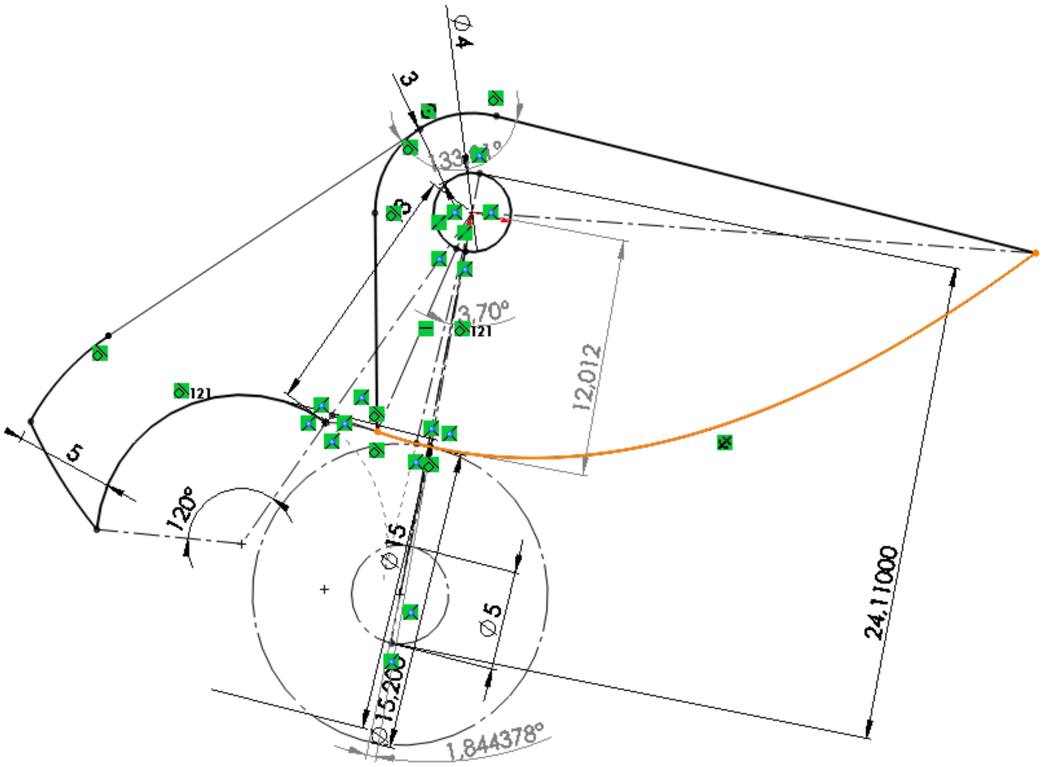


Figure 50 measurement contact angle superpawl

Analysing the movie of the test. In the movie the film stays in place and the pawl slips but stays on the film.
Result:
there was negative result. The pawl still slipped at a contact comer of 1.8 with a maximum torque of around 0.4 n*m.
Recommendation/conclusion:
This film is not an option for extra friction.

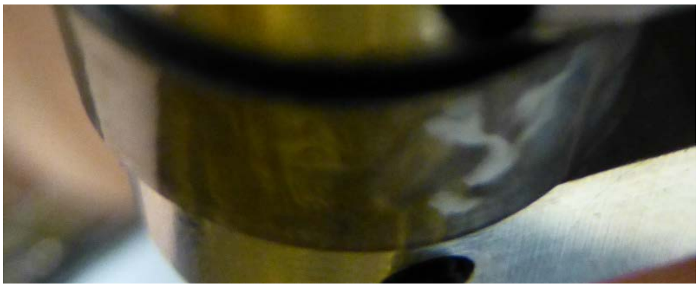


Figure 51 tested 3M film 8592

5.2.2 Keratherm Gluey Soft Coating

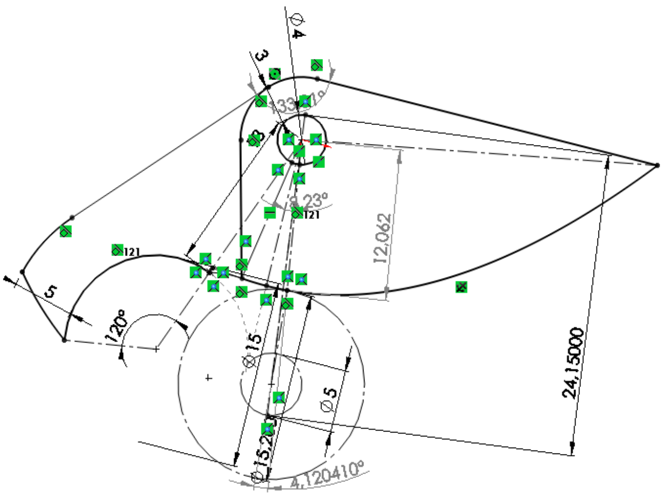
Measurement:
Contact angle of around 10 degrees. Distance of the shafts is 24.3 mm.
Result:
It is destroyed with a maximum torque of around 0.35 n*m

Figure 52 destroyed Keratherm Gluey Soft Coating

Analysing the movie of the test. The pawl gets grip and destroyed the coating en slips through.

Recommendation/conclusion:
This coating is not suitable.

5.2.3 Plastic Dip Coating



In different test are different thicknesses tested.

First test

Measurement:

The plastic dip is around 0.2mm thick and the surface is bumpy.(dipped 1 time)
Distance between the shafts (measured from the outer side) 24.07 that is equal to a contact angle of 4.12 degrees(corrected whit the thickness of the coating and corrected with the measurement flaw of 0.08 (see chapter: 5.1.1) measured in Solidworks model of superpawl).
a maximum torque of around 0.4 n*m.

Result:

Analysing the movie of the test. The pawl gets grip and then destroyed the coating.

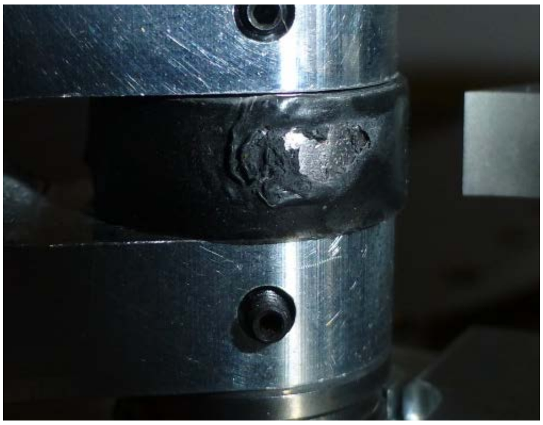


Figure 53 Destroyed plastic dip coating

Second test

Measurement:

The thickness of the coating is 0.05 mm(dipped 1 time), for a more uniform thickness to the coating was 30% thinner added. This resulted in a more uniform thickness(the reason of adding the thinner) but after drying the surface has small bobbles in it because of the evaporation of the thinner. (thinner is used in this test)
The drum with the coating is tested twice on each side of the drum.

Result:

First at a distance of 24.19 (with measuring flaw correction 0.08 mm) with a max torque of 0.18 nm with a contact angle of 6.86 degree
Second at a distance of 24.17 (with measuring flaw correction 0.08 mm) with a max torque of 0.07 nm with a contact angle of 6.35 degree
Both slipped first and when the pawl get grip the coating breaks

The third test

Measurement:

The thickness of the coating is 0.63 mm is dried for 3 weeks. The super pawl is defatted with acetone. Because of the compliance the rubber coating has a min and a max contact angle. The correction of the distance of the shafts is 0.125 mm.

Result:

Drum	measurement	distance	corrected distance	contact angle min	contact angle low max	Friction coefficient min	Friction coefficient max	lock
rubber	10	25,010	25,135	12,970	18,810	0,230	0,341	yes
	11	25,840	25,965	20,400	24,089	0,372	0,447	yes
	12	26,460	26,585	24,189	27,090	0,449	0,512	no
	13	24,730	24,855	8,800	16,490	0,155	0,296	yes

Figure 54 results rubber coating 0.63 mm

In this test a small compliance is seen, but an acceptable compliance . The rubber locked very well with contact angle till 20-24 (min-max) degrees. That is promising.

Recommendation/conclusion:
The problems of the coating is the bonding on the drum and tensile strength of the coating. The coating has an acceptable compliance and locks very well at a thinness of 0.63 mm.
For this coating is probably better for more grip that the contact surface of the pawl is defatted, that the coating is dried for more than 3 weeks and the coating is relative thick(0.6 mm).
Recommendation is to search for a type of rubber who is stronger, same elasticity and stick better on the drum.

5.2.4 Nickel silicium carbide Coating

The first test a distance with correction between the shafts is measured of 26.13 mm. That means a contact angle of 24.61 degrees and a friction coefficient of 0.45. A noteworthy observation is that the 'arm' doesn't change angle when locking occurs(very low compliance). The forces will be lowered by 3.7. this means a positive result. This coating will be further researched especially on sustainability and the exact friction coefficient.

5.3 Test Friction (superpawl)

Whit this test the friction between the superpawl and contact surface is tested. The friction of the old drums and the silicon carbide drums are tested.

The questions:

1. What is the friction in the existing system?
2. What is the friction from the new coatings or material combinations?

3. What are the draw backs the good things of the new coatings or material combinations? And what is the influence? (Such as compliance in the system.)

5.3.1 Test procedure

1. Assembly drum and pawl combination.
2. Check for contamination in the test rig.
3. Goal is to test the coefficient of friction 5 different point on the drum.
4. Number of test
    a. The contact angle between the 45 and the 0 degree will be tested with steps of 5 degree.
5. Test steps
    a. Tighten light the horizontal m6(number 21 in the assembly drawing[1]) nut and the two vertical m6 nuts(number 9 in the assembly drawing[1]). So light that the 'houder'[1] still moves.
    b. Put the 'houder'[1] at the right distance (use micrometer to measure distance). Use if needed nut number 24 in the assembly drawing[1] to change the distance.
    c. Tighten hard nut number 21 in the assembly drawing[1]
    d. Tighten hard nuts number 9 in the assembly drawing[1]
    e. Un tighten nut number 24 in the assembly drawing[1]
    f. Start the measurement.
        i. distance between the shafts (4 and 8 in the assembly drawing[1])before the tests
        ii. picture directly when the pawl get grip on the drum
        iii. picture when the "arm" doesn't move any more. If the "arm" keep moving make the picture when the unster is 8 newton
        iv. Measure the angle change between picture 1 and 2.
        v. Write down if the noticeable things.
        vi. distance between the shafts (4 and 8 in the assembly drawing[1])after the tests (check if distances are the same if not stop measurement)
        vii. Measure in the picture the contact angle
        viii. Measure in the picture the angle of pawl
        ix. Measure the angle of the pawl
        x. Write down the forces when the
6. Results
    a. The reel contact angle of the in the different test, also an average and a different and the calculate coefficient of friction.
    b. The force in the system when the measurement are taken.
    c. Conclusion

[1] drawing in attachment: **10.2 2d drawings test rig**

5.3.2 Friction test old drums

To test exactly the friction of the old drums this test is done. The was a drum who is wired sparkled and used in the old test rig and a drum made for the silicon carbide coating from automatenstaal. Before the test the drums and the super pawl are defatted by acetone. The test is done by procedure **5.3.1**. The correction on the distance is 0.125 mm.

5.3.2.1 Results

Drum	measurement	distance	corrected distance	contact angle	Needed friction coefficient	lock
Wire sparked	1	24,039	24,164	7,400	0,130	no
	2	23,890	24,015	2,300	0,040	yes
	3	22,970	23,095	5,600	0,098	half
	4	22,900	23,025	2,880	0,050	yes
automatenstaal	5	24,030	24,155	7,200	0,126	yes
	6	24,360	24,485	12,580	0,223	no
	7	24,150	24,275	9,550	0,168	no
	8	24,060	24,185	7,860	0,138	half
	9	24,025	24,150	7,090	0,124	yes

Figure 55 results old drums

In the result the wire sparked drum the friction coefficient is between 0.05 and 0.098. The Theoretical friction coefficient is 0.12 (see page 212 (Bruijn, 1997)).
The friction coefficient of the 'automatenstaal' drum is between the 0.126 and 0.138. That is more like the theoretical friction.
That the friction of the coefficient of the wire sparked drum is lower probably comes from high contact forces.

5.3.2.2 Remarkable points

From the test results of Pieterse (pieterse, 2012) showed that his seven degrees pawls (small and big) work properly. In field tests of Peerdeman it shows that these don't function all the time. In reality the 7 degrees pawl have a contact angle of 5.5 degrees. A contact angle of 5.5 needs a minimal friction coefficient of about 0.096. This is back up by friction test of the old drums.

5.3.2.3 Conclusion

The results of a friction coefficient is between 0.05 and 0.098 for the wire sparked drums and a friction of the coefficient of 0.126 and 0.138 of the 'automatenstaal' drum have logical explanations.

5.3.3 Friction test silicon carbide coating

The test is done on 2 drums, drum 1 and drum 2. The test is done by the test procedure **5.3.1.** Each measurement is on 5 points on the drum and in the same 80 degree area of the drum. Drum 1 use also used for the trial test of the coating. Each drum has put to the test more than 50 times in the 80 degrees area. The torque what is used to test if the system lock is 0,7 N*m.

5.3.3.1 Results

Drum	picture	measurement	distance	corrected distance	contact angle	Friction coefficient	lock?	further information
1	no	1	25,952	26,032	24,160	0,449	no	
1	no	2	25,415	25,495	20,990	0,384	half	
1	no	3	24,955	25,035	17,589	0,317	half	
1	no	4	24,535	24,615	13,495	0,240	yes	
1	no	5	25,030	25,110	18,202	0,329	half	
1	1	6	25,755	25,835	23,070	0,426	half	sticks only on 'new' part of the drum
1	2 and 3	7	25,845	25,925	23,582	0,437	half	sticks only on 'new' part of the drum
1	4	8	25,915	25,995	23,966	0,445	no	
2	no	9	24,910	24,990	17,210	0,310	yes	
2	no	10	25,460	25,540	21,280	0,389	yes	
2	7	11	25,970	26,050	24,261	0,451	half	
2	no	12	25,505	25,585	21,570	0,395	half	first grip forces become higher the system slips
2	no	13	25,455	25,535	21,250	0,389	half	first grip forces become higher the system slips
2	no	1	25,165	25,245	19,250	0,349	no	

		4						
2	8	15	24,930	25,010	17,380	0,313	half	lock occasionally, if it locks the lock is capable of high forces of 10 newton
2	9	16	24,360	24,440	11,262	0,199	yes	
2	no	17	24,650	24,730	14,755	0,263	no	
2	10	18	24,500	24,580	13,083	0,232	half	lock occasionally, if it locks the lock is capable of high forces of 10 newton
2	11	19	24,215	24,295	8,930	0,157	half	
2	no	20	23,825	23,825		0,000	no	no solution possible

Figure 56 Results test silicon carbide coating

See for the not corrected pictures of the test chapter: 10.7. These pictures are not corrected on the lens of the camera (Panasonic DMC-TZ30) with Adobe Photoshop.

5.3.3.2 Remarkable points

Remarkable is that the friction becomes lower when more the drum is tested in the same area. During measurement 6 and 7 the new not tested surface was also tested. On the new not tested surface the lock locks every time. After the test the pawl and drum showed wear, see **Figure** 57, **Figure** 58 and **Figure** 59. Also remarkable is that the wear is not on the full contact length of the pawl and drum that indicates maybe for a vertical misalignment of the drum and the pawl.

Figure 57 Drum 1 after testing

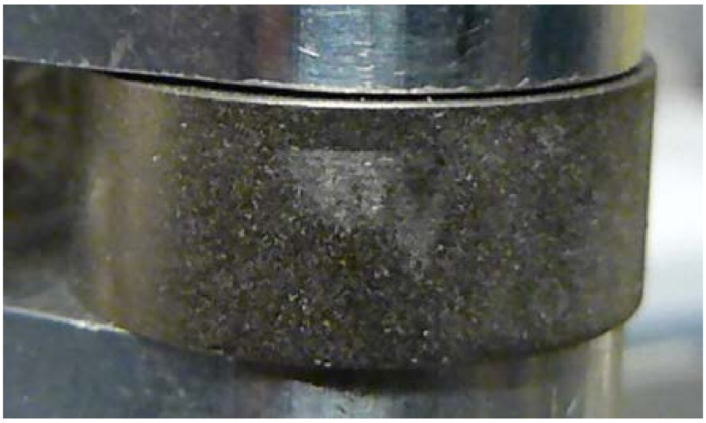


Figure 58 drum 2 after testing

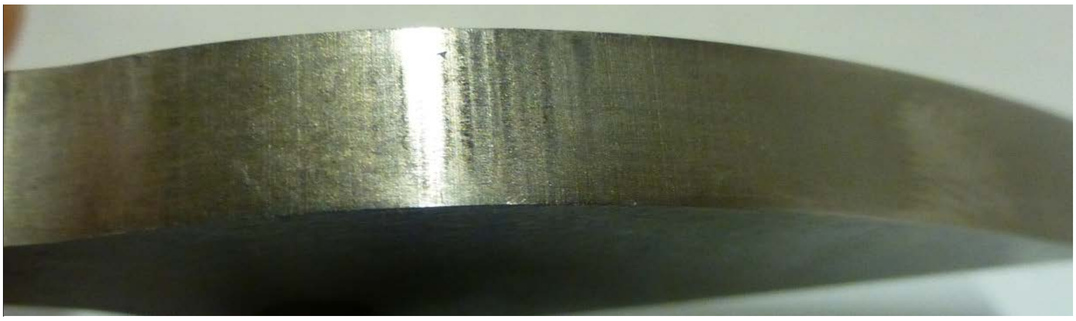


Figure 59 super pawl after testing drum 1 and 2

There is also a test done with the pawl of Pieterse with a contact angle of 13 degrees(FA6). The drum and pawl is 25 times test on two point of the pawl and drum(total 50 times). Al the times the joint locks but also a misalignment and wear is seen on the contact surfaces.

Figure 60 Wear on pawl (contact angle 13 degree) and drum

5.3.3.3 Analysing results

The largest problem is that the level of friction drops during use. This can be caused by any of following reasons.

Damage super pawl,

- the nickel transfers to the pawl.
    - Solution: use a anti stick coating on the pawl like BLC
- Normal wear on the pawl caused by the forces.
    - Solution used a hard coating on pawl like TiN from Balzers or Ionbond. These coating can also increase the friction and also lower the wear of silicon carbide coating.
    - Solution Do not used silicon carbide particles.
- The pawl pushed the particle out of the nickel.
    - Solution use a thin rubber layer on the pawl

Damage drum

- the nickel transfers to the pawl.
    - Solution: use a anti stick coating on the pawl like BLC
- Silicon carbide particles are gone
    - Solution: Get other carrier material than nickel.
    - Solution: Use a kid of rubber coating on the pawl.
- Silicon carbide particles are pushed in the drum
    - Solution use tool steel ore other harder materials.
    - Solution: coat before the silicon carbide coating the drum with a hard coating.
- Maybe the coating wasn't done properly. Because of production problem or the surface of the drum to rough for the coating despite the surface met the demands of the supplier.
- The lager holes in the contact surface of the drum are most likely cause by a cleaning agent used by the supplier of the coating. The surface is etched away although the material was up to the specification of the supplier.
    - Solution: Choose a drum material who can withstand the cleaning agent like tool-steel.

5.3.3.4 Analysing microscope results.

The next step is to take a closer look at the material using a optic microscope and an electron microscope. First is an optic microscope to measure the surface structure in microns. That is shown in **Figure** 61 and **Figure** 62. These picture shows that the silicon carbide particles have disappeared.

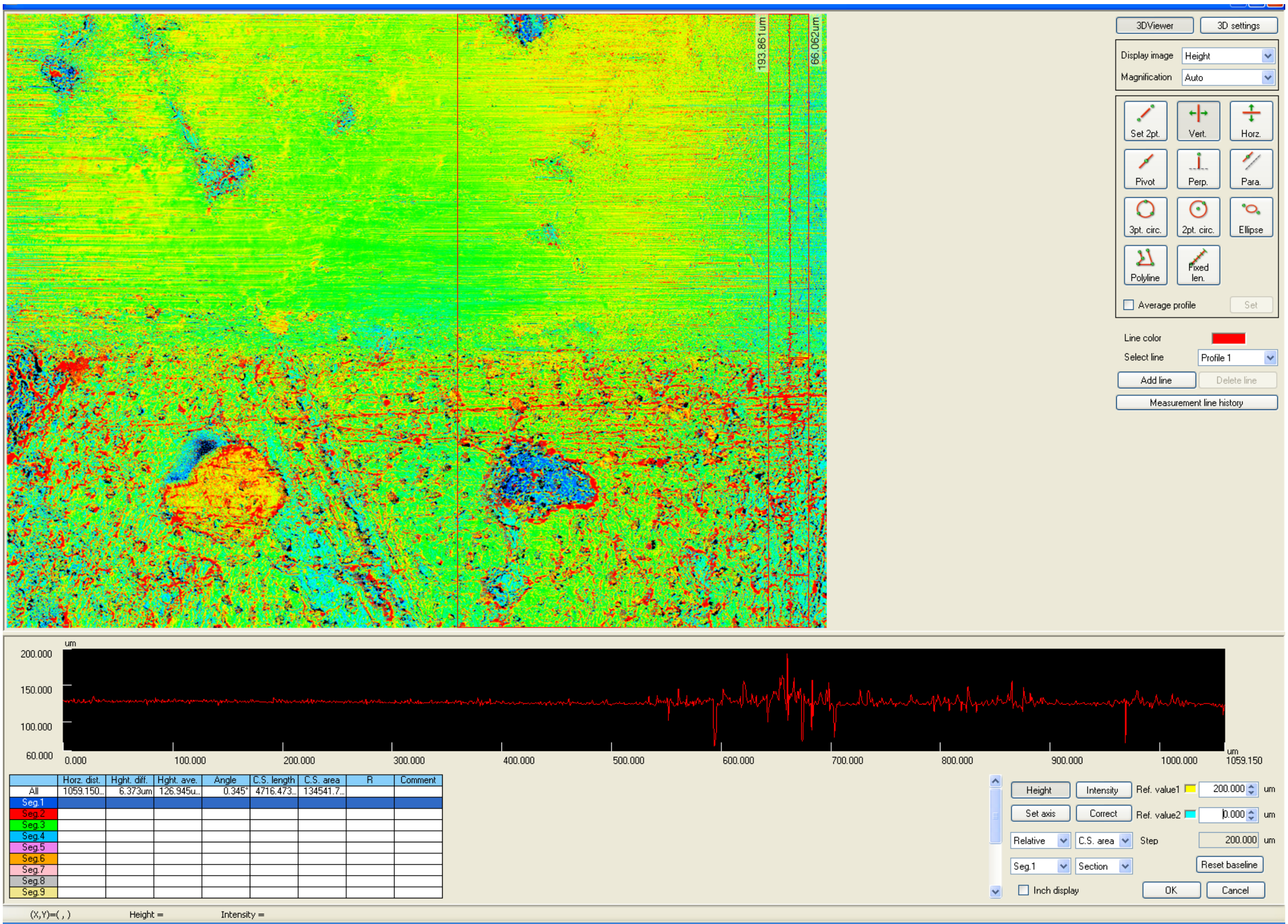


Figure 61 profile picture of contact surface of the drum with damaged silicon carbide coating

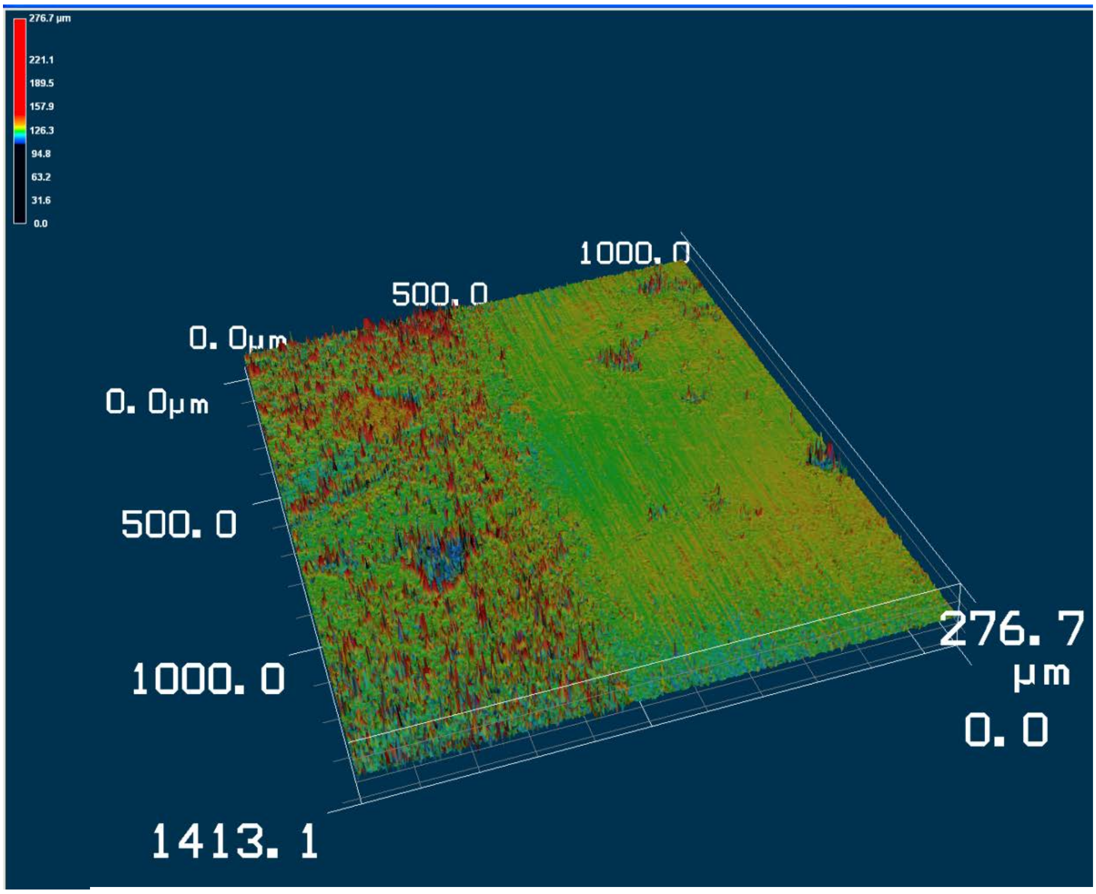


Figure 62 3d picture of contact surface of the drum with damaged silicon carbide coating

The optic microscope also took pictures of the wear inflicted on the pawl. This to get more information if the nickel is transferred from the drum to the pawl or some material from the pawl is removed. In **Figure 64** is clearly shows some wear and in Error! Reference source not found. there is a profile made of the surface. In the profile shows only static is showed on the wear points. It cannot be concluded that material was transferred nor can it be denied. Images who are more zoomed in are needed. Do to the pictures the electron microscope made of the drum, the nickel part of the coating was discarded. Further testing on this pawl will not be productive. Tests with an electron microscope to prove the presence of nickel would not help due to the fact that the base material contains nickel.

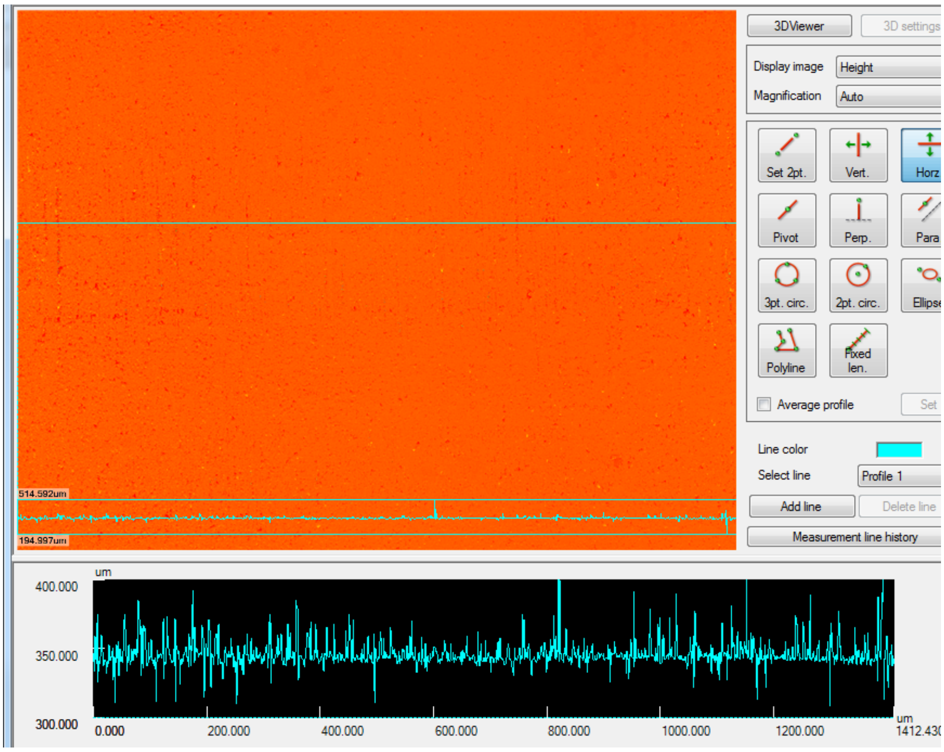


Figure 63 Profile of wear on the pawl

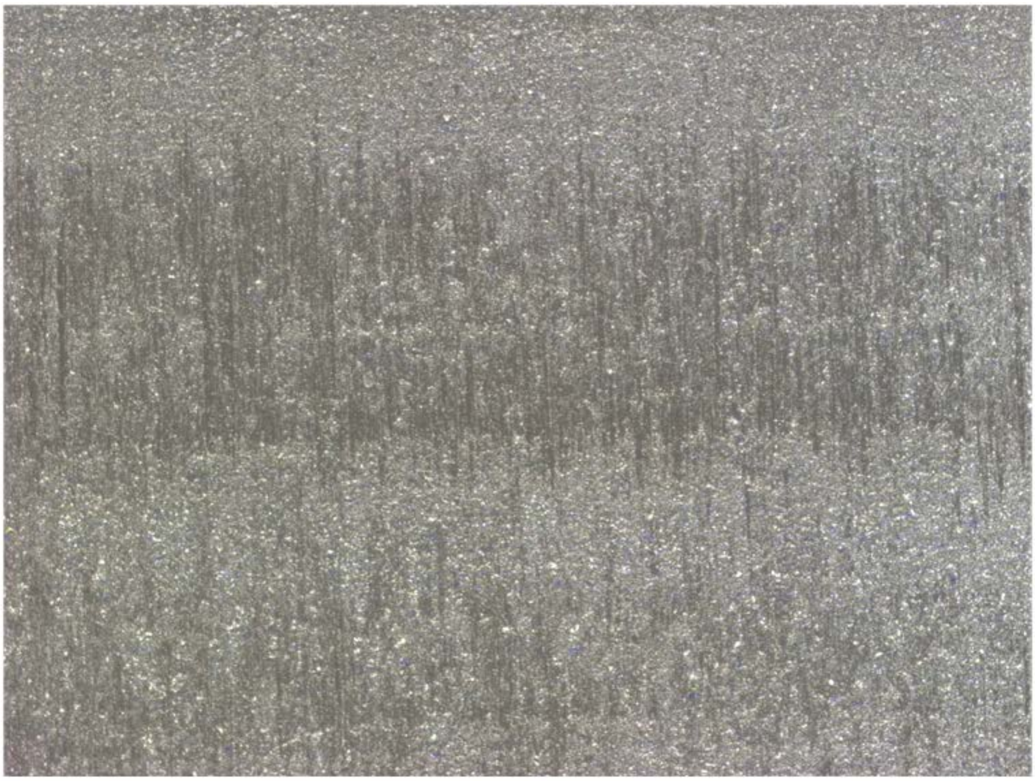


Figure 64 picture of the wear of the pawl

These pictures are made with an election microscope and show the drum surface with coating and the wear of the coating. In the top picture (**Figure** 65) the wear is clearly shown on the contact surface. In some places the nickel and the silicon carbide particles are both in place, however in others the particles are no longer in the nickel. And in some places the nickel and the particles are gone. The nickel structure and its reflexion is clearly shown in **Figure** 66, in **Figure** 65 it has been removed in some places and left the steel bare. The particles that have been removed cannot be destroyed and must either are pressed into, or are pushed off the drum. Since there are no holes visible where the particles are pushed in. A second option is that they were pushed in the material of the drum and covered with nickel or the material of the drum itself, and are no longer visible. There are no traces left that indicate that something has been smeared over. There are imprints of the particles in the drum material visible that were made before the failure of the coating. It can be concluded that they have been pushed off. Unfortunately we cannot investigate this more closely with an electron microscope used to identify the type of atoms in a material. This due to the fact that the base material also includes carbon and silicon. Based on the information at hand it is very unlikely that the particles are pressed in rather then pushed off.

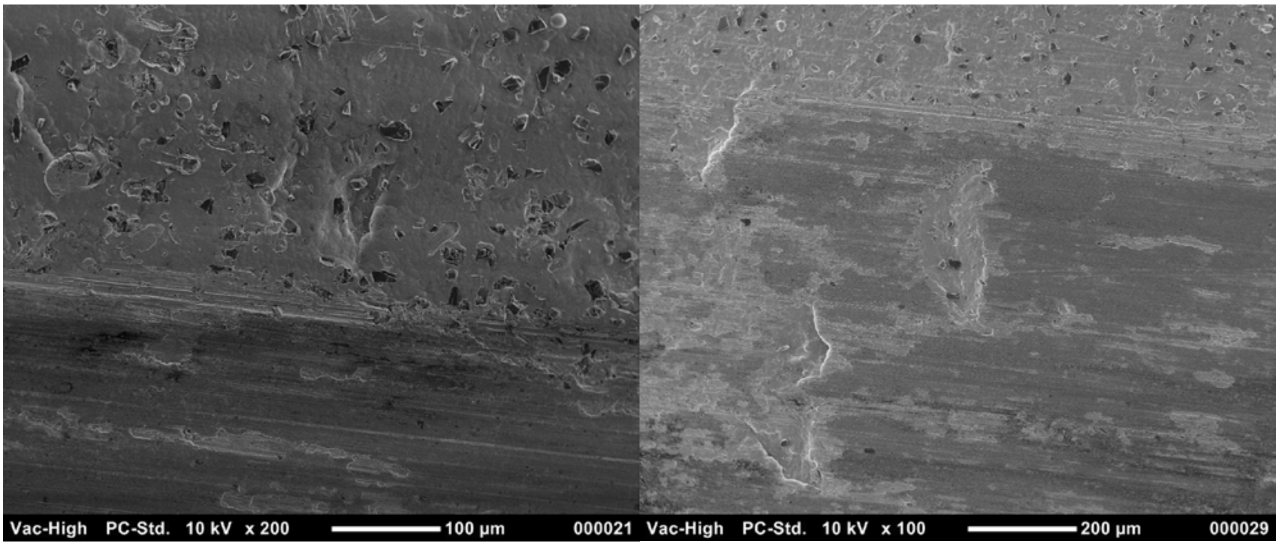


Figure 65 Pictures of the wear and the silicon carbide coating

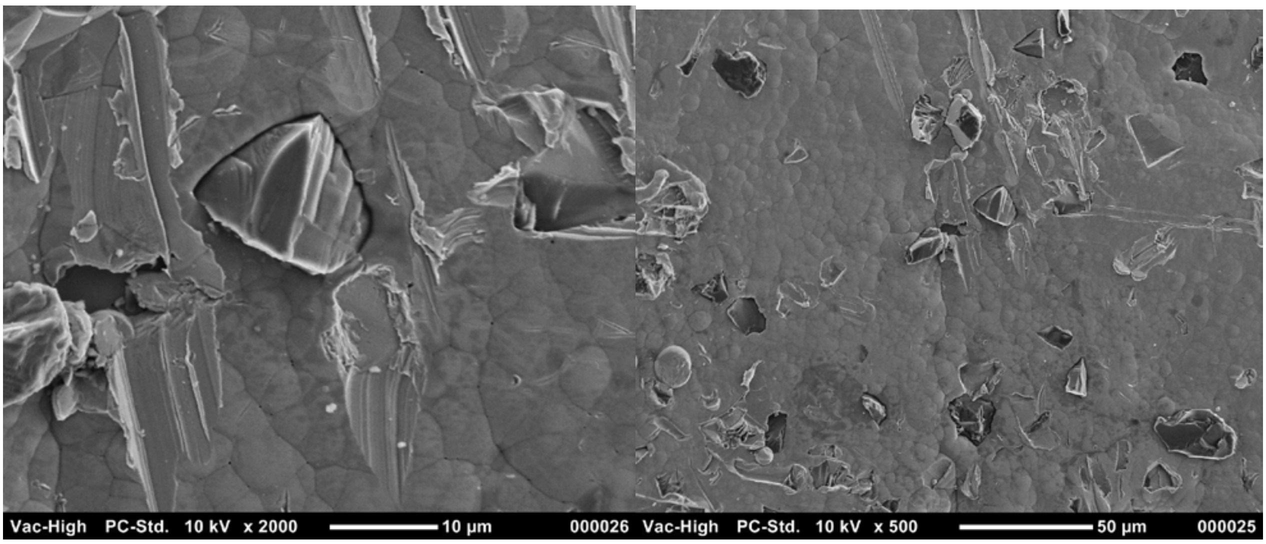


Figure 66 pictures of the silicon carbide coating

5.3.3.5 Conclusion and recommendation silicon carbide coating

The test results show that 10-11 micron silicon carbide particles increase the friction to a friction of 0.40. With nickel as carrier of the coating we see that the friction drops due to the fact that the particles are pushed away during use.
Here 2 things need to be closer examined. The first is the effects of the misalignment of the pawl and drum. Second is to find ways to put enough selenium carbide between the contact line of the pawl and the drum.(Contact AHC Benelux) Options are to find a better carrier compared to nickel, this is unlikely due to the fact that it has te be readily available and cost effective. Options to be explored are polymers (epoxy's etc.) and other metals. Another approach is adding the silicon carbide in use as an active anti-lubricant. For example by adding a silicon carbide gel with 10-11 micron silicon carbide particles.

5.4 The test for future work

Because of the lack of time some test are some test omitted. These test are in this chapter.

5.4.1 Test old pawls

This test is not done because the camera has to be calibrated to get the right contact angle and dimensions. The time (2 weeks) is needed was not available.

In this test the old pawls will be tested.

Questions:

1. What are the real positions and forces of the pawls?
    a. Contact forces?
    b. Positions of the shafts?
    c. Angle changing of the 'arm' when the arm locks?
        i. Where does it comes from?
    d. The torque of the arm?
    e. How does the spiral works in reality?

The test procedure needs more development.

1. Check for contamination in the test rig.
2. Goal is to test the friction on 3 different point on the drum and 3 different points on the pawl
3. Test steps
    a. Tighten light the horizontal m6(number 21 in the assembly drawing[1]) nut and the two vertical m6 nuts(number 9 in the assembly drawing[1]). So light that the 'houder'[1] still moves.

b. Put the 'houder'[1] at the right distance (use micrometer to measure distance). Use if needed nut number 24 in the assembly drawing[1] to change the distance.
c. Tighten hard nut number 21 in the assembly drawing[1]
d. Tighten hard nuts number 9 in the assembly drawing[1]
e. Un tighten nut number 24 in the assembly drawing[1]
f. Start the measurement.
    i. distance between the shafts (4 and 8 in the assembly drawing[1])before the tests
    ii. picture directly when the pawl get grip on the drum
    iii. picture when the "arm" doesn't move any more. If the "arm" keep moving make the picture when the unster is 8 newton If 8 newtons is not possible note the force maximum used.
    iv. Measure the angle change between picture 1 and 2.
    v. Write down if the noticeable things.
    vi. distance between the shafts (4 and 8 in the assembly drawing[1])after the tests
    vii. Measure in the picture the contact angle
    viii. Measure in the picture the angle of pawl
    ix. Measure the angle of the pawl
    x. Write down the forces when the

4. Results
    a. The reel contact angle of the in the different test, also an average and a remarkable points.
    b. The force in the system when the measurement are taken.
    c. Conclusion
        i. Difference between practice and theory.
        ii. Analysing where angle change from the 'arm' comes from.

[1] drawing in attachment: **10.2 2d drawings test rig**

5.4.2 Test vertical contact angle

In this test the contact angle vertical between the drum and pawl is tested.

The question: What is the influence of the angle vertical contact angle between the drum and the pawl?

a. What is the tolerance of it?
b. Is the tolerance acceptable for example milling or wire sparing.
c. What can be the solution?

Test procedure has to be further develop. The basic idea of the procedure is that the distance between the two shafts is set such that the needed friction coefficient for this distance and friction coefficient of the used drum and pawl(probably the superpawl) are the same. Then is small steps the vertical angle will be increased (with bold 10) and every step the locking will be tested. I the lock doesn't lock any more than the set vertical angle say something about the tolerances of the vertical contact angle.

5.4.3 Test friction of the bearings

In this test the influence friction of is tested.

Question:
What is the influence of the friction of the bearing on how the lock system works?

Test procedure has to be conceived.

5.4.4 Test influence of the bearings on the angle change of the 'arm'

In this test the angle change will be tested when the system lock is applied.

The question: What is the influence of the bearings on the angle change of the 'arm'?

Test procedure has to be further develop. In the test the angle change will be measured between to the same configurations (same drum and pawl), only difference is the bearings. The angle change will be measured at 4 different contact angles, at every contact angle the angle is tested on 4 different places on the drum. Also the difference in the play of the bearings are measured.

5.4.5 Test new pawls

There was no time to produce and design new pawls. The procedure is the same as the in the test of the **Test old pawls.**

Question: What is the performance of the new pawls?

6. Options for further design

In the process of the research several ideas are development for improving the joint lock design. To help the further improvement of the lock.

6.1 Bigger drum

When the drum becomes smaller the contact forces lower also. The problem with a bigger drum is space. With cutting the drum in half and repositioning the pawl and shaft, a bigger drum is possible with relative less space (**Figure** 67). Replacing the pawl is needed otherwise the pawl can't lock at the right angles.

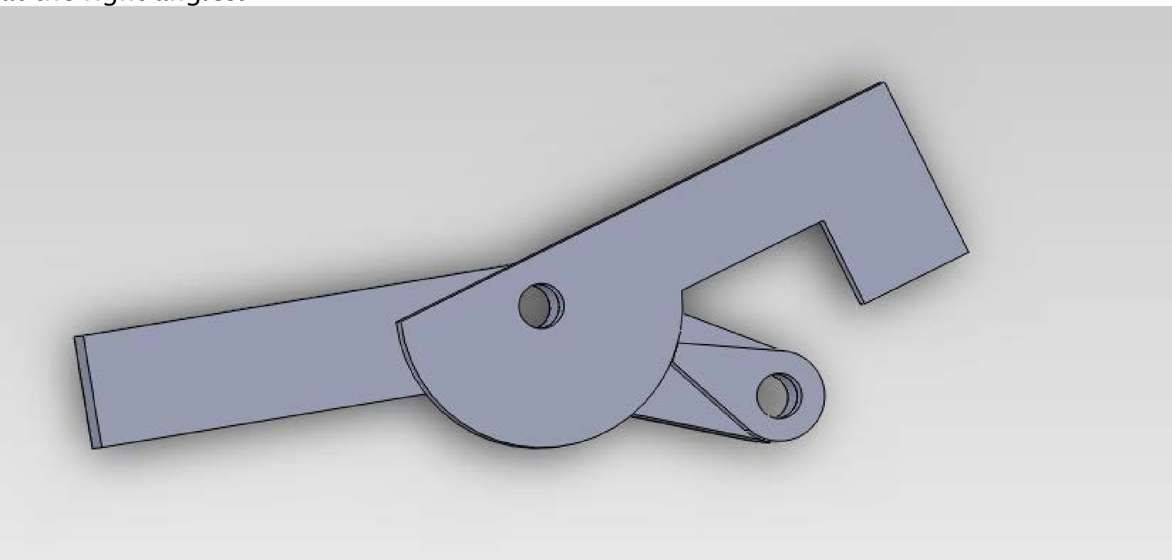


Figure 67 lock with bigger drum and another position of the pawl

Making drum and the pawl wider so the contact length becomes wider and the contact stress gets lower.

6.2 Saving space

Saving space in the design means more space for example a bigger drum or a wider line contact.
Integrate the drum with the phalange (like **Figure** 67) so there is no space needed for montage parts.
Make the bearing in the drum and pawl instead in the phalange. So the phalange can become smaller and for example the drum can become wider(to lower contact forces).
Also space is saved when parts are the integrated in each other like integrating spring in the pulley or integrate the drum with the pawl as one part.

6.3 Spiral scape

6.4 Overload protection

The problem with overload protection is that the most types ad more compliance and/or play to the system.

6.5 Bearings

By replacing the bearings by plain bearings from brass (messing, brons) or copper. The compliance problem so the angle change problem of the lock can be solved.
And when the bearings are made With low play and high accuracy the angle change and the compliance will be less.

6.6 Alignment

For better alignment from the pawl (better alignment means better force transition and less wear) on the drum are several options. A notch hinge in the pawl like **Figure** 68 is an expensive solution. Other solutions are the use of self-aligning bearings, use of elastic coatings or a aligning shafts.

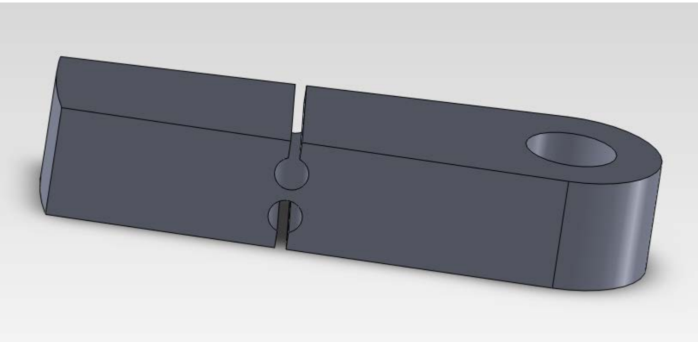


Figure 68 A pawl with notch hinge

6.7 The friction

For extra friction there are several ideas. Defatted the pawl and drum with acetone in a supersonic bath for extra friction. Another solution is to make a kind of dispenser who can put a kind of gel with for example silicon carbide particles in between the drum and pawl for extra friction. This solves also the problem that the carbide particles do not keep very well in nickel coating.
Another solution is to find anti-lubricants, some are already available.

6.8 Several ideas combined to concept

By using the several ideas a new concept is made. Integrating several components means saving space for bigger drums and a wider contact line. The key points:

- Spring is integrated in pulley
- Plain bearing integrated in the pawl
- The plain bearings are fitted with press fit in the holes.
- The shafts are fixed with a press fit
- The shafts are very accurate.

- The pulley is made from plain bearing material such as brass or self-lubricant plastic. The pulley runs in the phalanges.
- The system is fitted with plain brass bearings with very low play and high accuracy.
- The on the phalange the drum and the holes are wire spared. Also the pawl is wire sparked
- For less friction between the phalange- phalange and the phalange-pawl a very thin coating is applied. For example a ceramic coating from Blazer.

The benefits

- A low amount of parts.
- Easy to assemble
- Is capable of in this configuration to lower the contact forces 4 times. And 2 times less frictions coefficient is needed.
- It is easy to upscale the drum.
- Very low play and compliance .
- Maybe cheaper

Disadvantages

- Al lot of places get things stuck in the system like fingers of cloth. (Can be solved by a cover)
- In the depth it will takes more place when the drums get even bigger.

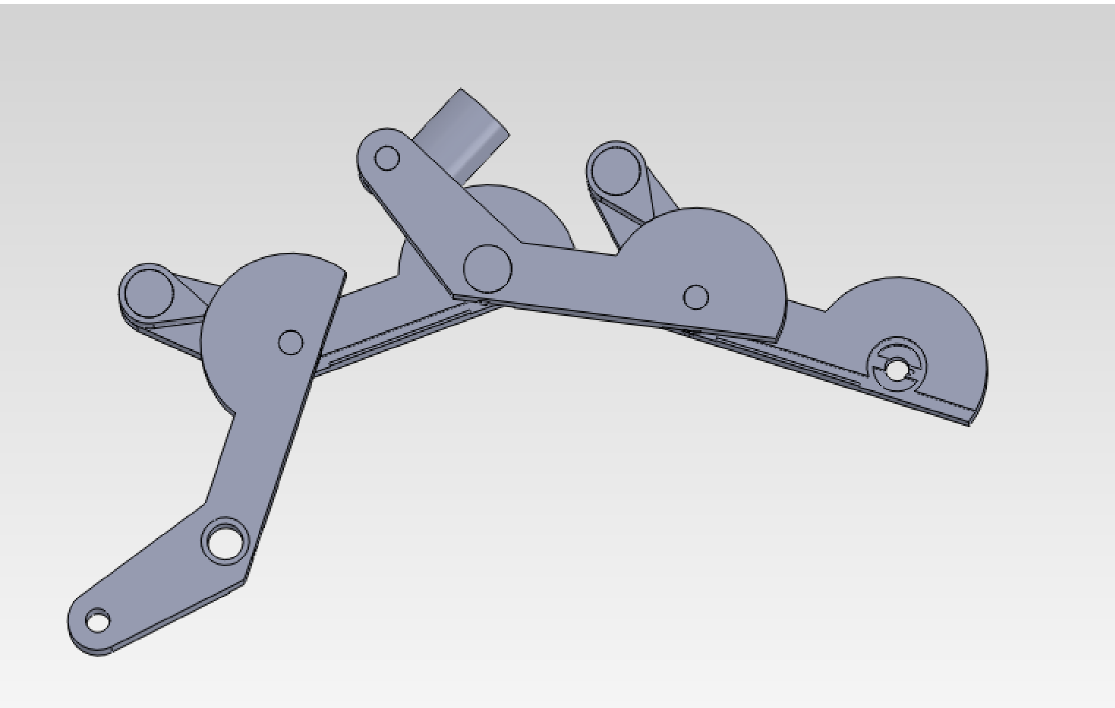


Figure 69 new concept

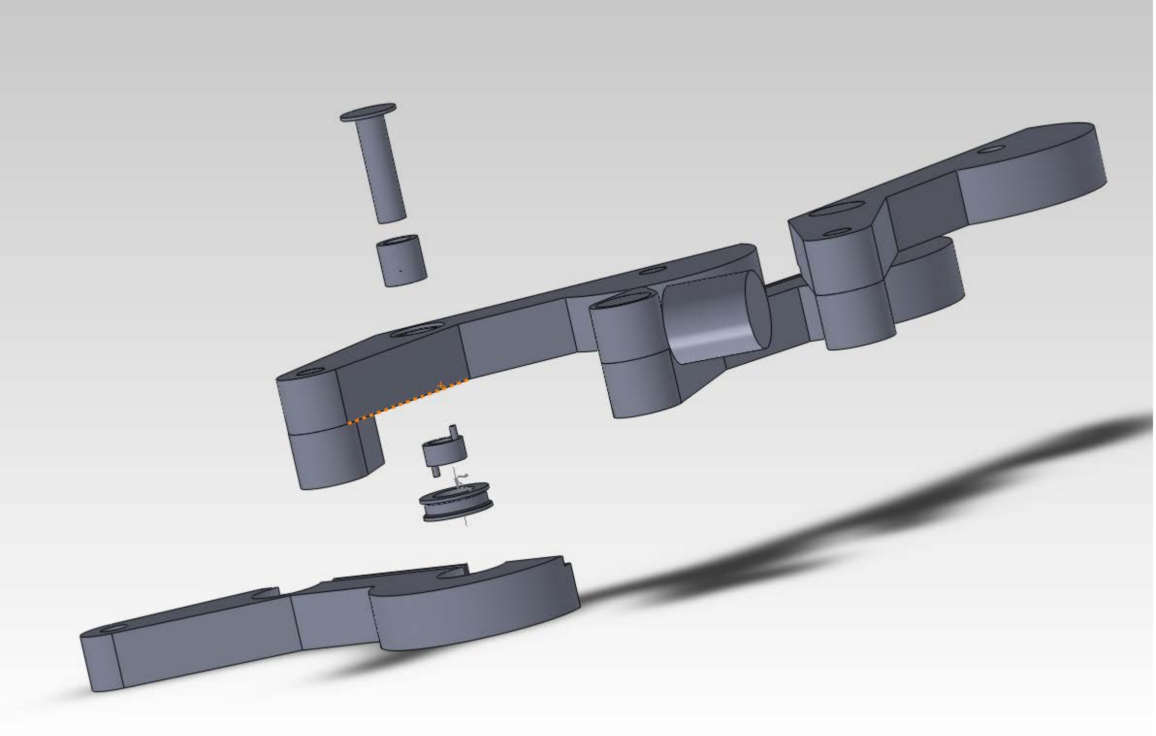


Figure 70 exploded view of the new concept

6.9 Random ideas

There also some random ideas.

- Use instead of actuators piezoelements. These can applied high forces and use small space. It can be used so that self-locking effect isn't used and a higher friction coefficient isn't needed.
- Use instead of actuators air pressure driven motion.
- Instead of using a spiral surface on the pawl, for using as composition of the tolerances and play, use an oval shaft.
- Make in the spiral the contact angle different. An use for it is to changing the forces in the system. For example a spring is put in the pawl or behind the bearings, when the forces become higher the distance between the shafts become greater(the spring becomes smaller). The pawl change angle and then the spiral is made in a way that the contact angle change linearly. So it change to a higher contact angle and the forces become lower. When the contact angle becomes too high for the friction the pawl slips and stops working and it works also as a.
- Another use of changing the shape of the spiral is for letting the spiral compensate for example a rubber coating on the drum. So is possible to get the same contact angle corner all the time.
- For putting the pawl against the drum is to put shrew thread on the pawl and the shaft of the pawl so the pawl can be driven vertical.

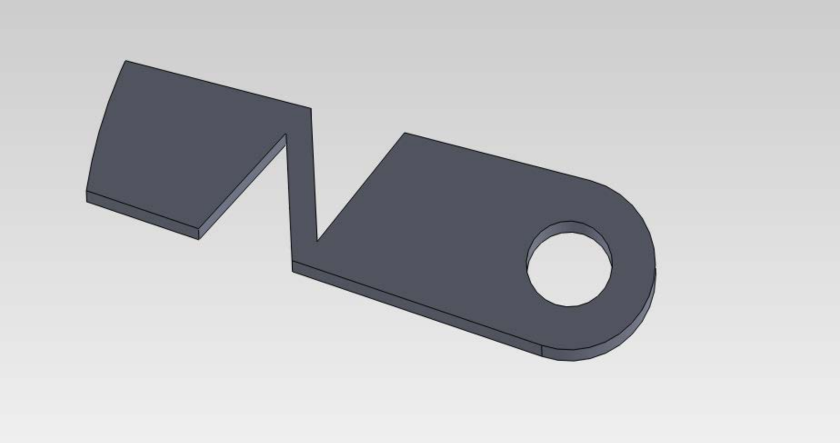


Figure 71 A schematic spring in a pawl

7. Conclusions and recommendations

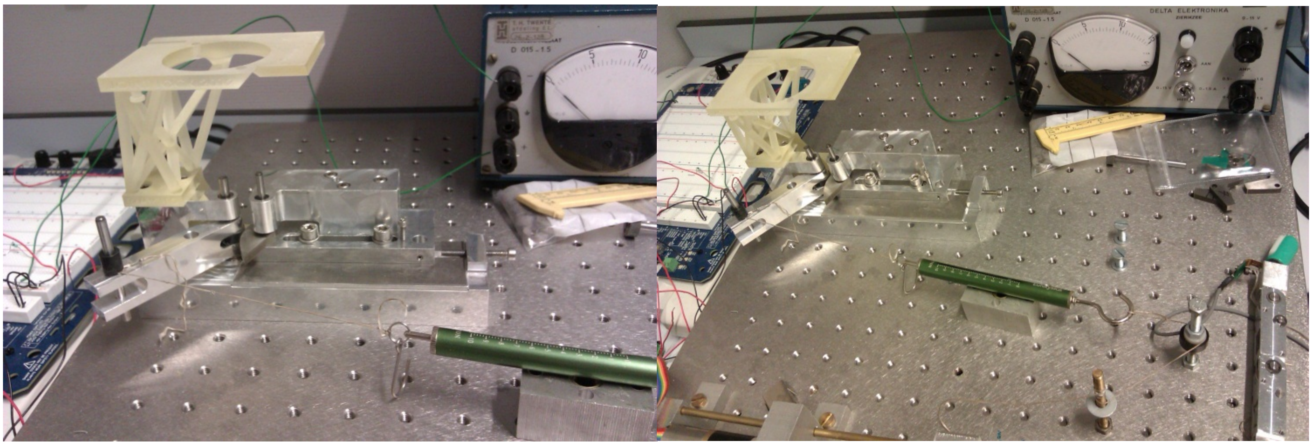


Figure 72 test rig

The pawls made by Pieterse have defects in different ways. The recommendation here is to keep this in mind while doing further testing or implementing them.

Research done on why flat surfaces do not work, resulted in that if there is a minor error in the distance between the shafts of the drum and the pawl results in a large deviation in the contact angle.

In theoretical research in the dynamics of the locking mechanism pointed out that the position of the contact point, the shaft of the pawl and the shaft of the drum in respects to one another are very crucial for the function of the joint lock.

In a model that is made to test what the theoretical influence is from different dimension of the lock on the forces, needed friction and contact tensions in the system. Conclusion of that model is that the forces and needed friction in the system with chancing radius of the drum, the contact length and the contact angle can be influence greatly.

There are several trial tests done to compare coatings. First the 3M 8592 film, the pawls sticks but slips on the coating, the coating is not suitable. Second coating is the Keratherm Gluey Soft Coating after the first test the coating breaks apart. This coating is not suitable. The third coating, the plastic dip coating, works very well at a thickness of 0.63 mm. However this coating is not strong enough in the end. Recommendation is to search for a type of rubber who is stronger, same elasticity and stick better on the drum.

A friction test is done with a drum made of 'automatenstaal' and wire sparked drum. In the result we see that the wire sparked drum the friction coefficient is between 0.05 and 0.098. The friction coefficient of the 'automatenstaal' drum is between the 0.126 and 0.138. This is a reflection of the practical experience.

The test results of friction of the silicon carbide coating show that 10-11 micron silicon carbide particles increase the friction to a friction of 0.40. With nickel as carrier of the coating we see that the friction drops due to the fact that the particles are pushed away during use.
There are two recommendations need to be closer examined. The first is the effects of the misalignment of the pawl and drum. Second is to find ways to put enough selenium carbide between the contact line of the pawl and the drum. Options are to find a better carrier compared to nickel, this is unlikely due to the fact that it has te be readily available and cost effective. Options to be explored are polymers (epoxy's etc.) and other metals. Another approach is adding the silicon carbide in use as an active anti-lubricant. For example by adding a silicon carbide gel with 10-11 micron silicon carbide particles.

Recommendations for future work are research on coatings with silicon carbide particles, rubber based coatings and ceramic coatings (like TIN). Some research as is described in this report what is not completed yet should be carried out. Several suggestions for improvement of the joint lock made in this report should be investigated. Taking in account the difficulties of this Joint lock design alternative designs could be considered.

8. Figures

9. Works Cited

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S.A.J. Spanjer, R. Balasubramanian, A.M. Dollar and J.L. Herder. (2012). Underactuated Gripper that is able to Convert from Precision to Power Grasp by a Variable Transmission Ratio.

Pubised at: The Second ASME/IEEE International Conference on Reconfigurable Mechanisms and Robots 2012 (ReMAR)

10. Appendix

10.1 Attachment of Underactuated Gripper that is able to Convert from Precision to Power Grasp by a Variable Transmission Ratio

S.A.J. Spanjer, R. Balasubramanian, A.M. Dollar and J.L. Herder, "Underactuated Gripper that is able to Convert from Precision to Power Grasp by a Variable Transmission Ratio", *The Second ASME/IEEE International Conference on Reconfigurable Mechanisms and Robots (ReMAR),* 2012.

(next page goes further)

Pulley with variable radius

Appendix ...

Abstract—**In this appendix the theory about the pulley with variable radius is explained. The pulley consists small pulleys, pins and disks with straight (radial) and curved slots. A set of mathematical equations is solved to get the optimal curvature of the slots. These slots are designed to have a relatively large angle with the radial slots in order to reduce friction and allow smooth movement of the pins. When the coefficient of friction between the pins and the slots is known, the minimal curvature of the slots can be calculated.**

I. Introduction

THE proximal pulley employs a mechanism that allows the effective radius to be changed by rotating a top and bottom plate with respect to one another (Fig. 1a). Shafts with small pulleys around which the tendon is routed (i.e. the effective pulley) are positioned in the slots of both plates. As the plates rotate with respect to one another, the shafts move radially, changing the effective pulley radius. The 2 plates are mirrored in order to get a symmetrical system that prevents shafts popping out of the slots (Fig. 1b). So there are 2 plates with straight slots and 2 plates with curved slots and these plates are aligned. The whole system is connected to the proximal shaft with bearings, so it can rotate freely.

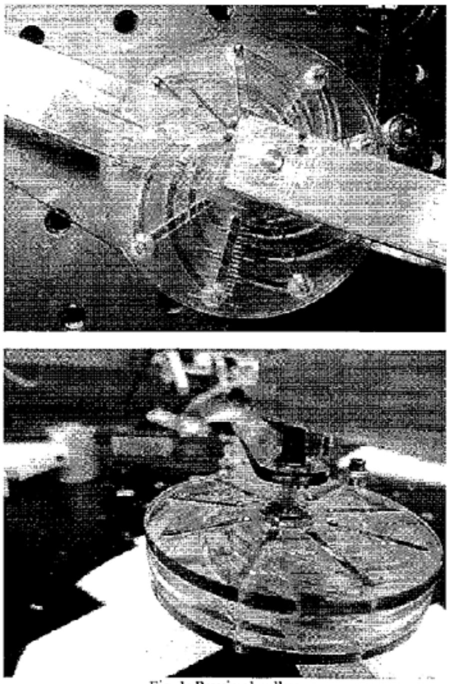


Fig. 1. Proximal pulley

The angle between a straight slot and the tangent of the curved slot at the position of the pin is constant for every radius. The curvature of the curved slot is calculated in section II. In section III the minimal angle between the straight slot and the tangent of the curved slot is calculated.

II. Curvature Calculations for a Constant Angle

In Fig. 2 a straight slot and a curved slot are depicted. For every angle θ the angle α must be constant. In Cartesian coordinates this results in the following relation:

$$\frac{dy}{dx} = \tan\left(\frac{\pi}{2} + \theta - \alpha\right)$$

A transformation to polar coordinates is achieved by

$$x = r(\theta)\cos\theta$$
$$y = r(\theta)\sin\theta$$

where $r(\theta)$ is a function of θ. Differentation of x and y with respect to θ and combining this with (1) results in the following differential equation:

$$\frac{r'\sin\theta + r\cos\theta}{r'\cos\theta - r\sin\theta} = \tan\left(\frac{\pi}{2} + \theta - \alpha\right)$$

where r' and r are functions of θ. Eq. (3) can be written as follows:

$$\frac{1}{r}dr = -\frac{\left(\cos\theta + \sin\theta\tan\left(\frac{\pi}{2} + \theta - \alpha\right)\right)}{\sin\theta - \cos\theta\tan\left(\frac{\pi}{2} + \theta - \alpha\right)}d\theta$$

After integrating both sides and some mathematical

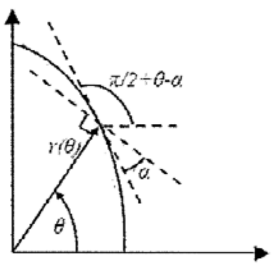


Fig. 2: Straight and curved slots

operations the solution is obtained:

$$r = c\, e^{\theta \tan\alpha}$$

c is a constant. If the smallest and largest possible diameter are known and the angle over which the top and bottom plate must rotate to achieve this, then c and α can be calculated. It is also possible to calculate the minimal value of α for which the disks still rotate.

III. Minimal angle between slots

The minimal value of α can be calculated from the force equilibrium between the pins and the slots (Fig. 3). The coefficients of friction μ_s between pin and straight slot and μ_c between pin and curved slot are used. By rotating the disk with straight slots the force F_A and the friction force $F_{F,A}$ apply on the pin. As a consequence, the contact force F_N and the friction force $F_{F,N}$ apply also on the pin. We assume that both $F_{F,A}$ and $F_{F,N}$ reach the maximum value before the equilibrium breaks:

$$F_{F,N} = \mu_c F_N$$
$$F_{F,A} = \mu_s F_A$$

Together with the two force equilibrium equations there are four equations, so the four unknowns (α, F_N, $F_{F,N}$, $F_{F,A}$) can be solved. This results in the following:

$$\alpha = \arctan\frac{1-\mu_c\mu_s}{\mu_c+\mu_s}$$

If the coefficients of friction are large, then α should be small. If both coefficients of friction are equal to 0.3, then the maximum value of α is 56° . For a larger angle the pins will not slide.

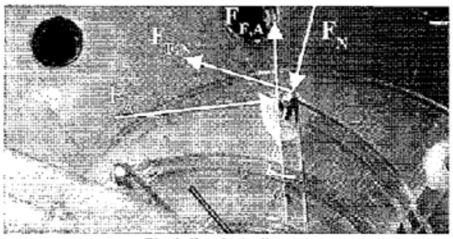
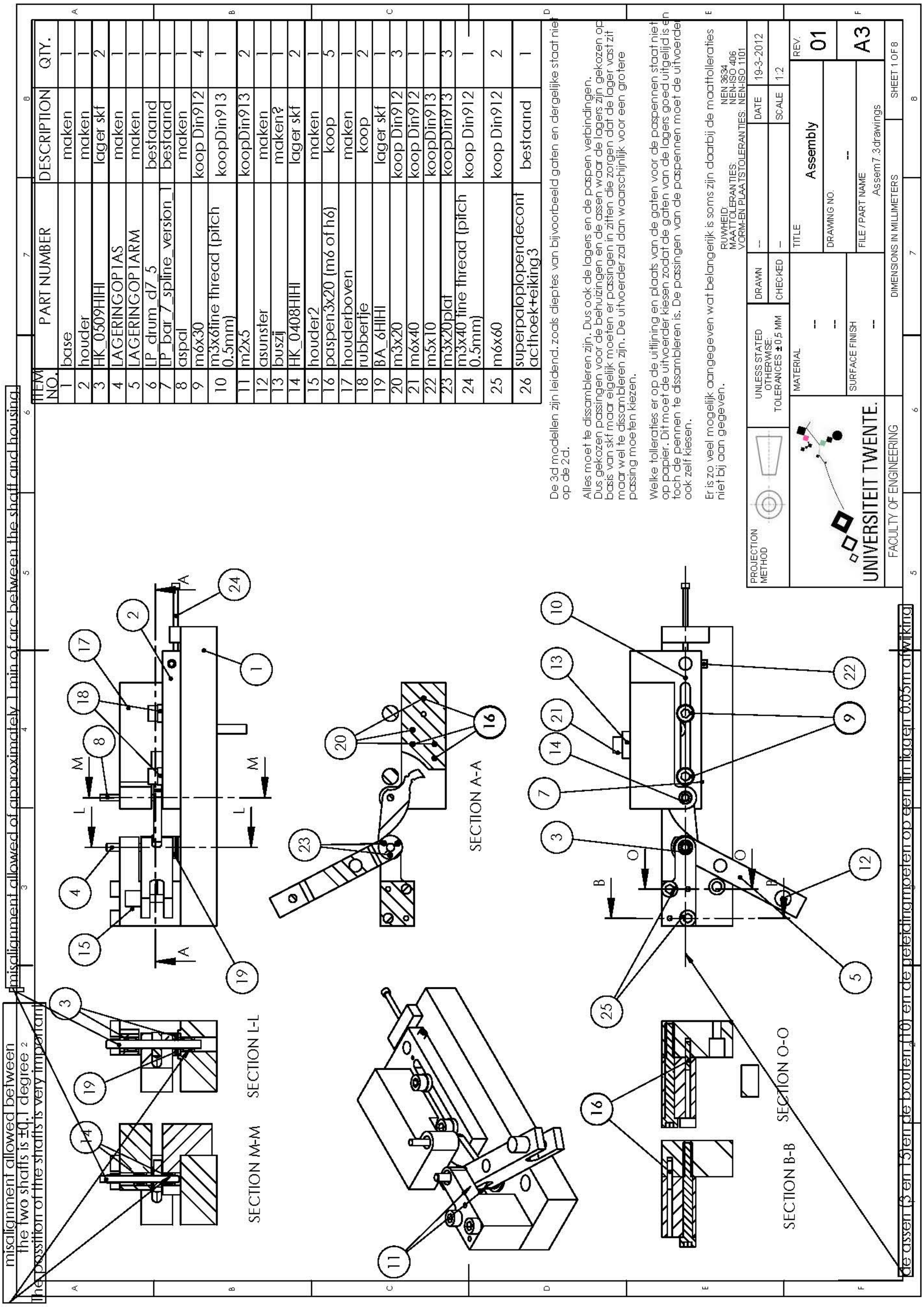
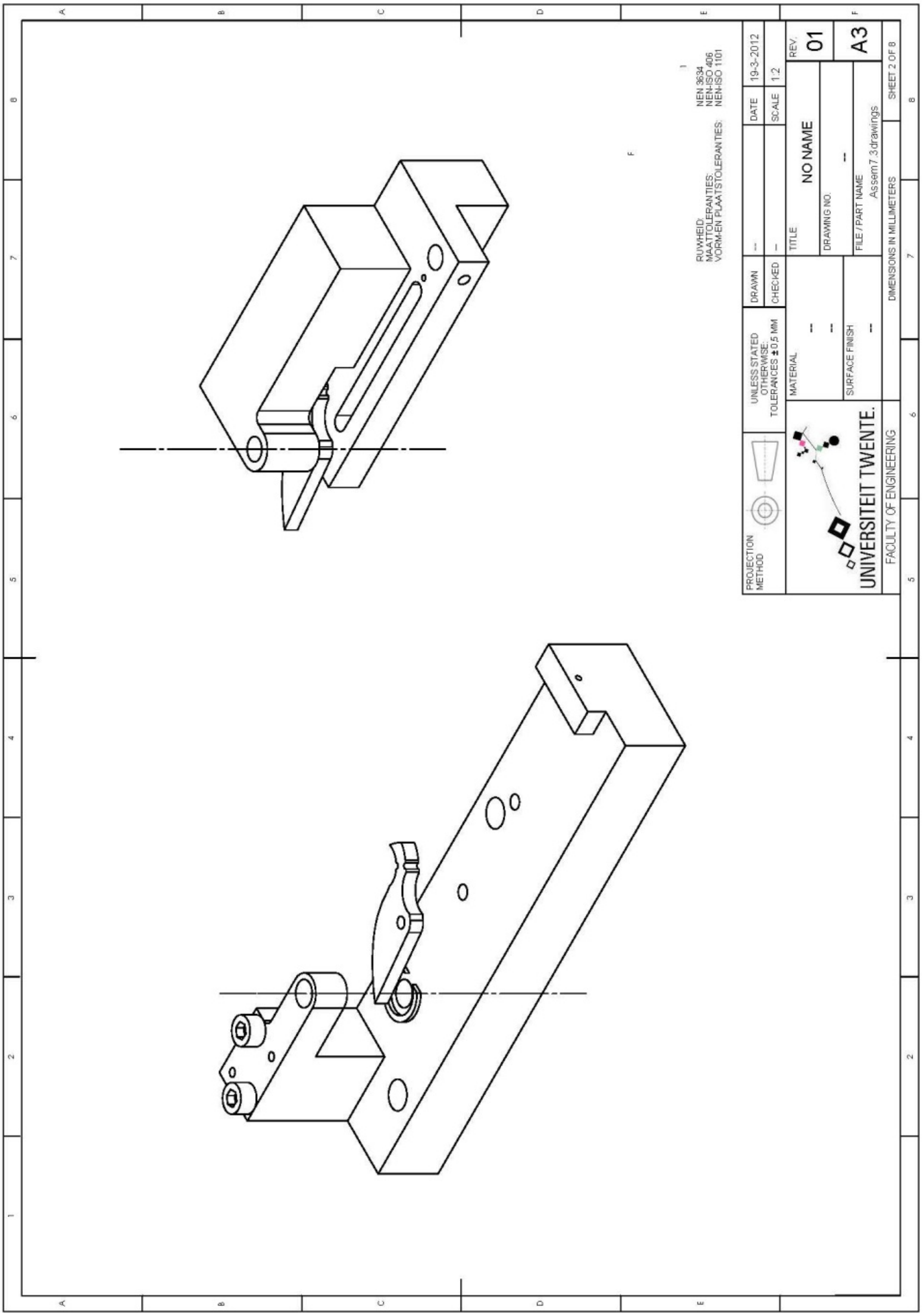


Fig. 3: Free body diagram

10.2 2d drawings test rig

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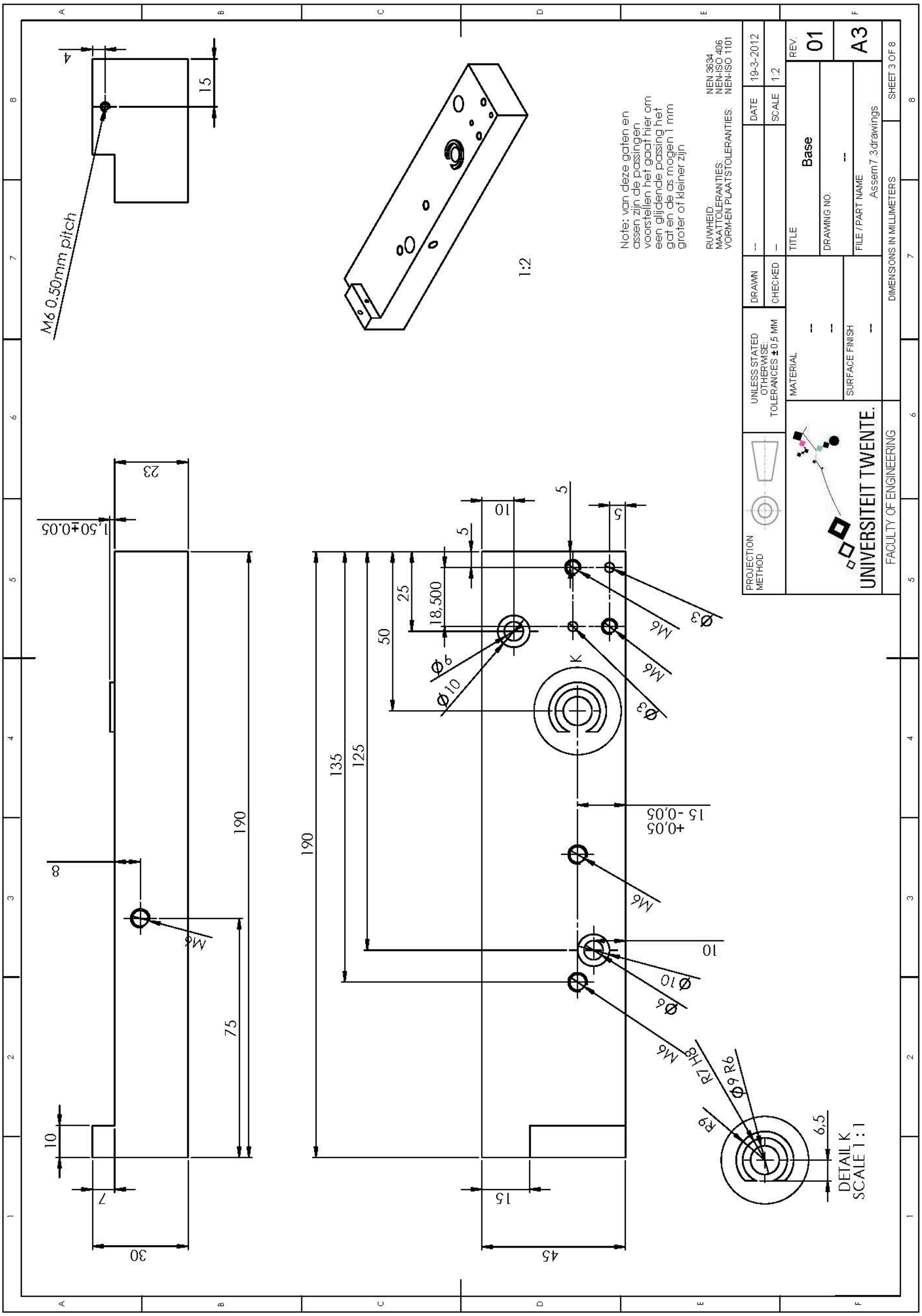
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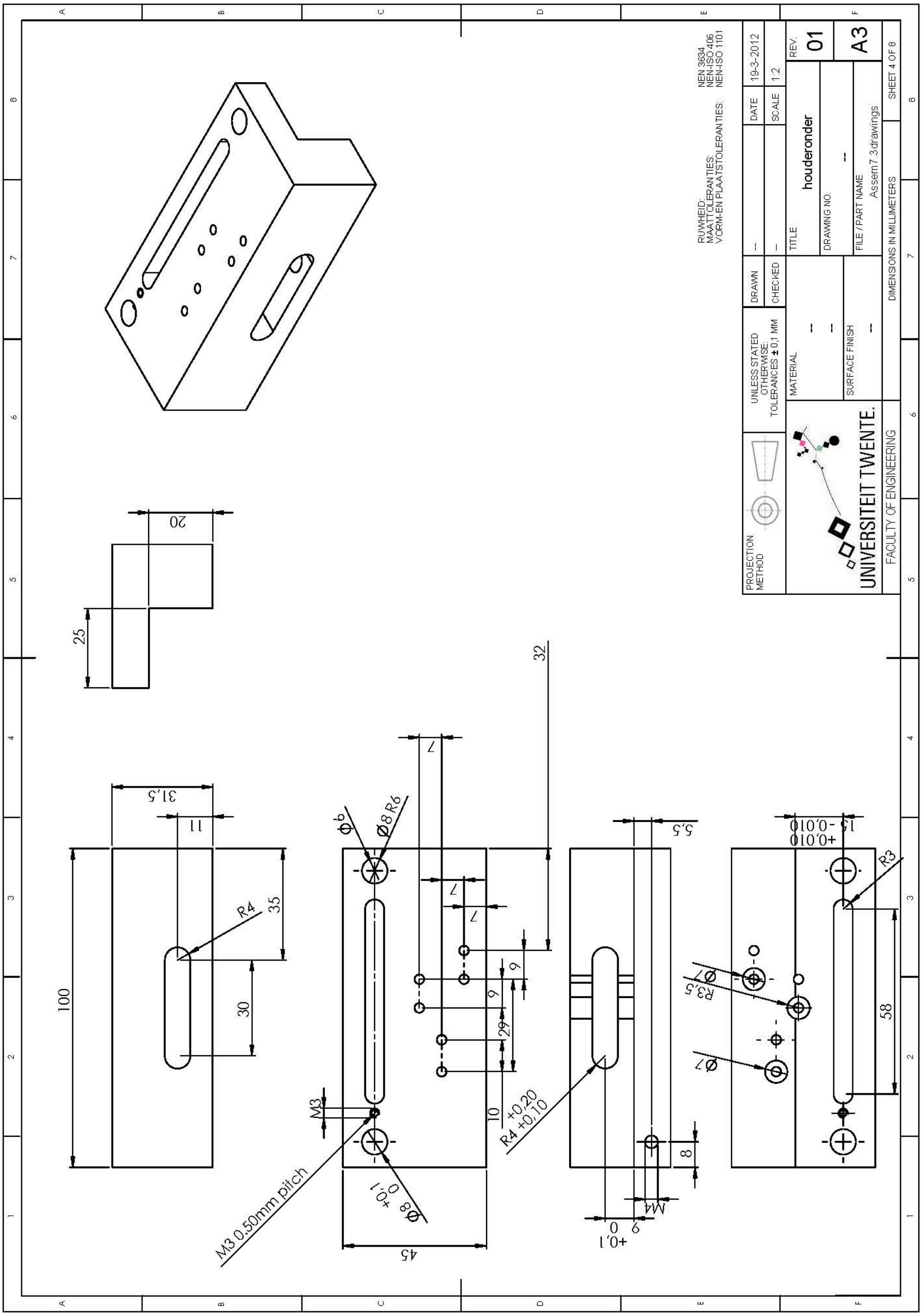
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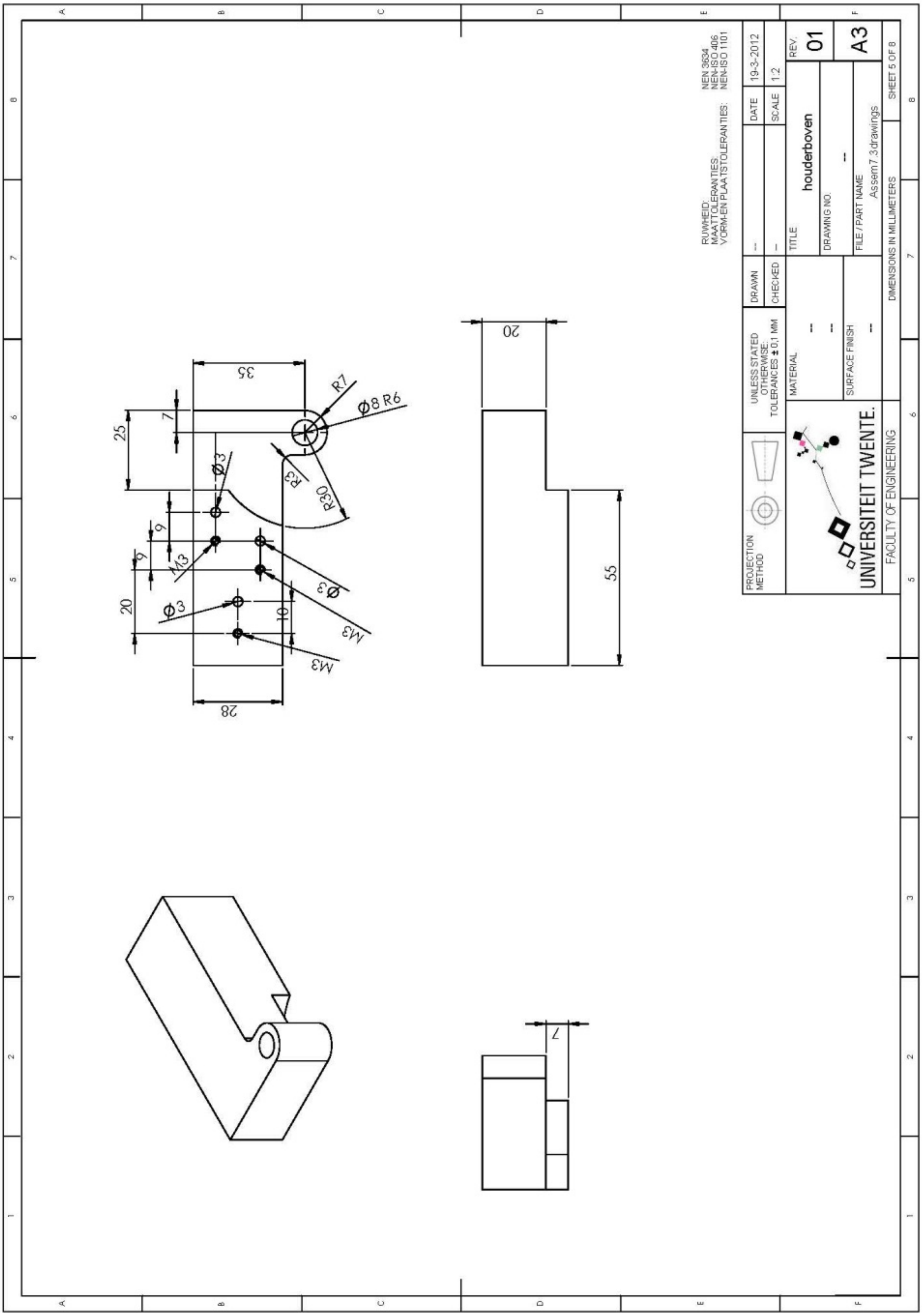
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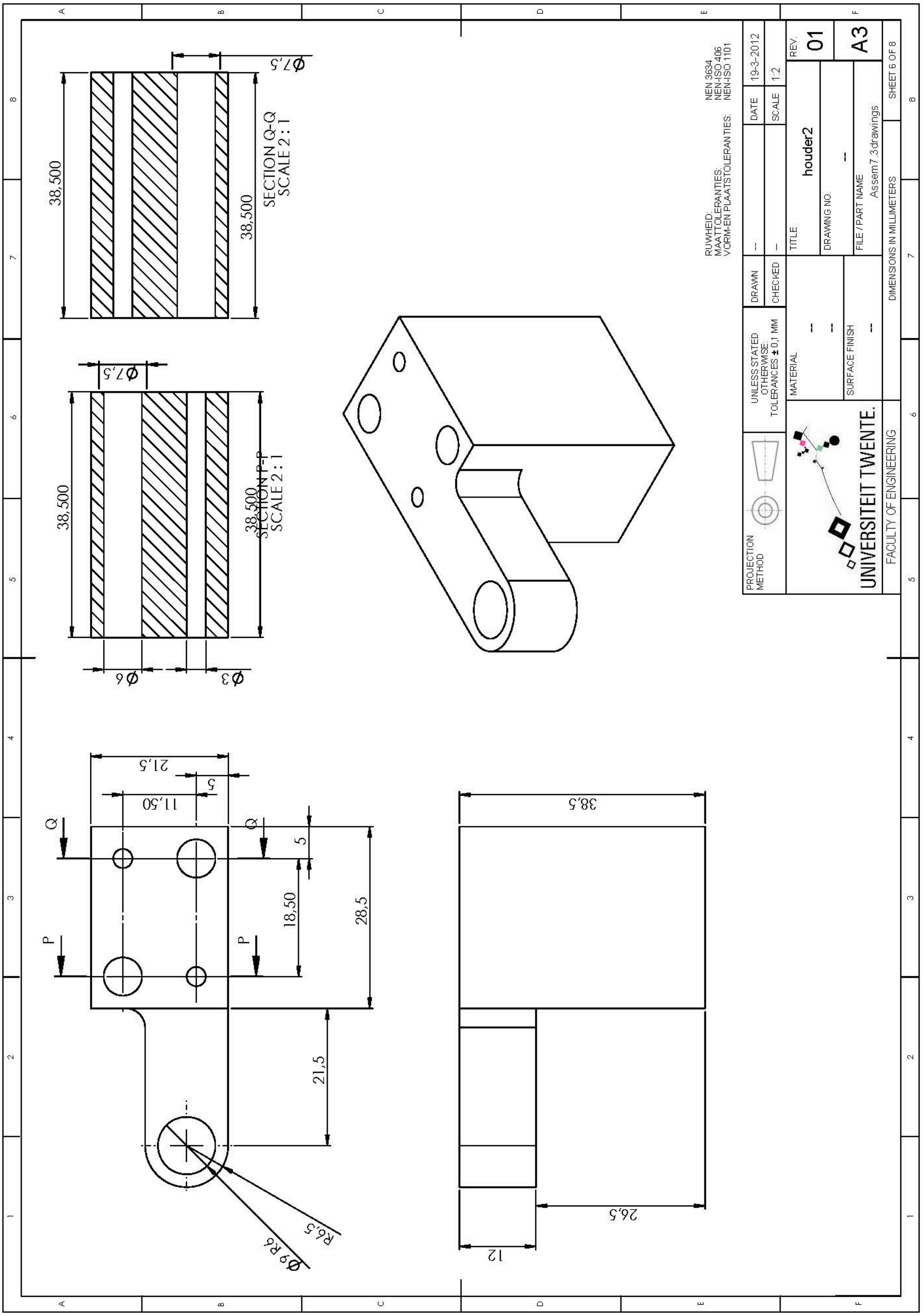
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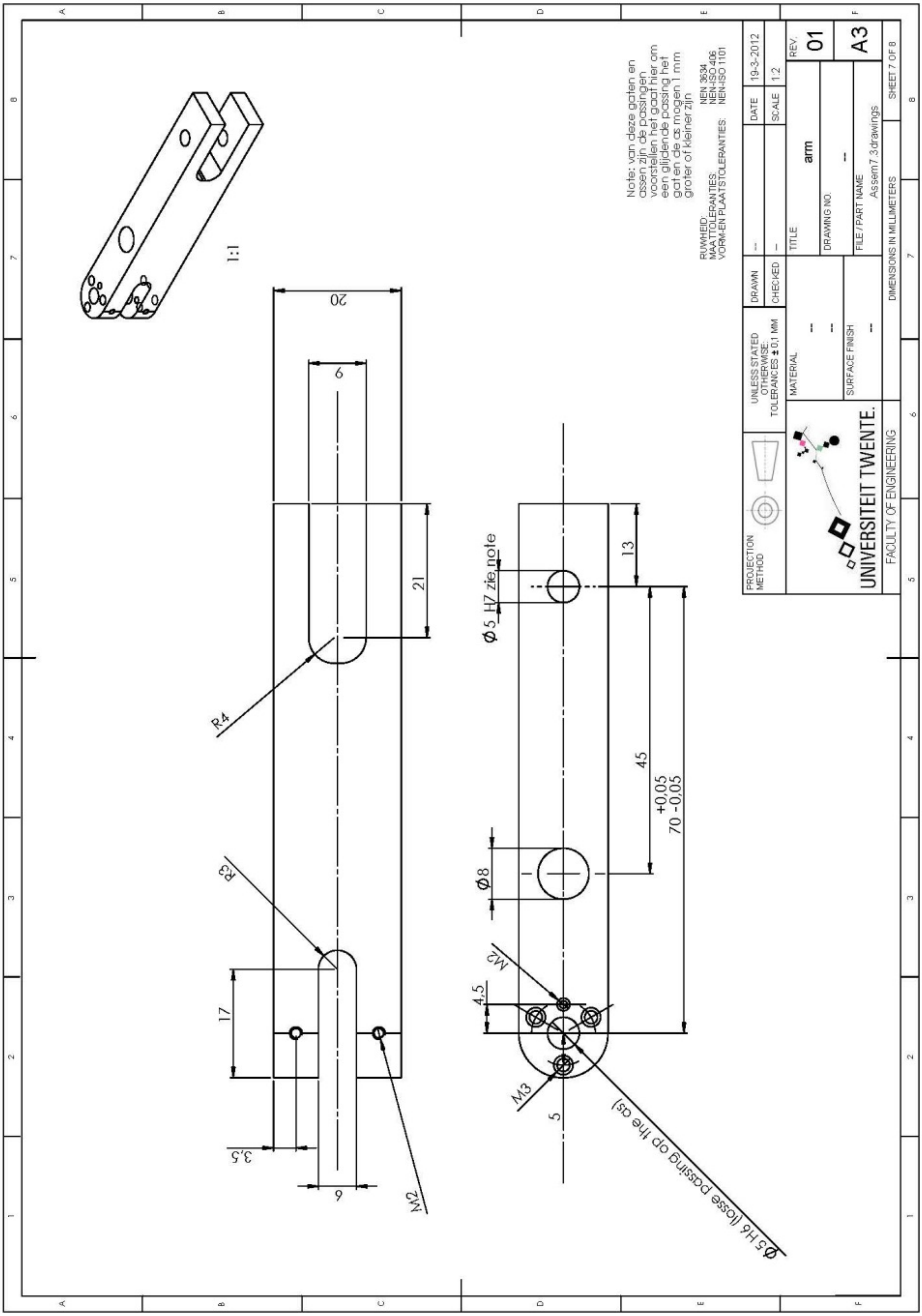
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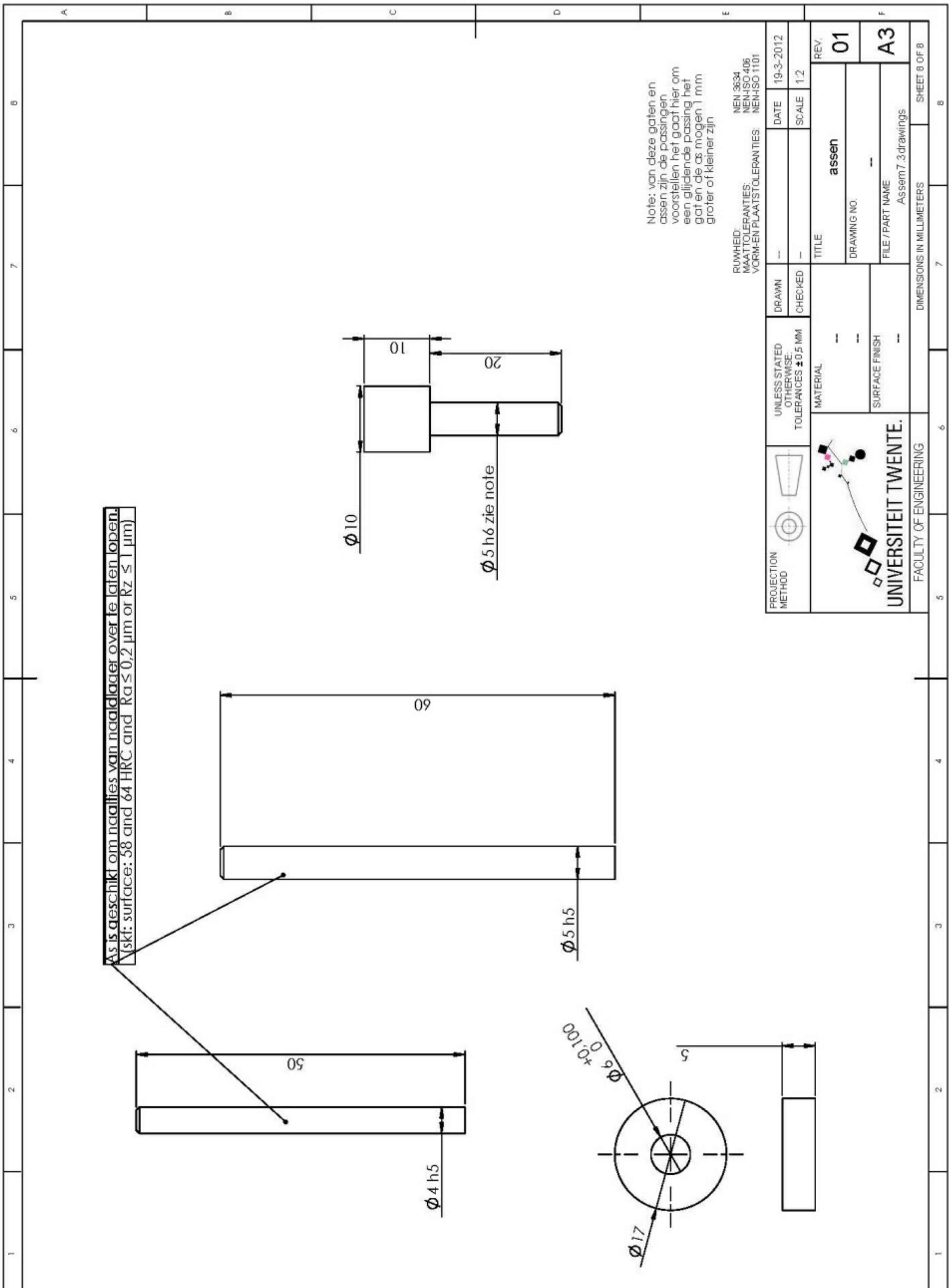
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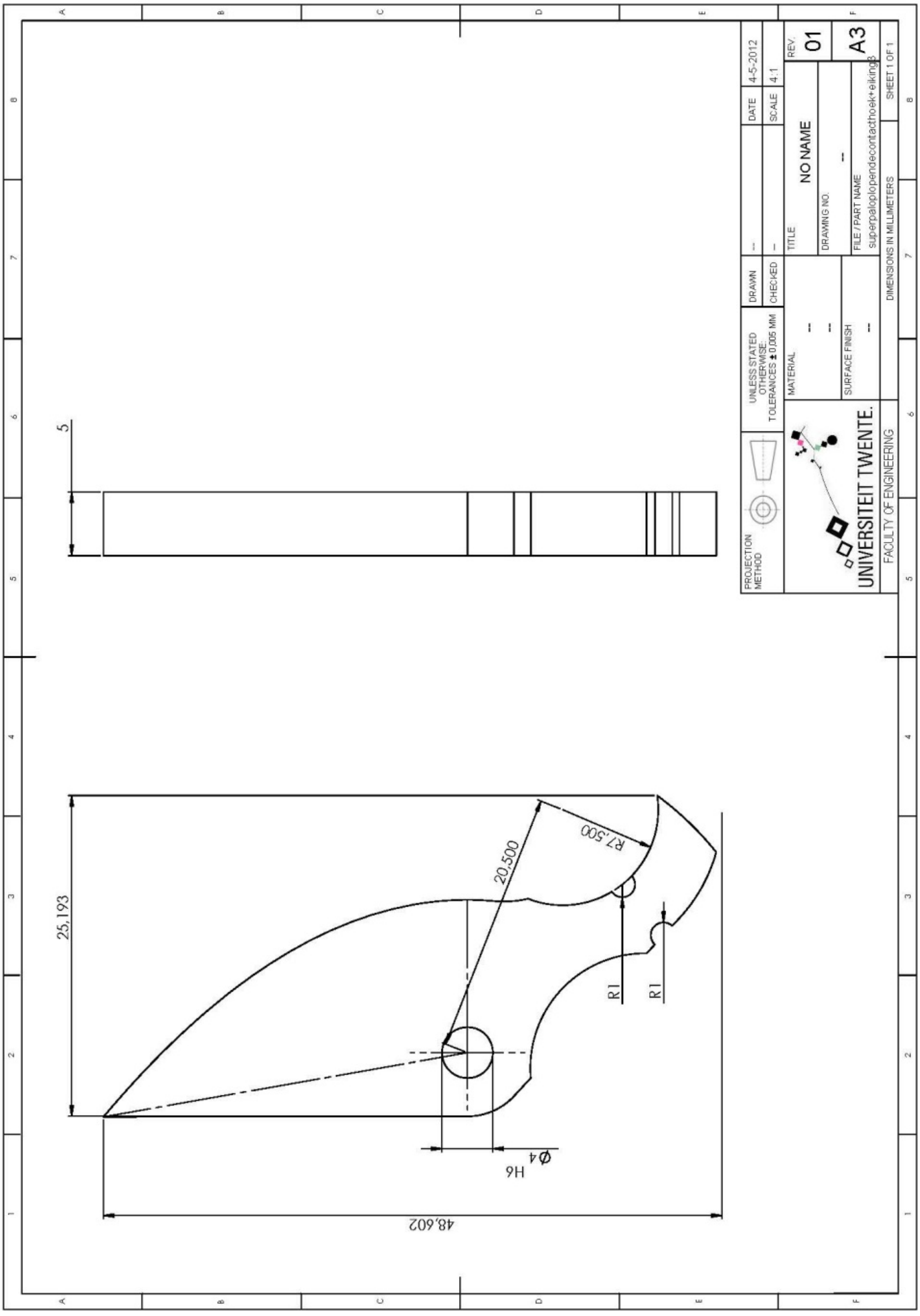
10.3 2d drawing superpawl



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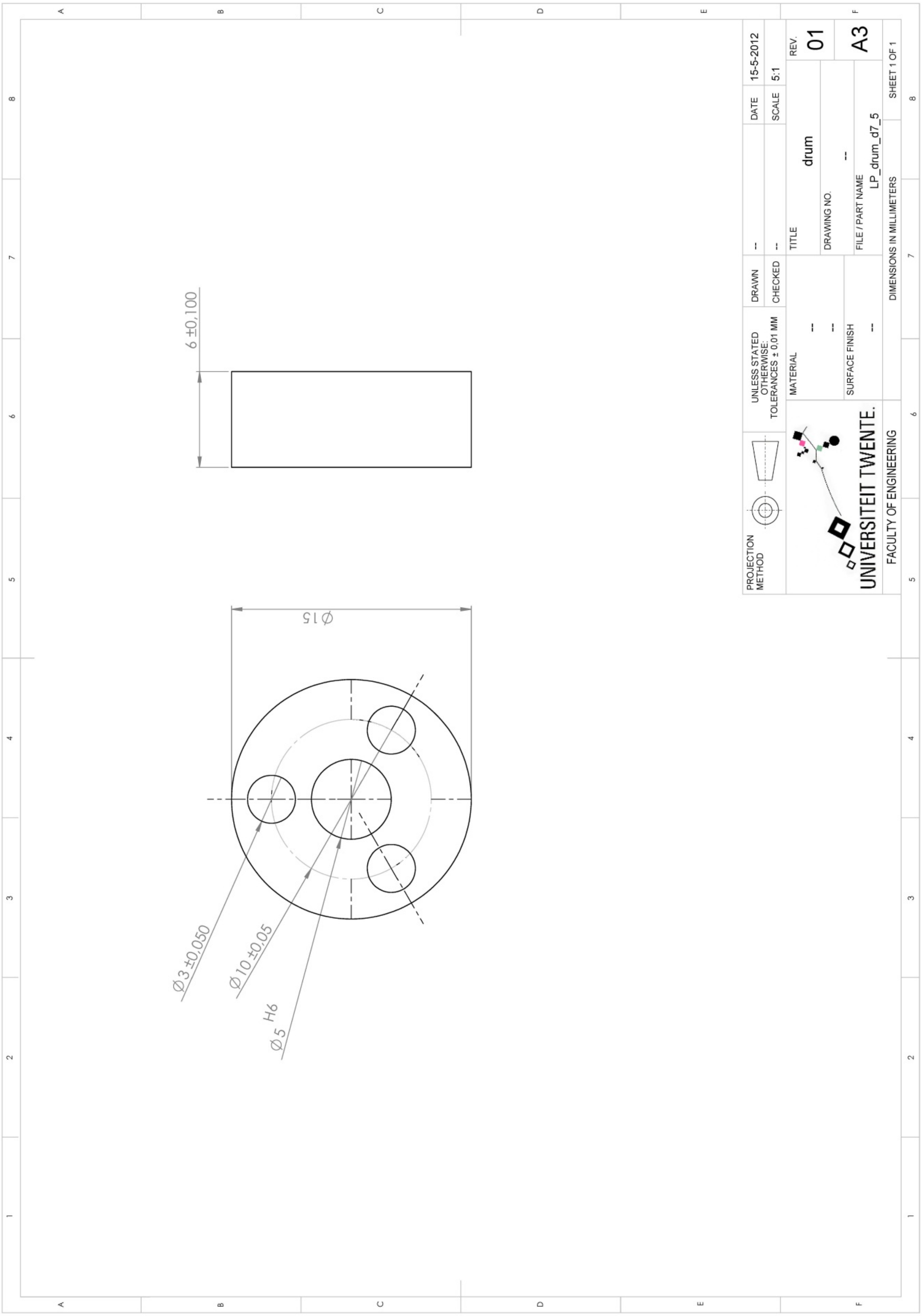
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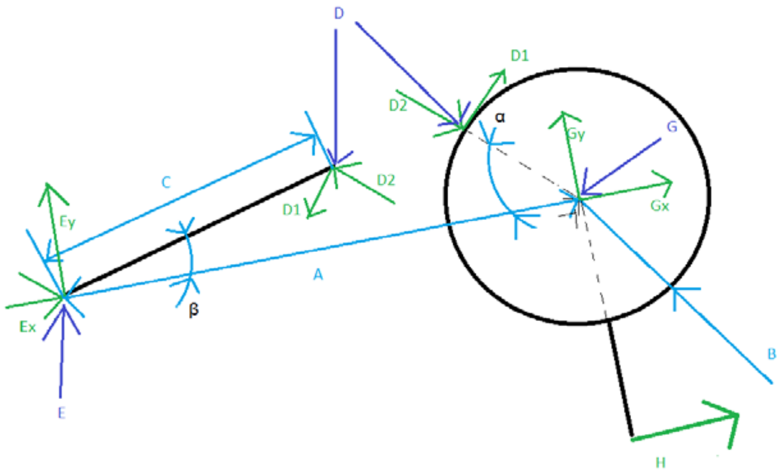
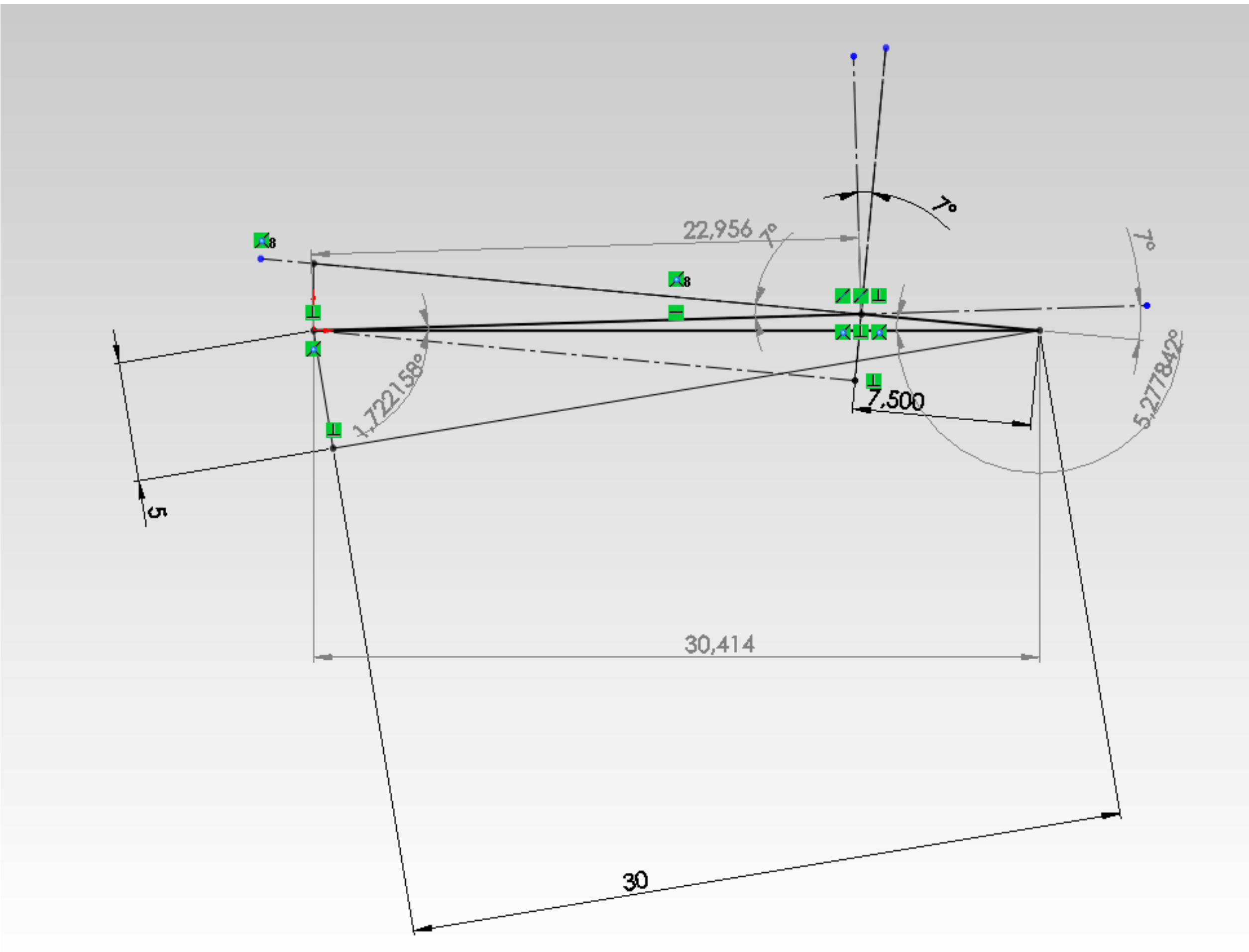
TITLE drum
DRAWING NO. --
FILE / PART NAME LP_drum_d7_5
REV. 01
A3
UNIVERSITEIT TWENTE.
FACULTY OF ENGINEERING
DIMENSIONS IN MILLIMETERS
SHEET 1 OF 1

10.5 Calculation model

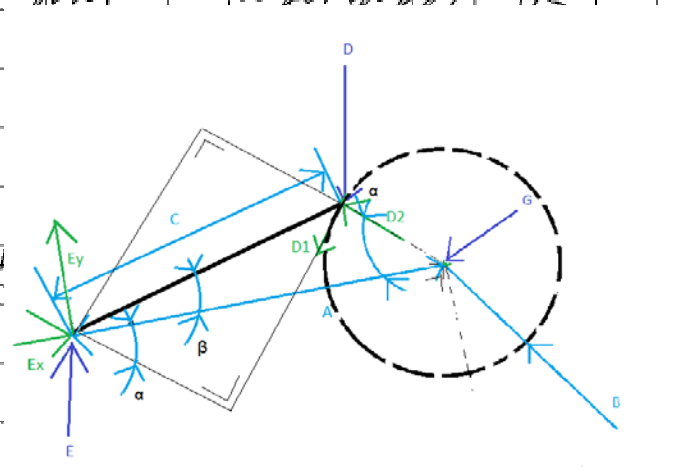
reactie	hoe berekent/ uitleg			in mm or N	in mm or N	in m or n or Nm
Constanten:				ingetypt	gerekent	gebruikt
μ		=	tan(D1/D2)	0,120	0,09	0,120000
arm H		=		100,00		0,100000
hoeken nieuw						
A		=		30,27	30,41	0,030270
B		=		7,50		0,007500
Hoek pal		=		7,00		7,000000
γ		=			173,00	173,000000
β		=	sinus regel		1,73	1,730342
α		=			5,27	5,269658
C		=			22,8121	0,022812
radius spiraal		=	bij benadering		37,50	0,037500
contact lengte		=		5,00		0,005000
v1		=	poisson ratio pawl	0,30		0,300000
v2		=	poisson ratio bus	0,30		0,300000
e1		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
e2		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
			Reduce rol radius		0,01	0,006250
			reduced Young's modulus		230769230769,23	230769230769,231000
Krachten						
H		=		20,00		20,000000
Momer	moment veroorzaakt door H om G	=	H*"arm H"		2,00	2,000000
D1	krachten plaatje bus	=	"moment H"/B		266,67	266,666667
D2	krachten plaatje pal met de momenten stelling met het moment om E z	=	D1*((C*SIN(90-α-β)/(A*SIN(α)))		2171,83	2171,825714
Ex	krachten plaatje pal in krachten in x richting	=	D2*cos(α)+D1sin(α)		2187,14	2187,137977
Ey	krachten plaatje pal krachten in y richting	=	D1cos(α)-D2sin(α)		66,07	66,072019
Ex	via momenten om snijpunt Ey en D1 in krachten plaatje pal	=	D1*((A*(COS(α))-B)/(A*SIN(α))		2171,83	2171,825714
Ey	via momenten om snijpunt Ex en D1 in krachten plaatje pal	=	(D1*B)/A		66,07	66,072019
Ex	via hoek beta	=	Ey/TAN(β/180*PI())		2187,14	2187,137977
Ey	via hoek beta	=	Ex*TAN(β)		66,07	66,072019
E	krachten plaatje bus in krachten in x richting	=	(D1^2+D2^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2172,83	2172,830514
E	krachten plaatje bus krachten in y richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
β(controle))		=	tan^-1(Ey/Ex)		1,74	1,742534
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
Gx		=	-H-D1sin(α)-D2cos(α)		-2207,14	-2207,137977
Gy		=	-D1cos(α)+D2sin(α)		-66,07	-66,072019
G		=	(Gx^2+Gy^2)^0,5		2208,13	2208,126709
locking						
μ		=			0,12	0,120000
		=	D1/D2		0,12278	0,122785
Wmax		=			260,62	260,619086
is er zelf locking		=			0,00	0,000000
herts contact spanning(n/mm)		=	(((D2/"contact lengt")*: Reduced Young's modules")/(2*pi()*"reduced rol radius"))^0,5		1597,67	1597,667834

10.5.1 Formulas and vls's





met berekening D



$$\curvearrowleft + \sum M_E = 0$$

$$0 = D_1 \cdot C \cdot \sin(180 - \beta - (180 - (90 - \alpha))) - D_2 A \sin \alpha$$

$$D_1 \cdot C \cdot \sin(90 - \beta - \alpha) = D_2 A \sin \alpha$$

$$D_2 = \frac{D_1 \cdot C \cdot \sin(90 - \beta - \alpha)}{A \cdot \sin \alpha}$$

pal

$\Sigma F_x = 0 \quad 0 = E_x - D_2 \cos\varphi - D_1 \sin\varphi$

$0 = E_y - D_1 \cos\varphi + D_2 \sin\varphi$

$\Sigma M = 0 \quad 0 =$

lens

$\Sigma F_x = 0 \quad 0 = G_x + D_2 \cos\varphi + D_1 \sin\varphi + H$

$\Sigma y = 0 \quad 0 = G_y - D_2 \sin\varphi + D_1 \cos\varphi$

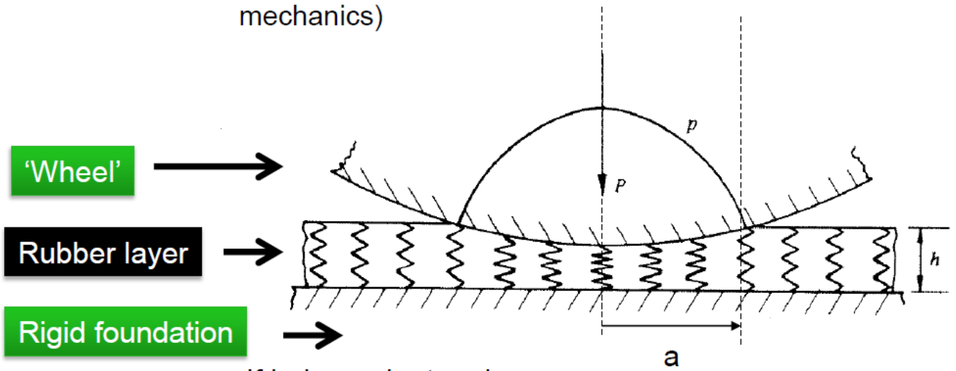
$\Sigma M_G = 0 \quad 0 = -M_H + D_1 \cdot B$

Hert vs elastic foundation

Elastic foundation model	Hertz line contact
Pressure distribution	Pressure distribution
$p(x)=\frac{E}{2Rh}\left(a^2-x^2\right)$	$p(x)=\frac{2F}{\pi bl}\sqrt{1-\left(\frac{x}{b}\right)^2}$
Half contact width	Half contact width
$a=\sqrt{2\delta R}\Rightarrow a=\left(\frac{3PRh}{E}\right)^{\frac{1}{3}}$	$b=\left(\frac{4PR}{\pi E}\right)^{\frac{1}{2}}$

Elastic foundation model

- Elastic foundation model on rigid backing (see book Johnson contact mechanics)



If independent springs:

$$u_z(x,y)=\begin{cases}\delta-z(x,y)\text{ if }\delta>z\\ 0\text{ if }\delta\le z\end{cases}\qquad p(x,y)=\frac{E}{h}u_z(x,y)$$

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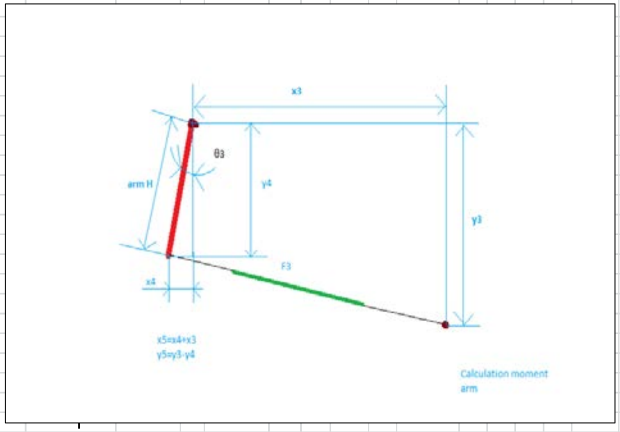
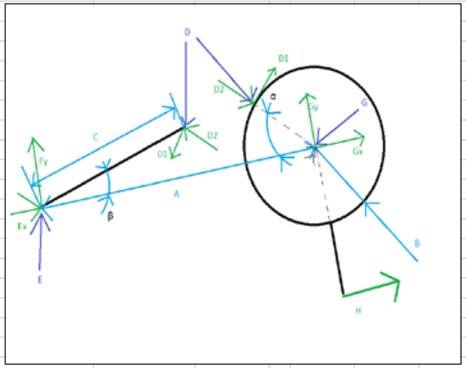
SAXION

reactie	hoe berekent/ uitleg			in mm or N	in mm or N	in m or n or Nm
Constanten:				ingetypt	gerekent	gebruikt
μ		=	tan(D1/D2)	0,120	0,09	0,120000
arm H		=		100,00		0,100000
hoeken nieuw						
A		=		30,27	30,27	0,030270
B		=		7,50		0,007500
Hoek pal		=		7,00		7,000000
γ		=			173,00	173,000000
β		=	sinus regel		1,73	1,730342
α		=			5,27	5,269658
C		=			22,8121	0,022812
radius spiraal		=	bij benadering		37,50	0,037500
contact lengte		=		5,00		0,005000
v1		=	poisson ratio pawl	0,30		0,300000
v2		=	poisson ratio bus	0,30		0,300000
e1		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
e2		=	elastic moduli pawl (n/m^2)	210000000000,00		210000000000,000000
			Reduce rol radius		0,01	0,006250
			reduced Young's modulus		230769230769,23	230769230769,231000
Krachten						
H		=		20,00		20,000000
Momer	moment veroorzaakt door H om G	=	H*"arm H"		2,00	2,000000
D1	krachten plaatje bus	=	"moment H"/B		266,67	266,666667
D2	krachten plaatje pal met de momenten stelling met het moment om E z	=	D1*((C*SIN(90-α-β)/(A*SIN(α)))		2171,83	2171,825714
Ex	krachten plaatje pal in krachten in x richting	=	D2*cos(α)+D1sin(α)		2187,14	2187,137977
Ey	krachten plaatje pal krachten in y richting	=	D1cos(α)-D2sin(α)		66,07	66,072019
Ex	via momenten om snijpunt Ey en D1 in krachten plaatje pal	=	D1*((A*(COS(α))-B)/(A*SIN(α))		2171,83	2171,825714
Ey	via momenten om snijpunt Ex en D1 in krachten plaatje pal	=	(D1*B)/A		66,07	66,072019
Ex	via hoek beta	=	Ey/TAN(β/180*PI())		2187,14	2187,137977
Ey	via hoek beta	=	Ex*TAN(β)		66,07	66,072019
E	krachten plaatje bus in krachten in x richting	=	(D1^2+D2^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
E	krachten plaatje bus in krachten in x richting	=	(Ex^2+Ey^2)^0,5		2172,83	2172,830514
E	krachten plaatje bus krachten in y richting	=	(Ex^2+Ey^2)^0,5		2188,14	2188,135746
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
β(controle))		=	tan^-1(Ey/Ex)		1,74	1,742534
β(controle))		=	tan^-1(Ey/Ex)		1,73	1,730342
Gx		=	-H-D1sin(α)-D2cos(α)		-2207,14	-2207,137977
Gy		=	-D1cos(α)+D2sin(α)		-66,07	-66,072019
G		=	(Gx^2+Gy^2)^0,5		2208,13	2208,126709
locking						
μ		=			0,12	0,120000
		=	D1/D2		0,12278	0,122785
Wmax		=			260,62	260,619086
is er zelf locking		=			0,00	0,000000
herts contact spanning(n/mm)		=	(((D2/"contact lengt")*: Reduced Young's modules")/(2*pi()*"reduced rol radius"))^0,5		1597,67	1597,667834
The elatic foudation model tension						
						0,000000
e3		=	elastic moduli rubber layer (n/m^2)	100000000,00		100000000,000000
h	thichnes rubber	=		0,10		0,000100
a	Halfcontact width		((3*D2*radius spiraal"*h)/e3)^(1/3)		0,6252	0,000625
elastic foundation contact spanning(n/mm)		=	(e3/(2*radius spiraal"*h))*((a^2)+('contact lengte'^2))		338,5443652	338,544365

10.5.2 test results model





used units	
moment	N*m
distance	mm
angle	degrees
force	N

standard valuse	
Arm	70

General information		The test valuse														Calculated results														
											mea	mea	mea	measur	theoretic						copy	measured	measured	measured	measured	measured	measured	measured	measured	measured
			θ3	F3	x3	y3	A	B			C	β	α	contact	contact (				Moment H		D1	D2	Ex	Ey	E	Gx	Gy	G	μ	herts conta
		result:		unster forc												result:	x4	y4	moment verd		krachten p	krachten p	krachten p	krachten p	krachten plaatje bus in krachten in x richting					
		formula:														formula:	"arm H"*sin(θ3)	"arm H"*cos(θ3)	COS(ATAN(		"moment H	D1*((C*SIN	D2*cos(α)-	D1cos(α)-D	(Ex^2+Ey^	-H-D1sin(α)-D2cos(α)		(Gx^2+Gy^	D1/D2	(((D2/"cont
Test number	used pawl	unit:	deg	N	mm	mm	mm	mm			mm	derg	degr	degrees	degrees	unit:	mm	mm	n*m		N	N	N	N	N	N	N	N		
test	test		5	29,82467	100	100	30,3	7,5			23	1,7	5,3		7		6,100901992	69,73362887	2		266,6667	2171,826	2187,138	66,07202	2188,136	-2187,14	-66,072	2188,136	0,122785	1597,67

10.6 Calculation model results

Legend: Light grey are fixed input values and drack grey is change in put values and whit are the calculated values.

A (mm)	B (mm)	C (mm)	Alfa(contact angle) (degrees)	D1(Newton)	D2 (Newton)	D((D1^2 +D2^2)^0,5)	Needed friction coefficient	Moment from h (N*m)	Herts contact tension (n/mm)	Herts contact tension * 5 mm contact line
30,27	7,5	22,77	0,0	266	15279	15281	0,02	2	4237	21185
30,27	7,5	22,78	1,0	266	15279	15281	0,02	2	4237	21185
30,27	7,5	22,79	2,5	266	6107	6113	0,04	2	2679	13395
30,27	7,5	22,81	5,0	266	3048	3060	0,09	2	1892	9460
30,27	7,5	22,86	7,0	266	2171	2187	0,12	2	1597	7985
30,27	7,5	22,92	10,0	266	1512	1535	0,18	2	1333	6665
30,27	7,5	22,96	13,0	266	1155	1185	0,23	2	1165	5825
30,27	7,5	23,11	15,0	266	995	1030	0,27	2	1081	5405
30,27	7,5	23,54	20,0	266	732	779	0,36	2	927	4635
30,27	7,5	24,50	30,0	266	461	532	0,58	2	736	3680
30,27	7,5	24,50	45,0	266	266	376	1,00	2	559	2795

Figure 73 table results changing contact angle

A (mm)	B (mm)	C (mm)	Alfa(contact angle) (degrees)	D1(newton)	D2 (newton)	D((D1^2+ D2^2)^0,5)	Needed friction coefficient	Moment from h (N*m)	Herts contact tension (n/mm)	Herts contact tension * 5 mm contact line
30,27	2,5	27,78	7	800	6515	6564	0,1228	2	4518	22590
30,27	5	25,3	7	400	3257	3281	0,1228	2	2328	11640
30,27	7,5	22,81	7	266	2171	2187	0,1228	2	1597	7985
30,27	10	20,32	7	200	1639	1651	0,1228	2	1231	6155
30,27	12,5	17,83	7	160	1303	1313	0,1228	2	1010	5050
30,27	15	15,33	7	133	1085	1093	0,1228	2	862	4310
30,27	17,5	12,83	7	114	930	937	0,1228	2	756	3780

Figure 74 table results changing B

10.7 Pictures test friction silicon carbide.

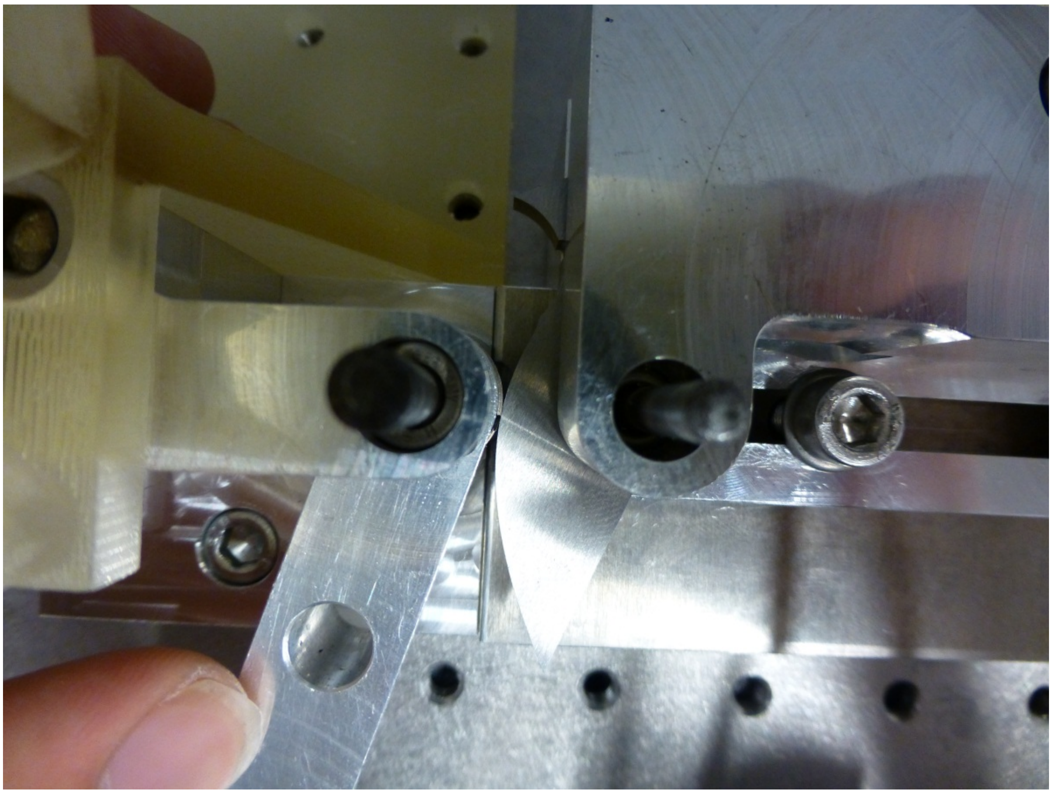


Figure 75 foto11,meeting19

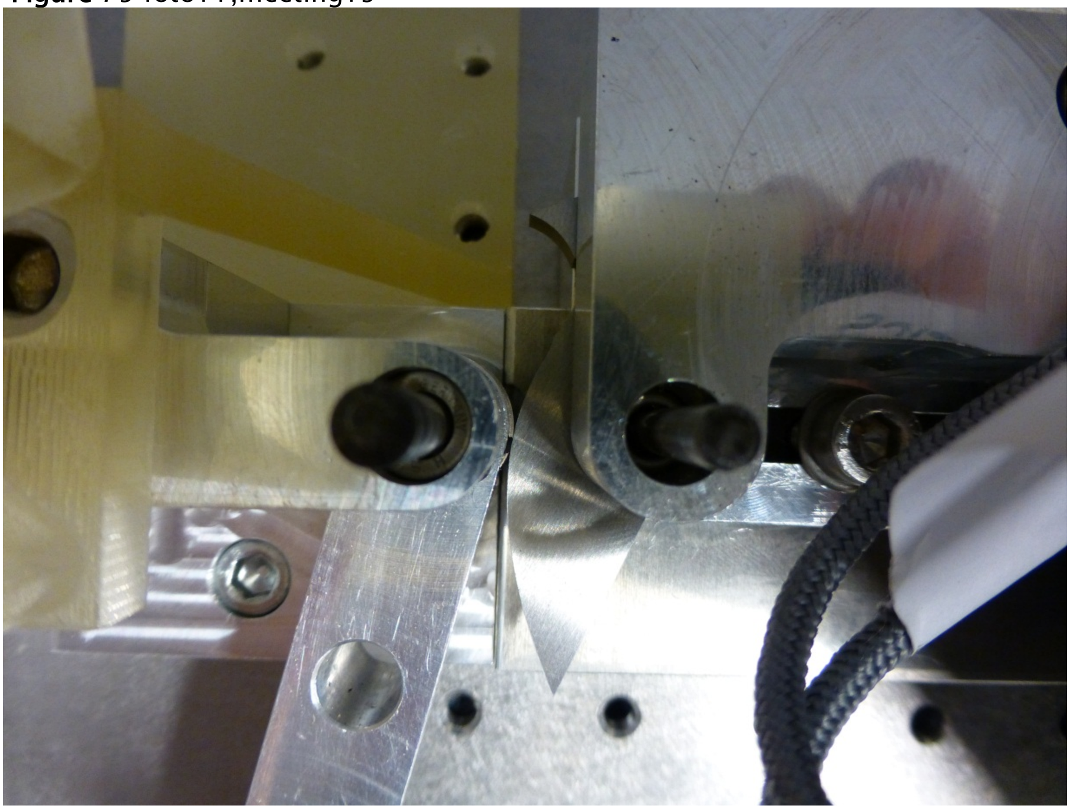


Figure 76 foto10,meeting18

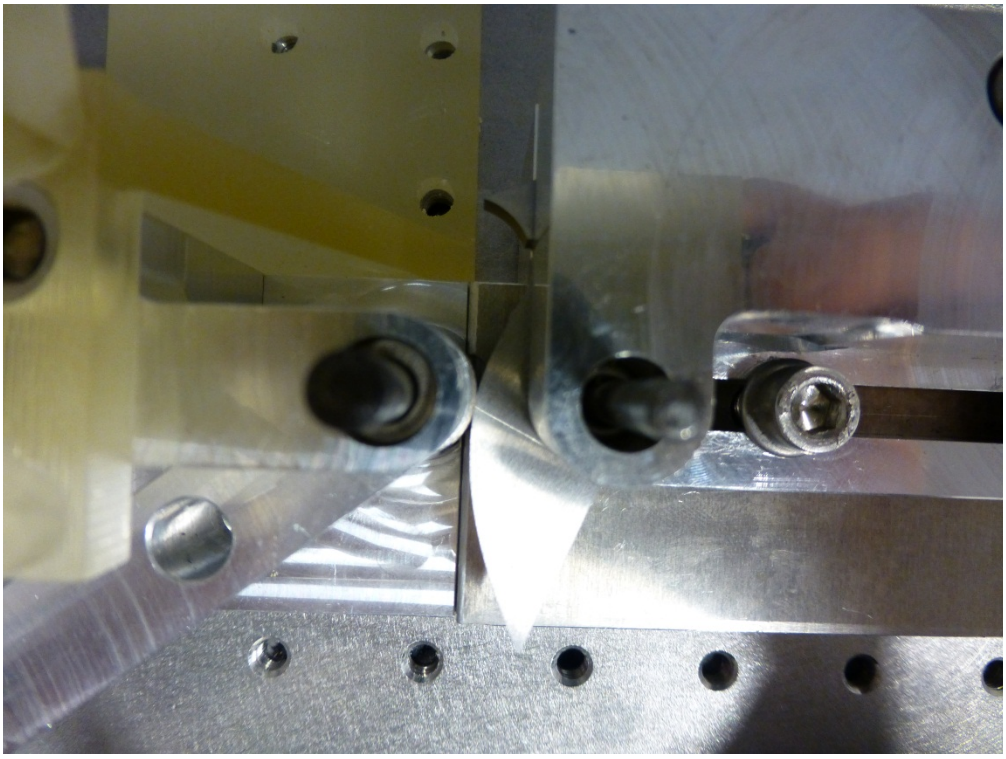


Figure 77 foto9,meeting16

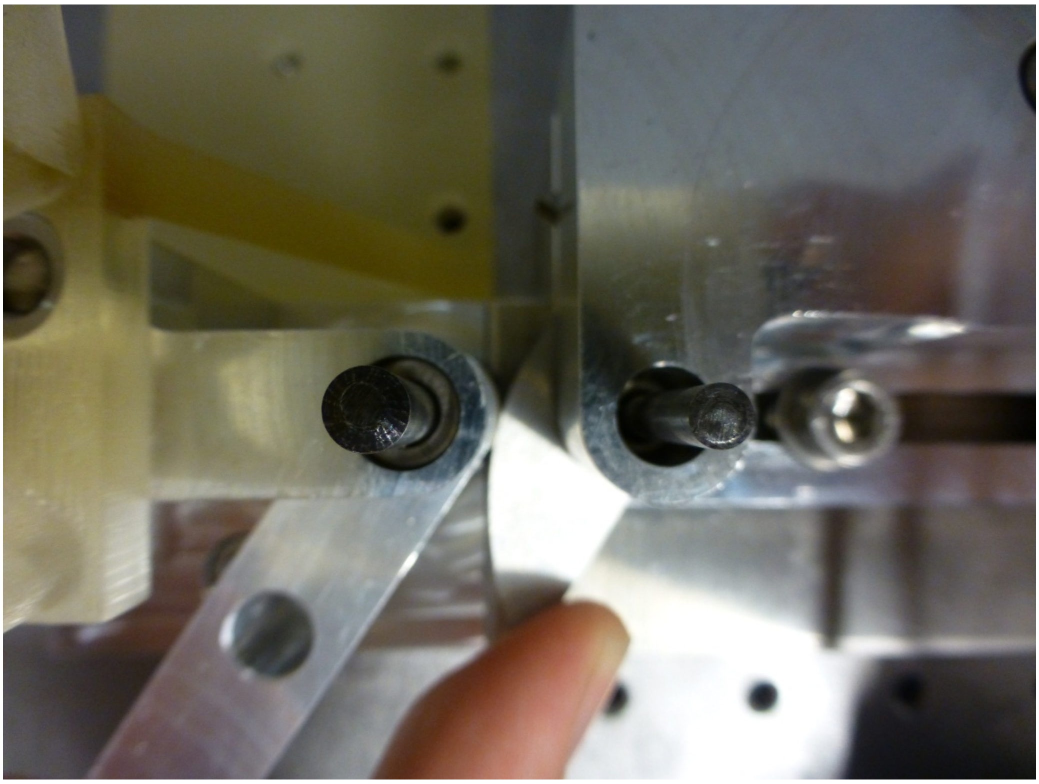


Figure 78 foto8,meeting15

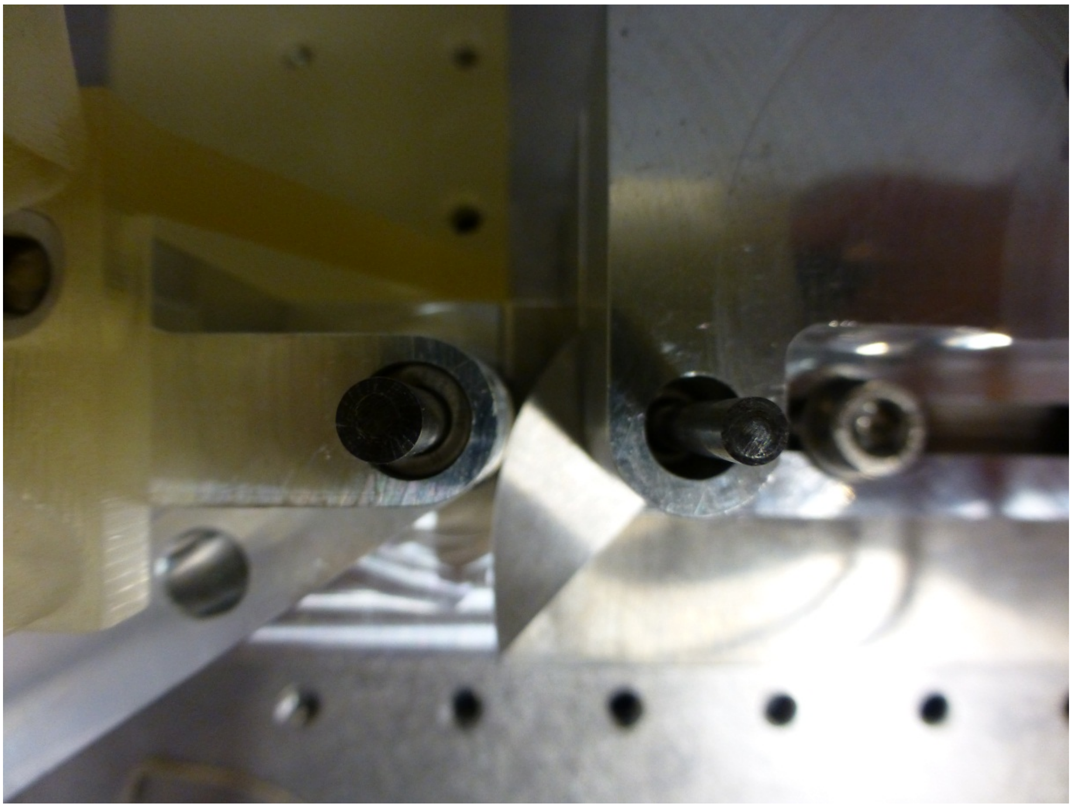


Figure 79 foto7,meeting11

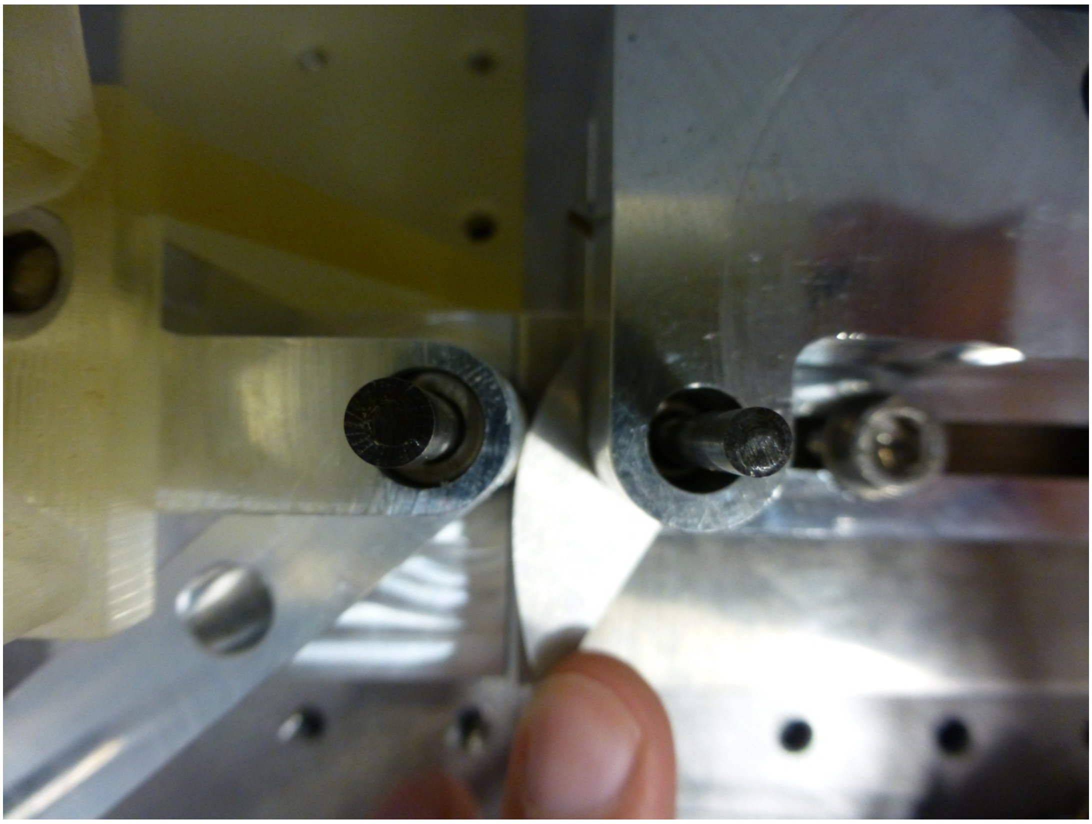


Figure 80 foto4,meeting8

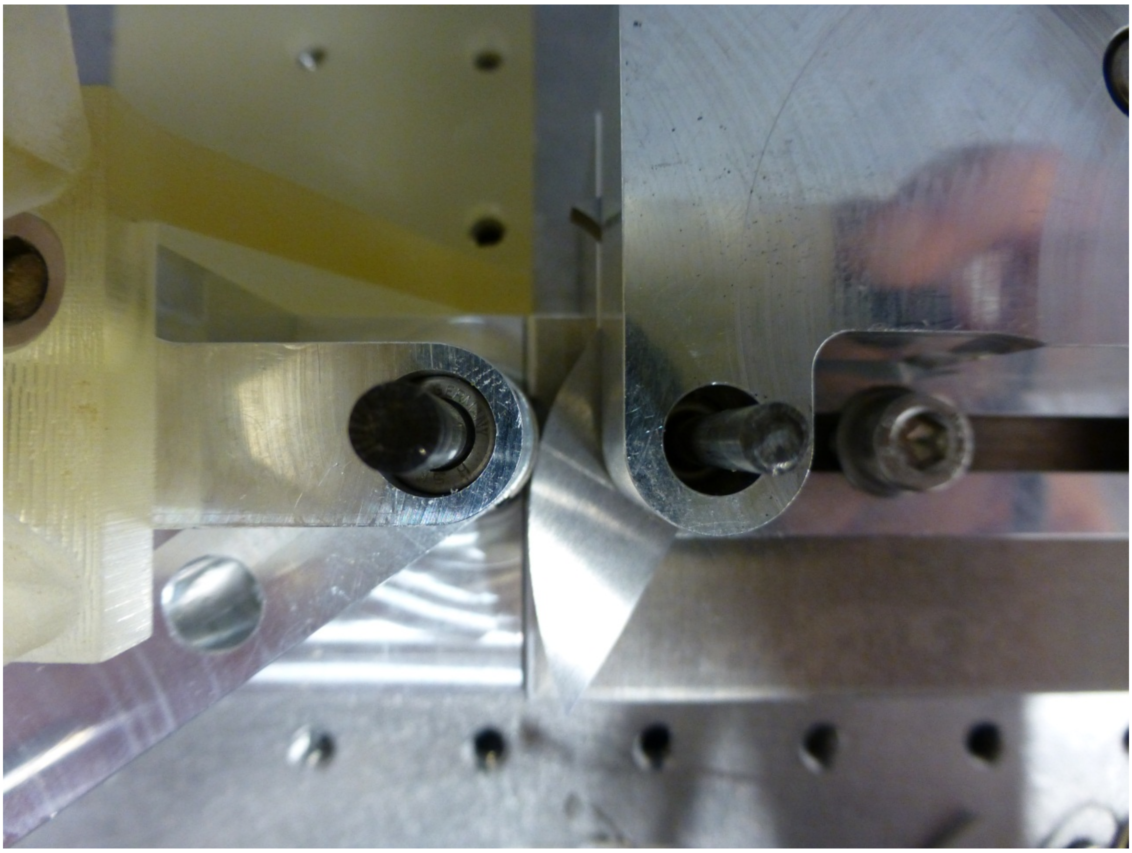


Figure 81 foto3,meeting7

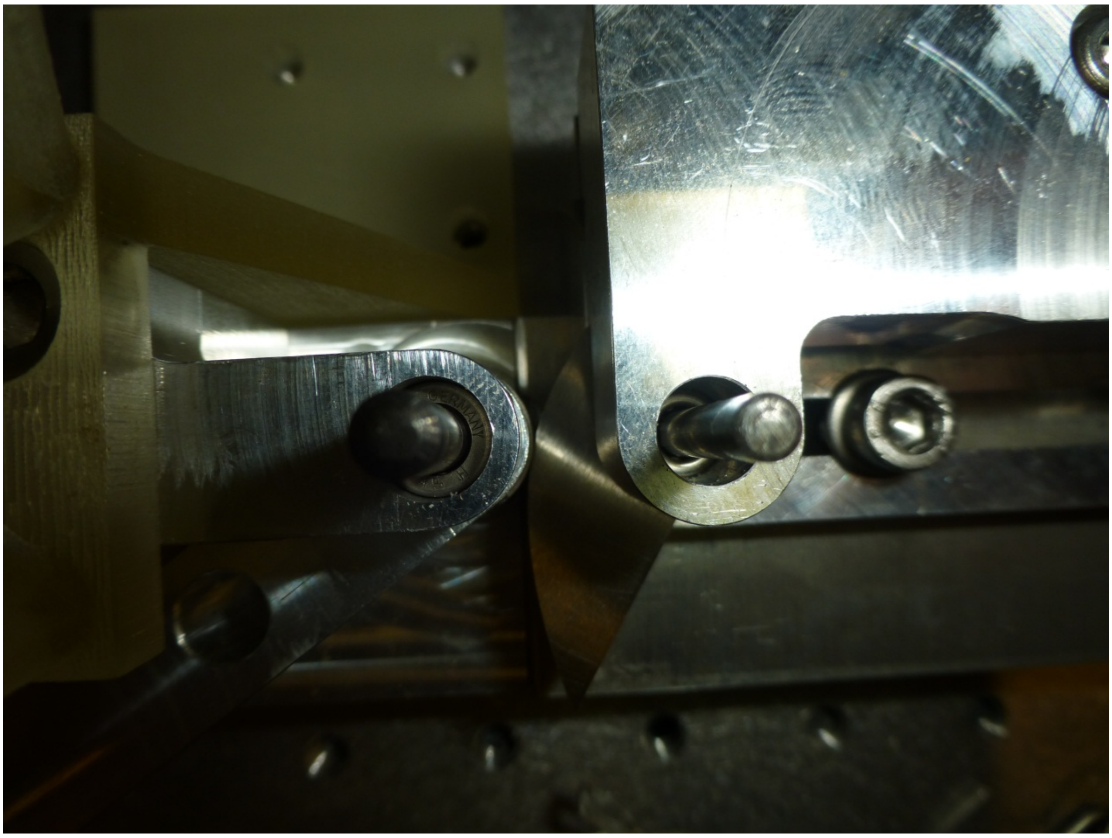


Figure 82 foto2,meeting7

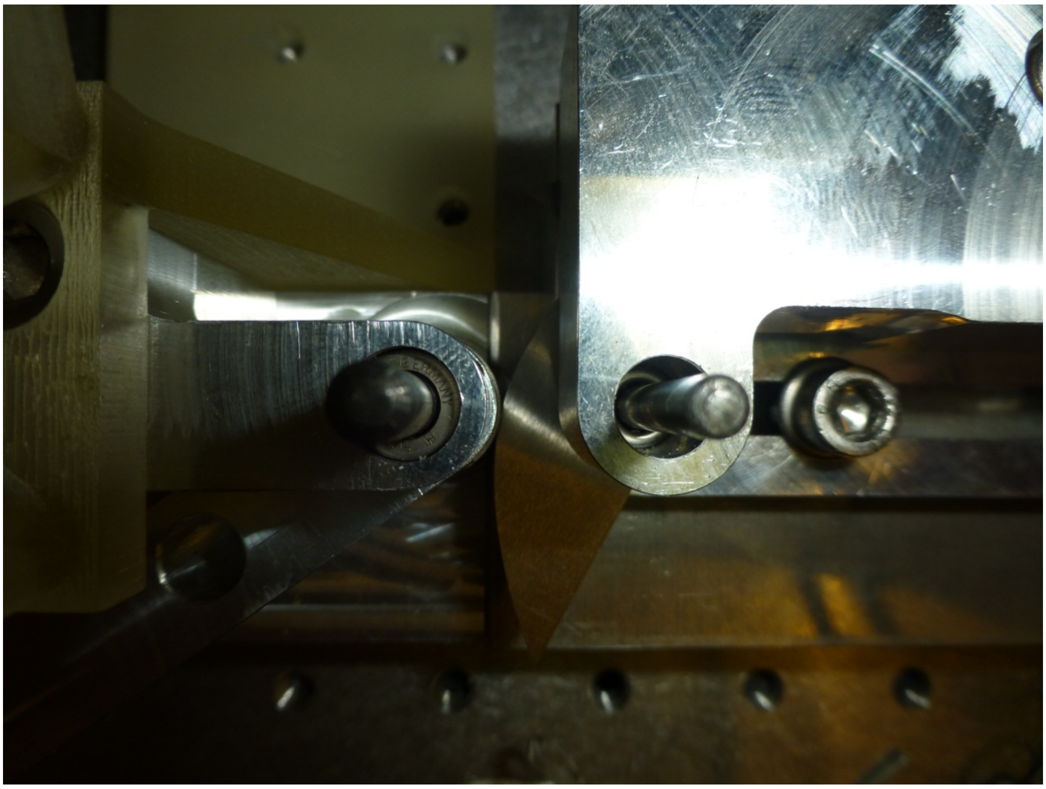


Figure 83 foto1,meeting6

10.8 Piece of art

In process in getting the plastic dip coating on the drum a piece of art is created. Also to the civilized cultivated society is a cultural contribution delivered.

Figure 84 piece of art