Delft University of Technology

Final report

ME46015

Final Report - Group 4B

Blade Mechanism and Balancing

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Contents

1 Conceptual Design Report

1.1 Subassignment Description

At the start of this course, the company ASML came up with a design assignment which was about *designing a machine with two blades that block the light in a scanning motion.* These two blades, or with another words the end effectors, have to move independently in the y-direction. The blades also have to move by any time-displacement profile, maximum acceleration, speed and range. The values are all specified in subsection 1.7.

The design assignment has been distributed over four subsystems: The frame, the drivetrain, the anti-crash system and the blade mechanism. For the subsystem in this paper the focus will be put on the blade mechanism, more specifically the precision mechanism that will house the independent linear movement of the blades while also balancing out the exerted forces. That is why the design description of this subsystem is: **Design a independent linear blade motion mechanism in the y-direction with a balancing mechanism to even out the exerted forces.**

In the final report announcement it was asked to bundle the Conceptual design report, the detailed design report and the critical design report together and merge it into a final document. This has been done together with the implementation of the feedback received from our peers. For this subsystem, the design has changed quite some times over the course of the three reports. To make the document more streamlined, the reports will have some overlapping features and references to future section.

1.2 Problem Statement

The problem statement is to design a precision mechanism that independently moves two blades in the y-directions whilst simultaneously cancelling out the exerted forces and keeping the accuracy high. This mechanism, along with the other subsystems, have to fit in a predetermined design space setup by ASML. A visualisation of this space and the dimensions can be found in Figure 1.

The two features belonging to this subsystem could be considered the most crucial part of the whole system because of the necessary high accuracy and extensive range of motion (ROM) with a single degree of freedom (DoF) in the y-direction. With this responsibility, various difficult design challenges have been determined by ASML and need to be considered for this problem:

- Provide the desired motion for +/- 80mm displacement with an extra 15mm on both sides to create room for a possible crash.
- The mechanism has to be suitable to operate in vacuum. This includes not releasing particles and an as low as possible out-gassing of the system itself.
- If the system crashes, it should not affect the rest of the system and it should be able to start up again soon after without major human interference.

This problem will be tackled with the following steps. First, in section 1, multiple concepts will be made from which a final concept will be chosen. This final concept will be further explained in the detailed design report in section 2, as more time needed to be spend on this concept. And in the final section 3, the model will be estimated on performance and finalized.

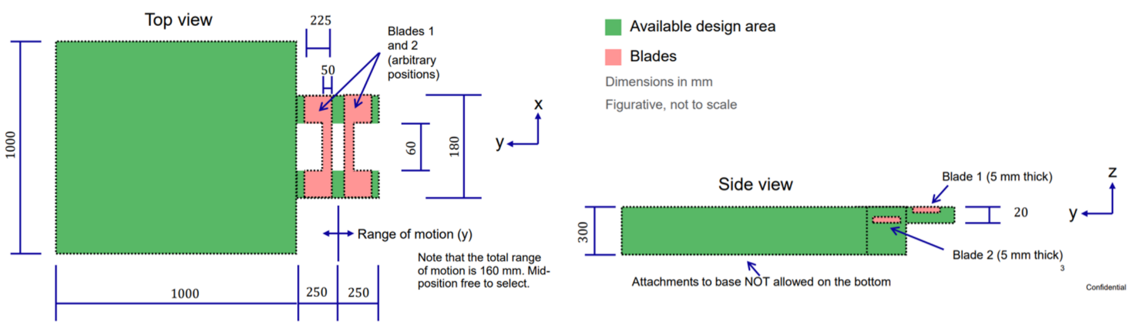


Figure 1: Schematic visualisation of designated space for the design assignment

1.3 Functional Analysis

The most important step before coming up with possible design solutions and a final concept, is setting up the functional requirements. This is of essence because this determines the path that can be taken towards coming up with possible design strategies, different concept solutions, design strategies and eventually certain concepts.

Function requirements are specifications that describe the essential features, capabilities, and behaviors a system must possess to fulfill its intended purpose and meet user needs. For this subsystem, three functional requirements have been identified and are as follows:

- Blades move independently in y-direction in a linear motion
- Mechanism accounts for exerted forces from the blade activation
- Blades connect to mechanism

The first functional requirement emphasizes on the fact that the blades have to move, independently, in a linear fashion along the y-axis. This has to do with the fact that these blades have to block out the laser very accurately in different places of the designated area. The drivetrain to accomplish this movement will be designed by another subgroup. That leaves the linear guide for this system to be designed. The magnitude and specifications of the necessary displacement are mentioned in subsection 1.7.

The second requirement focuses on the design specification that the blades should accelerate with $400 \mathrm{m/s}^2$, which is equal to a 40g load. This force cannot be exerted on the rest of the machine to avoid damaging components due to vibrations. That is why a balancing mechanism has to be integrated into this subsystem. Multiple options to solve this problem have been given in Figure 2. This is the most crucial functional requirement as in the other subsystems it will be very difficult to balance out these and other forces.

And the third functional requirement is mentioned to account for the loss in accuracy created by incorrect blade attachment. The blade are already specified and weight 650 grams each. To avoid shifts in the blade attachment under the $400 \mathrm{m/s}^2$ movement, the connection has to be appropriate. This functional requirement has been because of the accuracy constraint, however each concept that will be found later in this paper will have sufficient strength. That is why not a lot of emphasis has been put on this requirement in this final paper.

1.4 State of the Art

Gathering possible design solutions could be difficult without a proper research in the literature. This section has been dedicated to review the mechanisms and principles that are already in place in similar mechanisms.

For the balancing mechanism, one of the first principles that will be thought of while talking about this topic is weight balancing. This principle has greatly been described in Weight Balancing of Precision Mechanical Instruments by H. Hilpert [1]. The general description is given by Newton's Second Law of Motion of which the sum in respective to the base has to be zero. In the paper by Hilpert, multiple ways of mass balancing has been described. The most simple way is with the use of the counter mass on a seesaw or for example a pantograph that has a certain transmission ratio.

Other ways of dynamics balancing would be for example a flywheel, as described in Experimental Results for Stabilizing of a Bicycle with a Flywheel Balancer [2]. In this paper, a flywheel has been used to balance out a bicycle and its corresponding loads. Because of the specifications and the environment of the mechanism being a vacuum, this is probably not the best way to balance the mechanism because of the particles released in the bearings of such flywheels.

The mechanism that will be designed is the transmission from the actuator to the blade, which has to move in a straight line without any backlash or shift in the x-direction. State of the art for linear motion nowadays are linear guide ball bearings, which are mostly used in CNC machines or measurement tools. The benefit of these bearings are that they are very rigid and accurate, but the disadvantage is that they rely on the low friction between the bearing balls and bearing surface. Even though these bearings are designed for low friction, these products still produce particles which make them impractical to use in this system.

A second option would be a linear magnetic rail [3] or the use of air/fluid bearings. These bearings do not make contact with their surroundings, but are therefore not very stiff. Plus the use of these products are not ideal in a vacuum environment.

The third and probably best option for this functional requirement is the use of compliant mechanisms. Compliant mechanisms gain their motion from deflection of elastic members [4], which is very useful in this application because of the absence of friction and therefore particles. These kind of mechanisms can be used in very complex shapes and could there be used to create linear motions.

The connection between parts in precision mechanisms are very broad, but have to be rigid to prevent the mechanism from creating particles. Therefor welding can be one solution, although it might create parts that have out gassing. Connection via bolts and nuts is also possible but the clamping force of the bolts should be bigger than the inertia forces of the blade otherwise it will shift inside the hole and create friction and thus particles. It is difficult to find state of the art literature about this functional requirement because of the simple nature of the problem and will not be further elaborated.

1.5 Design Strategies

Several design concepts were developed for the comprehensive blade mechanism by breaking it down into three primary components: the balancing mechanism, the movement guide and the blade attachment. A morphological table aided in subdividing the mechanism and generating multiple ideas for each part based on existing literature. These ideas are described in subsection 1.6.

While coming up with these ideas, all possibilities of mechanisms that could complete the functional requirement are taken into consideration. Although the table has been kept concrete and to the point. For example for the movement guides, particular attention was given to enabling the independent blade motion in a linear fashion. Similarly, appropriate balancing mechanisms were carefully selected to ensure the mechanism could effectively handle exerted forces.

After this, the design specifications and requirements will be mentioned. This will help with the selection of three concepts out of the morphological chart and combine it to whole system concepts. These concepts will thoroughly be explained and evaluated after which a final concept will be chosen. In section 2, this concept will be explained in detail. The objective of this design strategy is to create a complete concept that fulfills all the prescribed functional requirements outlined in subsection 1.3.

1.6 Morphological Chart

Using this strategy and the functional requirement that have been selected, a morphological chart has been made. This chart can be found in Figure 2.

For the balancing mechanism the following possibilities have been selected: Dual actuators for independent blades and the counter mass, actuation both masses with a single actuator, using a parallelogram and an actuator, using a seesaw mechanism, a conveyor belt and a flywheel.

For the linear guides, the following mechanisms have been selected: A magnetic linear guide, a linear bearing, using leaf springs/compliant mechanism to get a one degree of freedom system, using cross flexures, hydrostatic bearings and air bearings.

And for the blade connection the following can be used: Bolted connection, glued connection, welding the components together, using a tapered fit and making the arms and blades out of the same component.

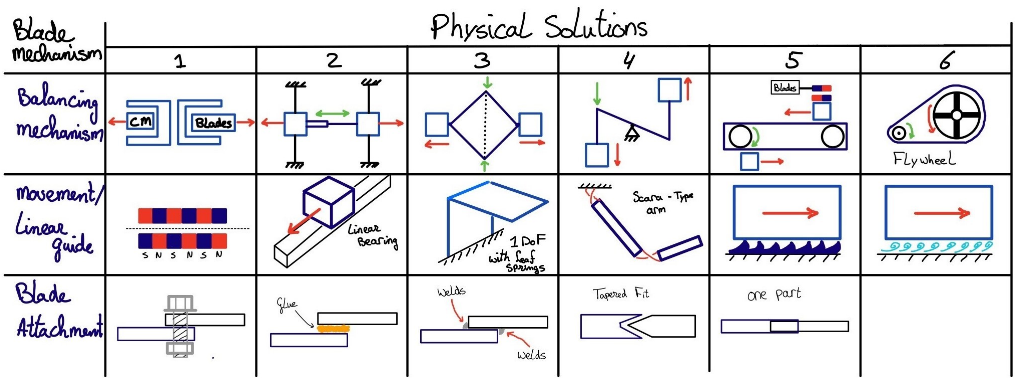


Figure 2: Morphological Chart

1.7 Design Specifications

The client, ASML, has setup up specification for which the whole system should account for. The general description of these specifications have been mentioned below:

- The blade mechanism must not exert forces greater than 400N.
- Must have two separate controllable blades with 1 DoF.
- Both blades must be able to move by any time-displacement profile.
  - The specified acceleration is $400 \frac{m}{s^2}$.
  - The mechanism must allow blade movement of +/-80mm (+ crash distance)
- The mechanism must operate in vacuum environment
- The subsystem must not produce more heat than budgeted
- The subsystem must not outgas more than budgeted
- When displaced and returned to the original position, the deviation from the original position should be smaller than budgeted for.

These requirements have been divided and distributed over the subsystems. This means that certain subsystems have more play when it come to, for example, outgassing or heat production. The specific budget per subgroup can be seen in Table 1. This means that the blade mechanism itself can take up a maximum of 30% of the total space and should exert less than 45% of the exported forces budgeted on 400N. This table has been made during a brainstorming session in the beginning of this project and is expected to change over time.

	Frame	Drive-train	**Blade mechanism**	Anti-crash
Space	25%	30%	30%	15%
Thermal	4%	80%	8%	8%
Gas	30%	30%	30%	10%
Forces	0%	50%	45%	5%
Deviation	10%	70%	10%	10%
Start up time	0%	0%	0%	100%

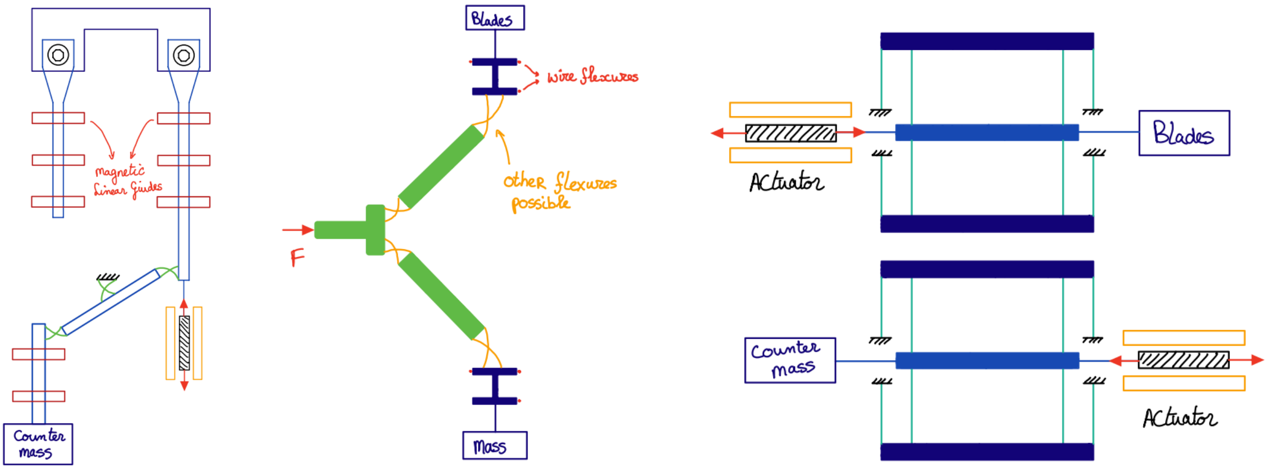
Table 1: Budgets

1.8 Concept Generation

These design specifications have been used to evaluate the different ideas that have been generated in the previous chapters. By doing this, various ideas would already drop from the considerations. For example, because of the system having to operate in vacuum the linear bearing guide would already not work. This also counts for the air/hydrostatic bearings as they do not operate ideally in these conditions and are generally not stiff enough in the desired directions. This leaves the magnetic linear guide and both compliant options the best solution for this functional requirement.

For the blade connection, multiple ideas have been considered that are all viable except for the glue solution. With the blades becoming hot, it is possible that the glue will trap gas within a solid which then can be released. So this is not an option. The others are all solid solutions and will be mentioned in the concept, but further elaboration will not be given on this.

For the balancing mechanism, the flywheel and the conveyor belt are not viable as these will release particles when actuating. The other concepts all have interesting solutions that will be used in the three concepts that will be evaluated. These are visualised and described below:



(a) First concept (b) Second concept (c) Third concept

Figure 3: Concepts representation

1.8.1 First concept

The first concept, which can be seen in Figure 3a consists of the following ideas:

- A see-saw mechanism, magnetic guidance of the blades and countersunk bolts for the blade connection

This system uses compliant mechanisms as joint to make sure the seesaw motion is created without particle release. The seesaw makes sure the blades and the counter mass move in opposite directions. It uses magnetic linear guides as a way of housing the linear y-direction motion. The actuator will be placed on one of the ends of the seesaw.

1.8.2 Second Concept

The second solution has the following solutions and can be seen in Figure 3b:

- A Parallelogram/one actuator to move the blades and counter mass, a compliant Scara type arm providing linear motion and a connection between the blades and the mechanism that is welded

For this concept, options 2 and 3 of the balancing mechanism has been taken into account. The idea of this subgroup is to use the actuator base, which has quite a significant mass, as the counter mass for this system has it easily balances out the exerted forces with the correct ratio. That is visualised in option 2. However the mechanism to accompany this ratio has yet to be figured out, so for now the parallelogram is used to visualise this idea. Here the input at the bottom of the picture causes the blade and the counter mass to move in opposite direction with the same value. The linear motion of the blades is forced by the use of wire flexures in a system that makes a 1 Degree of Freedom (DoF) system.

1.8.3 Third Concepts

The third and final concept, Figure 3c, will consist of the following solutions:

- Two independent actuators to actuate the blade and the counter mass, compound leaf springs to allow one DoF movement and the blades are integrated in the arms

This concept uses two actuators for the movement of both masses, for the completed system there will be three actuators: two for the blades and one for the counter mass, which will be programmed to counter out the exerted forces. This system is relatively easy and only needs a linear guide for the mover of the actuator. This linear guide consists of compliant mechanisms in the form of compound leaf springs. These leaf springs are widely discussed in the Compliant Mechanisms Course.

1.9 Concepts Evaluation

The first concept presents a magnetic levitation linear guide and a see-saw mechanism. Despite the benefits of zero friction and thus wear on the parts, which are perfect for this application, this type of linear guide has several drawbacks. Indeed, in order to produce magnetic levitation a complex system of actuators and controllers has to be used. Thus the system would become very expensive compare d to other technologies. Another drawback is the very low stiffness in direction perpendicular to the desired motion direction, which could lead to error in the positioning of the blade. The see-saw mechanism seems a valid option to meet the motion requirements. However its design is less compact than other options.

The second concept suffers of similar drawbacks relative to the space occupation in the system. The motion of the second concept mechanism is based on compliant joints and wire flexures. This, if well designed, guarantee an higher stiffness in direction perpendicular to the desired motion, thus reducing parasitic motion. It would be great to use the actuator base as counter mass to simplify this concept, this really intrigues this subgroup. Due to time constraints, the use of a pantograph will be considered in the next paper to get a ratio between the mover and the actuator. The pantograph guarantees a linear motion which is ideal for this situation as this would put the two functional requirements into one system.

The third concept differently from the other two, present two independent counter masses for each one of the blade, and thus more freedom and balancing capabilities. The motion is based on compliant mechanisms, in particular it uses compound leaf flexures. This type of flexure systems results very simple in design, but it enables great performances. Indeed, it compensate the parasitic movement of the end affector and guarantee high stiffness in direction perpendicular to the movement direction. This flexure systems is over-constrained, however the over-constraints can be easily removed by modifying the leaf flexures in the system.

1.10 Concept Selection

To select a concept in a general case, a weighted table would be an ideal solution to choose the best one. However with two crucial functional requirements, this is deemed to be a little ineffective. That is why the decision has been made by looking at the evaluation of the two concepts.

For the first concept, it has been decided that the amount of room and the seesaw mechanism is not ideal. The seesaw itself has no linear motion and that makes the use of leaf springs or other compliant mechanism difficult, as it would require an extra beam between the joints. This would make the system difficult to estimate the movement.

The second and third concepts both do not have a lot of negative setbacks. The third concept is a lot easier than the second concept because it is better to predict the existing movement. However this concept would be very heavy and cost effective because of the need of three or four actuators and more counter masses. Together with this, the intrigue towards the usage of the actuator base as counter mass is very high. The second concept would be more challenging and more interesting than a mechanism with two actuators for a single blade, which is why the decision has been made to focus on the second concept for the next section.

The *Detailed Design Report* will start of by explaining the finalization of the chosen concept, with mainly the mechanism that needs to be specified more. After this the working principle and perfomance will be evaluated.

2 Detailed Design Report

2.1 Detailed Design Presentation

In this part the new concept and design of the mechanisms will be discussed in detail. To visualize the chosen concept, a 3D model was created in SolidWorks. Pictures of the working principle and the CAD model can be found in subsubsection 2.1.2 and subsubsection 2.1.3 respectively. The dimensions, PRBM and kinematics of this system can be found in subsection 2.4.

2.1.1 Finalized Concept

As explained in subsection 1.10, the selected concept had not been fully worked out. This is because the idea of this subgroup was to design a mechanism that is not as straight forward as using 2 Lorentz actuators, one for the blade and one for the counter mass, but rather to look at a more original idea. Here the idea arose of using the actuator base as counter mass of the blades itself. The idea was to work out this design in the time between the deadline of the conceptual design report and the start of the detailed report. The mechanism should be capable of moving the mover and base of the actuator in opposite direction at a certain ratio that it would cancel out the exerted forces.

While looking back in the papers of H. Hilpert [1] and D.C.H Yang and Y.Y Lin [5], the final decision has been made that a pantograph will be the mechanism that facilitates the ratio between the mover and the base. Exactly why will be explained in the next section.

2.1.2 Working principle

A pantograph is a mechanical linkage that establishes a connection by utilizing the principles of parallelograms. The mechanism is widely in use to copy and scale diagrams or sculptures, because of an intriguing property of the pantograph. The ratio between two of the end-points of the pantograph is directly determined by the length of its members P1-P2 and P2-P3 in Figure 4 and is therefore fairly easy to implement.

These are also the reasons why it would be great to place this mechanism in this subsystem. The base of the actuator is a factor of 2.5 heavier than the actuator mover and blade, which means the lengths of the arms of the pantograph need to have a ratio of 2.5. The ratio is better explained in subsection 2.2. The kinematics of the mechanisms is explained in subsection 2.4. A scheme of the mechanisms and how is connected to the system is shown in Figure 4.

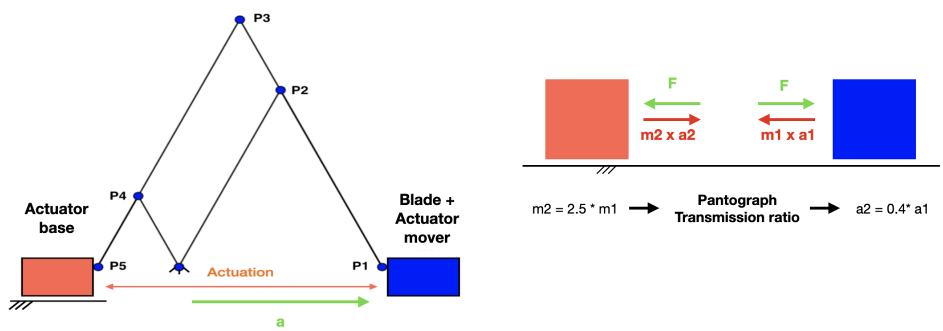


Figure 4: A schematic view of the mechanism and balancing principle

For this mechanism, the middle point of the pantograph is fixed and both end will move. The ratio between the two does not change. This has been done to make sure that the actuator base and mover move in opposite directions. In this way, the ratio will case for the mechanism to balance itself out.

But if the middle point is a joint and rigidly connected in space and the other 2 points are also considered joints fixed to the actuator, the mechanism has 4 Degrees of Freedom (DoF): Translation in the y direction of the end points, rotation around the base joint and both rotations of the actuator around the end points. But this system has to be a 1 DoF system. This has been solved as follows:

- The base will be linearly guided. This will be done by the drivetrain subgroup and will not be explained here. This makes sure that the left endpoint cannot freely rotate anymore.
- The same linear guide also counteracts the rotation around the base joint of the mechanism because it is physically restrained.
- And lastly this also contraints the movement in the y-direction as the base cannot move to the top and bottom anymore, but only to the left and right (Figure 6a)

This leaves 2 DoF: The wanted linear motion of the blades and the rotation around the joint on the blade side of the mechanism. To counteract this, and to make the system more stable and rigid, a second pantograph has been added. This pantograph has the same rotation as the already existing one, but in the other direction. In this way the pantographs cancel out the rotation of the mover and it will move purely linear with 1 DoF.

The last mentioned modification was implemented only in the *Critical Design Report* as can be observed from the CAD models in Figure 11a. The relative explanation for the DoF has been inserted here for better clarity of the mechanisms functioning principle. This was the final step from the **Detailed Design** tot the **Critical Design**. In the next section, it can be seen that two actuators both have only one pantograph each. The reason for this is because during the research for the second report, the guidance of the mover was not taking into account.

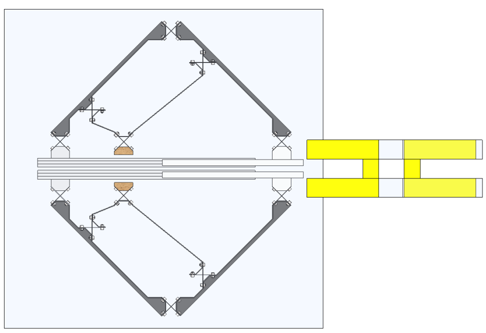
2.1.3 Single Pantograph - CAD Model

A model has been made of the new concept to explain it in more detail. The middle part of the pantograph, connected to the brown base, will be a rigid pillar connected to the frame directly. The grey base on the short part of the mechanism will be attached to the actuator base and on the other side the mover will be attached. In Figure 6 the two actuators can be seen, but as have been said this will change in the *Critical Design Report*.

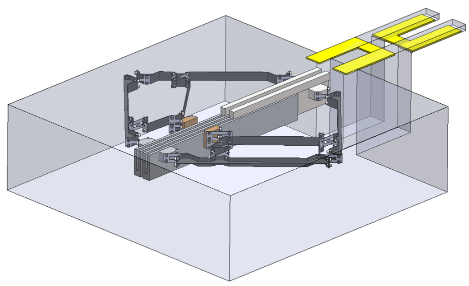
(a) Mechanism with actuator on the right

(b) Mechanism without actuator

Figure 5: Different views of the mechanism



(a) Top view inside design space



(b) ISO view of mechanism in design space

Figure 6: Both actuators with 2 mechanisms in the design space

2.1.4 Part List

This part list has been developed for one pantograph and blade mechanism only. This bill of material (BoM) is a stepping stone towards the final part list given in subsubsection 3.1.4. The connection from the pantograph to the actuator has not been designed yet and will therefore not exist in this BoM.

Bill of Materials		
Parts	Quantity	Description
Set of Cross Flexures	6	Consists of 2 leaf flexure that form the cross
Mover Connection	1	Connects the pantograph to the mover
Actuator Base Connection	1	Connects the pantograph to the base
Top Pantograph Arm	2	Identical arms on the top
Small Arm	1	Smallest arm connected to the ground
Big Arm	1	Biggest arm connected to the ground
M6x10 Bolts	48	Bolts to connect the parts
M6 Nuts	32	Nuts to connect the bolts

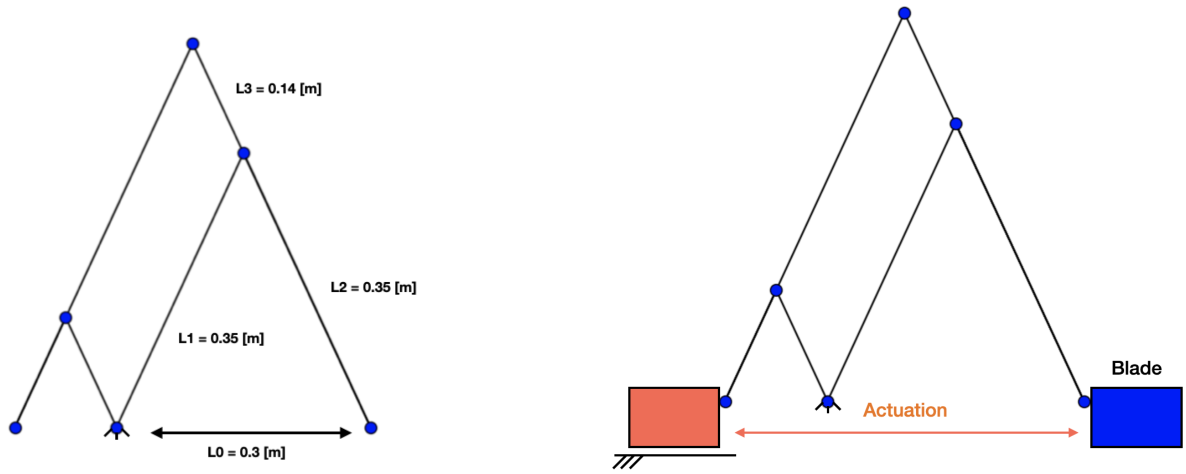
Table 2: Bill of Materials for one Pantograph

2.2 Presentation of Main Dimensions

After some design iterations with the kinematic model explained in subsection 2.4, a good combination of values was found:

$$L1 = 0.35m \quad L2 = 0.35m \quad L3 = 0.14m \tag{1}$$

The resulting transmission ratio is given by $\frac{L3}{L1} = 0.4$. Thus for a movement of the blade of 1 the counter-mass would move of 0.4



(a) Dimensions of the mechanism

(b) Scheme of the mechanism

Due to design space limitations and feedback received, during the last part of the design process the dimensions were modified to:

$$L0 = 0.3m \quad L1 = 0.3m \quad L2 = 0.3m \quad L3 = 0.12m \tag{2}$$

These dimensions give the right ratio, but what is also important is the angle on which the pantograph is in rest. Multiple simulations have been done on the angular displacement of all the members on multiple angles, for example 45° and 60°. It turned out that at a higher angle, the angular displacement where less resulting in a lower actuation force and the compliant joints being closer to an actual revolute joint. That is why L0 has been taken as 300 or in other words a 60° angle as a reset position. In this way the actuator also fits between the end points which is necessary for the

Speaking about the compliant cross flexure joints, the dimensions of these has been set to a length of 60mm and a width of 10mm. This results in a stiffness that is calculated in subsubsection 2.4.2. With these dimensions, the force needed to actuate the mechanism is very low which is desired. The out-of-plane stiffness, which is evaluated in subsubsection 3.2.2, will be slightly higher because of this. But the low stiffness is needed to compensate for the weight of all the arms. If this would not be the case, the pantograph will go higher than the budgeted weight for it and the drivetrain subgroup would have to think of a new solution for an actuator.

2.3 Identification of Bottlenecks

A list of the subsystem's components has been compiled to highlight anything that could go wrong or have a negative impact the mechanism behavior. These limitations are divided into groups based on certain functions or factors.

2.3.1 Limitations due to the design space

- The size of the actuator has a significant impact on the concept. Despite having a huge design area, the size of the actuator is still constrained by the actuation force that it must deliver.
- The mechanism needs to be constrained in the design space. Due to this restriction on the mechanism's size, the mechanism's functionality may be impacted.

2.3.2 Limitations due to the design constraints

- One of the system's primary limitations is the low value of out-of-plane stiffness. Considering that an insufficient out-of-plane stiffness will cause the object to move in the undesired z-direction under its own weight. Simulations for parasitic movement will be conducted to see what the effect is.
- While the double pantograph should theoretically output a linear translation, it is heavily dependent on the linear guidance system of the actuator base which will be designed by the drivetrain subgroup.
- Cross flexure joints can decrease the load-bearing capacity of the pantograph mechanism. The deflection and bending under load can exceed the allowable limits, leading to structural failure or compromised performance [6].
- It's possible that the parallel pantograph mechanism won't have enough flexibility or will be too stiff in the y direction.
- The cross flexures' non-linear stiffness characteristics may require that they be given a little bit more stiffness than is necessary. The stiffness can vary throughout the mechanism's range of motion as a result of this non-linear behavior. Due to this, the pantograph might display non-linear motion in various positions, which would affect its accuracy and repeatability.

2.3.3 Limitations due to the constraints from other subsystems

- Due to the limitation of the mass of the complete system including the actuator, the mechanism and the blades, the maximum weight of the guidance mechanism should not exceed 1.87kg.
- The entire mechanism should have a range of motion of $+/-95$mm (including +/- 15mm for crash), which is distributed between the movement of the actuator's base and the blade itself. The transmission ratio will determine this. The actuator's operating range has a significant impact on the crash mechanism's range of motion.

2.4 Analyses, Studies and Calculations

In this section two different performed analysis are described: and *The Kinematic Analysis* and the *Pseudo Rigid Body Model Analysis*. The code relative to this part can be found at: Code repository

2.4.1 Kinematic Analysis

The *Kinematic Analysis* has the aim of characterizing how the mechanism move and what are the relations between the coordinates which describe the mechanism. The chosen mechanism consist of a pantograph, which imposes a transmission ratio between the movement of the blade and the movement of the counter-mass (body of the motor). The transmission ratio is determined from the length of the mechanism's links.

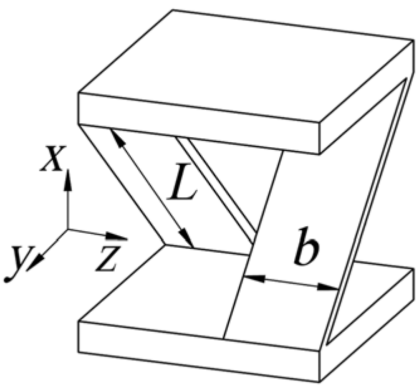
Due to the strict limitation on friction between mechanism's components, all the joints in the mechanism would be made out of rotational flexure components. The angular range of motion of these joints is often limited, thus the goal of the kinematic analysis is to find the best combination of link dimensions and initial position in order to reduce the angular displacement while meeting the desired functional requirements.

The chosen pantograph configuration can be observed in Figure 7b. It consist of four links and seven rotational joints. The central base joints are fixed to the frame, while the other two free ends are connected to the blade and to the counter-mass respectively. Without any other constraint, this mechanism has two Degrees of Freedom. To limit the movement to only a linear translation of the blade, and remove the rotational DoF, one more compliant linear guide has to be added to the motor base, which correspond to the counter-mass.

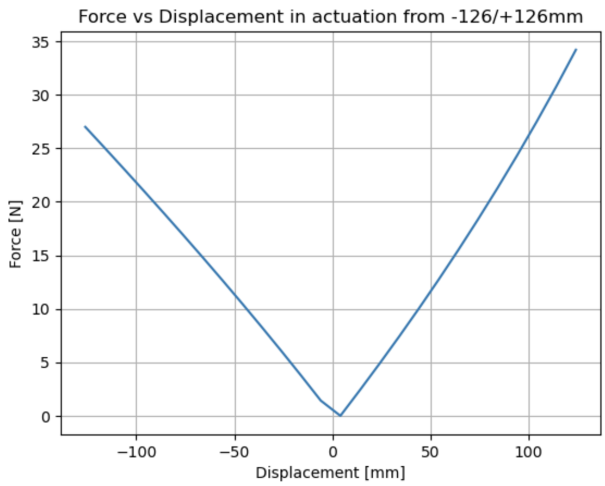
A kinematic model has been realized in Python with the use of the *Sympy* and *Sympy.mechanics* libraries. The input of the model are the geometric dimensions of the links: L1, L2, L3 and the desired linear range of motion (7a). While the output consist of an array containing the angular displacement of all the joints needed to achieve that linear motion.

2.4.2 PRBM - Analysis

To understand the actuation force needed to achieve the desired linear displacement, a Pseudo Rigid Body Model (PRBM) of the mechanism was implemented with python. In correspondence of each rotational joint a rotational spring was placed. The stiffness of the spring would depend on the type of flexure, its dimensions, and its material. Combining the kinematic model, with the stiffnesses values, and applying the virtual work principle, a force versus displacement curve was estimated.



(a) image of a cross flexure



(b) Force vs Displacement curve

The necessary steps to perform the PRBM study are:

1. Generate a PRB model of the given compliant mechanism. Incorporate spring constant, the effective lengths and the angles
2. Perform kinematic analysis and find displacement relationships between the angles, and the first order kinematic coefficient (velocity analysis).
3. Write Lagrange coordinates for each spring (absolute angular displacement of each spring).

4. Drive virtual displacement of the Lagrange coordinates.
5. Apply virtual work principle
6. Replace the Lagrange coordinates variation in step 4 into virtual work formula in step 5
7. Replace the first order kinematic coefficient from step 2 into step 6.

The PRBM, combined with later FEM simulations was a useful tool to dimension the cross flexures in order to guarantee low driving stiffness, and high transversal stiffness. The code with more details and explanation can be found at the repository: Code repository. In the repository the mentioned code is contained in a jupyter notebook called *PRBM - model.ipynb*.

2.5 Performance Parameters, Budget and Interfaces

The interfaces to the blade mechanism are now more clear thanks to the estimated actuator dimensions. The base of the actuator, the mover, and the fixed frame serve as the three points of attachment for the pantograph-like blade mechanism. As a result, there are two interfaces with the drive-train subsystem actuator and one interface with the frame subsystem. The actuator itself turns out to be quite large, as indicated in Figure 9. Consequently, the blade mechanism needs to accommodate this size and becomes a relatively large mechanism itself.

Figure 9: Actuator dimensions

The drivetrain team's provided actuator has a $1200N$ force capacity. This imposes a maximum allowable mass of $\frac{1200}{400} = 3kg$ on the mover side when taking into account the desired acceleration of $400m/s^2$. We have a maximum mass budget of $3 - 1.13 - 0.65 = 1.22kg$ for the blade mechanism after deducting the masses of the blade $(0.65kg)$ and the mover $(1.13kg)$. The mechanism's actuation stiffness must also be taken into account. The design of the mechanism is severely constrained as a result.

These design changes require an update in the budgets. The revised budgets for the blade mechanism group are presented in Table 3 and call for further discussion with the system engineers.

Blade Mechanism	Old	New	Explanation
Space	30%	40%	The design of the pantograph takes up more space
Thermal	8%	0%	Thermal output is not expected to be zero with the current mechanism
Gas	30%	0%	This mechanism will not outgas so it can be shifted to other subsystems
Forces	150N	210N	Mechanism will account for more exported forces
Deviation	10%	30%	With the pantograph, more parasitic movement is introduced
Start up time	0%	0%	As the pantograph is a mechanical linkage, it will not need start up time

Table 3: New desired budget of this subsystem

3 Critical Design Report

3.1 Final Design Documentation

Due to the explanation given in subsubsection 2.1.2 about the mover having an extra DoF due to it not being constrained by the actuator base, the design had to get a final change. These were the options:

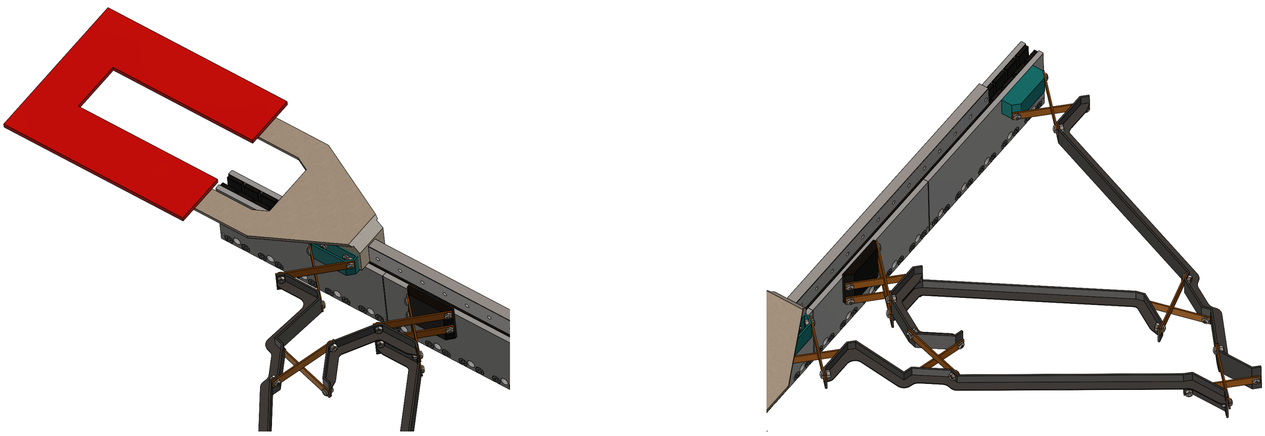
1) Add reinforced folded leaf springs: With the mover having an extra DoF, specifically the rotation around the z-axis, reinforced folded leaf springs could be added to create a one DoF system. The linear motion in the y-direction would be reinforced and the rotation will disappear. Reinforced folded leaf springs are ideal, because of their large range of motion that is needed for the mover as it has to move around 100mm in both directions. One big downside of these compliant mechanisms is that there is enough space needed to implement these flexures. With the mechanism already sitting in the top part of the design space, there is no room at the top or bottom to add the leaf spring that counteracts the rotation.

2) Add a second pantograph: With the actuator base being linearly guided, the end effectors of the pantograph will move in a linear motion. But due to the use of compliant mechanisms, the mover will rotate because of the stiffness in the flexures. If the pantograph would te be mirror to the other directions, this rotation will be cancelled out due to the mirrored motion of the newly added pantograph. In addition to this, the out-of-plane stiffness will also be divided over two mechanisms and the symmetric design allows for the cancellation of inertia force components in the direction perpendicular to the blade's motion. However, one downside of this new design is that the actuators can no longer be located next to each other. For this the other subgroups had to be informed one last time and in the end it got accepted.

Final Design: Thus the decision has been made to add another pantagraph on the other side of the actuator. This means the actuators will be moved from next to each other to a location where they are on top of each other. How this fits in the design space can be seen in subsubsection 3.1.3. The motion of the whole system is now very symmetric, which is ideal for the exported forces of the blade actuation. The exact motion of the new mechanism has been visualised in CAD in subsubsection 3.1.2 and the FEM analysis can be found in subsubsection 3.2.1. An updated Bill of Materials has been given in subsubsection 3.1.4 together with the material selected. Some detailed pictures of the CAD file including the connection to the blades can be found below, only one pantograph has been displayed to keep is clear and structured:

3.1.1 Details of the Mechanism

In Figure 10a, the attachment of the blade to the mechanism can be seen. The blade itself is integrated in a bracket, which can be connected to both the mover and the mechanism via 2 bolts. This makes for a more simple connection with less parts which is ideal. In Figure 10b, the pantograph can be seen in more detail.

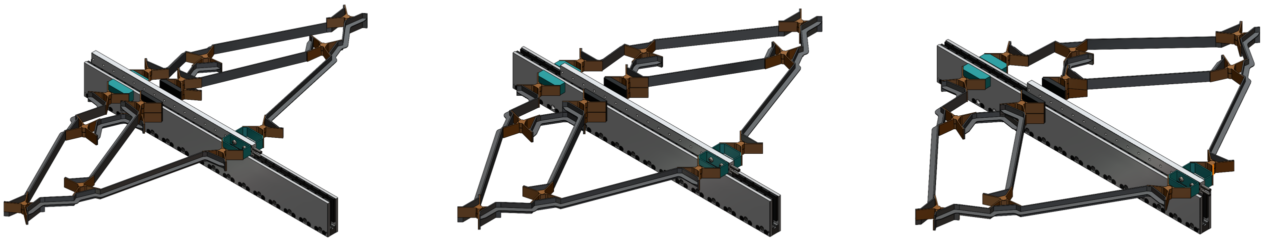


(a) How the blade is attached to the mechanism

(b) Detailed view of the Pantograph

Figure 10: More details about the finalized design

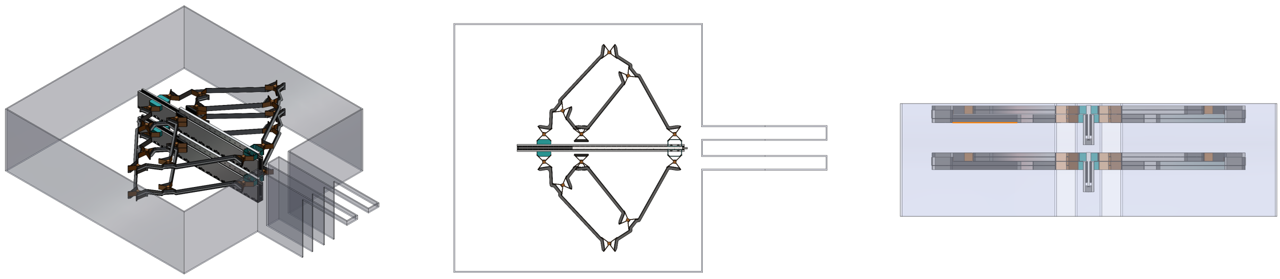
3.1.2 Movement of the Mechanism



(a) Blade to the Left (b) Mechanism in Rest Position (c) Blade to the Right

Figure 11: Visualisation of the mechanism in 3 different positions: $-100mm$, rest position and $+100mm$

3.1.3 Mechanisms in the Design Space



(a) View of the stacked mechanisms (b) Top view of the mechanism (c) Mechanism stack visualised

Figure 12: Visualisation of the mechanism in the design space/budgeting

3.1.4 Bill of Materials

This bill of material has been made for one pantographs. For the whole mechanism this list therefore has to be multiplied four times. This has been done to keep the list nice and structured. The actuator has been left out of this bill of materials because that belongs to another subgroup.

Bill of Materials for one Pantograph			
Parts	Quantity	Description	Material
Flexures	12	2 will be used in each joint	Stavax Supreme
Frame Connection	1	Connects the pantograph to the frame	Carbon Fibre
Mover Connection	1	Connects the pantograph to the mover	Carbon Fibre
Actuator Base Connection	1	Connects the pantograph to the base	Carbon Fibre
Outside left arm	1	Left outside pantograph arm	Aluminium 7075
Outside right arm	1	Right outside pantograph arm	Aluminium 7075
Inner left arm	1	Left arm on the inside of the pantograph	Aluminium 7075
Inner right arm	1	Right arm on the inside of the pantograph	Aluminium 7075
M4x8 Bolts	30	Bolts to connect the leaf springs and actuator	Steel
M4 Nuts	22	Nuts to connect the bolts to the leaf springs	Steel
Blades	2	Blades to block the laser	Stainless steel
Blades Bracket	1	Connect the blade to the actuator/pantograph	Carbon Fibre

Table 4: Finalized bill of materials for one Pantograph

3.2 Subsystem Performance Estimation

In this section, we will provide an explanation of the *Performance Analysis* conducted on the mechanism. The evaluation focuses on two key aspects: the **Driving Stiffness** and the **Supporting Stiffness**. This assessment allows us to determine the required driving force for the actuator and also characterize any parasitic displacements of the blade. Additionally, a **Multibody-Dynamics (MBD) Simulations** is provided to assess the balancing performance of the mechanism and the magnitude of exported forces.

In order to assess the stiffness performances, first a Kinematic model and a Pseudo Rigid Body Model (PRBM) of the mechanisms have been realized in python (code and further description accessible at: link). This models helped us also in the design phase, to estimate faster the optimal dimensions for links and flexure-joints as explained in subsection 2.4. In addition different FEM simulation have been conducted. The software used is *COMSOL - Multiphysics*.

To perform the Multi-Body Dynamic Simulation (MBD), another python script was written(code and further description accessible at: link). With python a simplified enough dynamic model of the system was implemented and studied.

3.2.1 Driving Stiffness Estimation

The first simulation objective was to assess the driving stiffness. For this purpose two equal and opposite boundary-loads were applied on the base and the mover to simulate the force produced by the actuator. The values of deformation corresponding to different actuation force values were recorded. Afterwards, the corresponding deformation was compared with the estimated values obtained from PRBM. The comparison can be observed in Figure 13. The resulting estimated stiffness is around $K_u \sim 0.7 * 1e3N/m$.

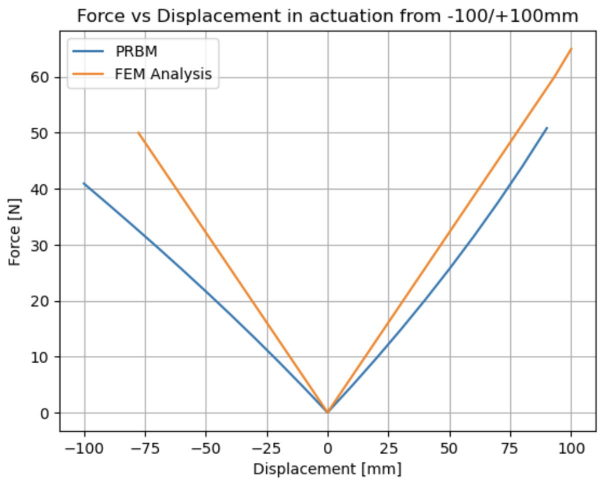


Figure 13: Comparison between PRBM and simulation results

Moreover two other plots were produced and can be observed in Figure 14. One, representing the total deformation magnitude in all points of the mechanism, and the other, representing stresses in the joint. The maximum stresses result less then the maximum Yield Strength (1.28 GPa) of STAVAX SUPREME, the material used for the flexures. Both of these plots correspond to maximum static displaced position (100 mm).

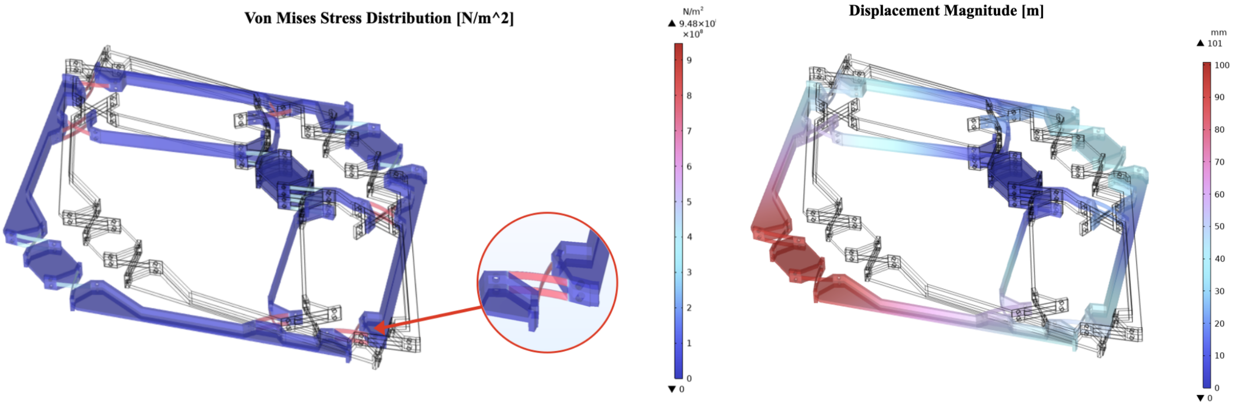


Figure 14: Driving stiffness FEM analysis results

3.2.2 Support Stiffness Estimation

In order to evaluate the supporting performances of the mechanism, another FEM simulation was performed. In this case, the boundary condition were maintained the same, but two more loads were added. The first one is the weight of the mechanism, in order to understand the mechanism self-supporting performances. The second one is the weight of the blade an the actuator mover, added in correspondence of the mover. To understand better the position of these components in the mechanism, the scheme in Figure 4 can be observed.

In Figure 15 these parasitic deformations due to gravity can be observed. The deformation are shown in correspondence of the maximum positive linear displacement of the mechanism (100 mm), because is the position where the moment arm of the blades and mover loads is the highest.
Poorly the parasitic deformation is quite high $w \sim 1cm$. Moreover it has to be noted that the weight of the actuator's base has not been taken into account, since it would be supported by another linear guide developed by the drive-train subsystem.
For this last simulation, a slightly different model, compared to the final designed one, was used. However, all the dimensions are correct, and for the purpose of the simulation the two models can be considered equivalent. Another assumption regards the boundary load of the mover. Indeed, the load due to the weight of the actuator's mover was considered as a concentrated force on the end-effector of the mechanism. However, the mover in the real design is not supported exactly at his center of mass. Thus, a bending moment will be generated and would impact even more the displacement of the mover.

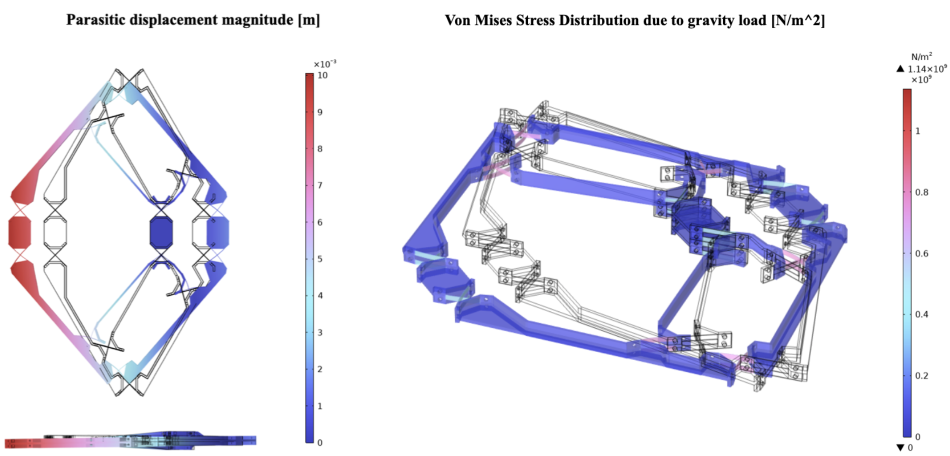


Figure 15: Support stiffness FEM analysis results

3.2.3 Multibody Dynamics Simulation

As explained at the start of the Section, a dynamic model of the mechanism was implemented with the use of python and mainly the *sympy.mechanics* library. Different assumption were made:

- The links of the mechanism are approximated as bars with a uniform mass distribution
- The spring force due to the flexure-joints is not considered. (interest in inertia forces)
- The mechanism is studied in the plane, thus only two reaction forces(Rx, Ry) are calculated.
- A single pantograph was considered due to the symmetry of the problem.

To estimate the exported forces during the normal functioning of the mechanism, a movement with $400m/s^2$ of acceleration and a cycle frequency of $5Hz$, with $100mm$ displacement has to be simulated(information provided by ASML). The desired acceleration profile is displayed below:

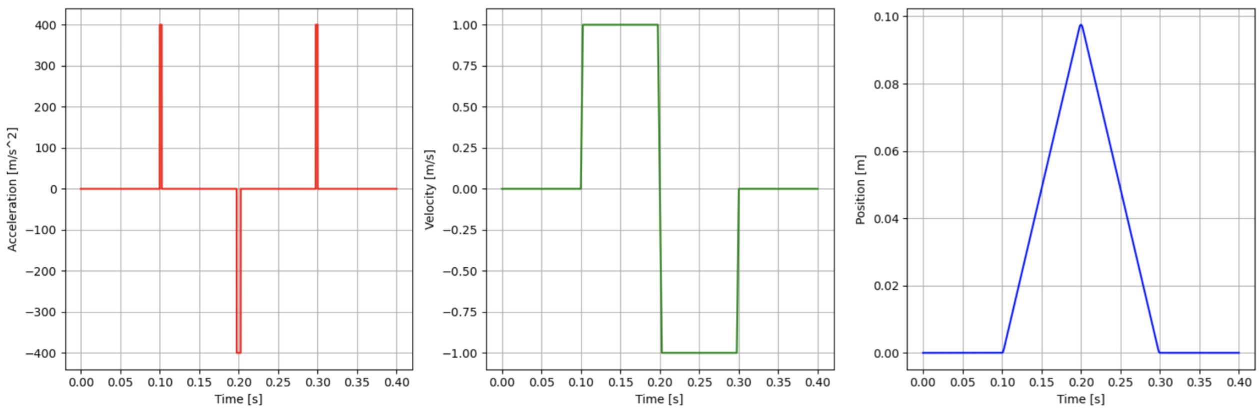


Figure 16: Desired acceleration, velocity, and position profiles of end-effector during operation

Once the desired acceleration profile is defined, a force with the same temporal profile can be defined and applied to the actuator mover and to the actuator base, in two opposite directions. This force simulates the real force applied on the mechanisms by the actuator. The amplitude value of the force applied is $F = 800N$.

Once the temporal force profile is set, the equation of motion is integrated. Afterwards, the output values of the integration (generalized speeds, force profile, and generalized coordinate) can be used to compute the magnitude of the exported force on the fixed base. The temporal behavior of all the coordinates and reaction forces can be observed in Figure 17.

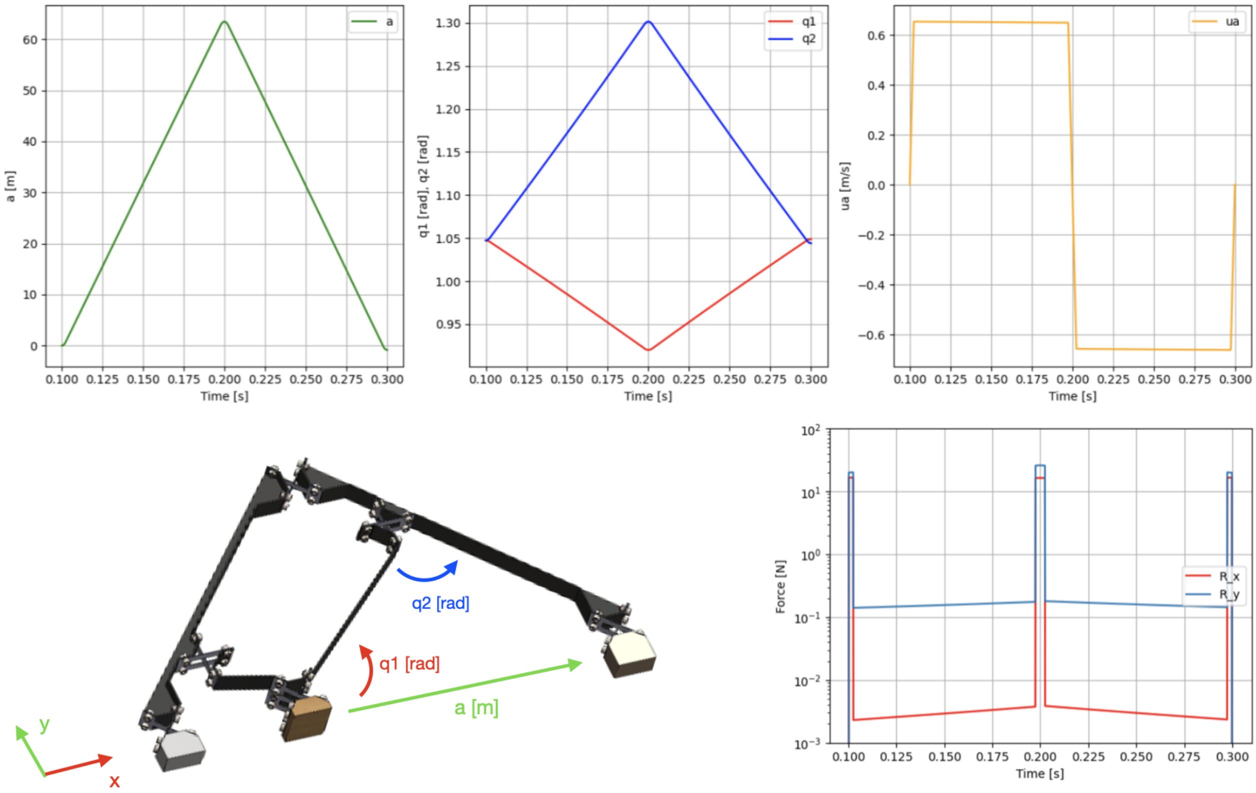


Figure 17: MBD simulation - results

From the plots, it can be observed that the maximum exported force both in x and y is around $20N$. However, due to the symmetry of the final design, not taken into account in this simulation, the y component would be automatically equal to zero. Moreover, the x component of the reaction forces, Rx, can be brought to zero too. Indeed, by performing different simulation and tuning the mass of the mover (additional $0.81Kg$ in this case reduced the reaction force from $200N$ to $15N$), also the Rx component can be almost balanced completely.

The simulation was useful to prove that the mechanism can be completely balanced by tuning the masses. This perfect tuning was not performed in this case, because the model remains an approximation of the real model. To tune correctly the mechanism, the simulation model should be improved and made more accurate.

3.2.4 Displacement of the center of gravity

Looking at the plot in Figure 18 the trajectory of the COM of one pantograph during deformation can observed.

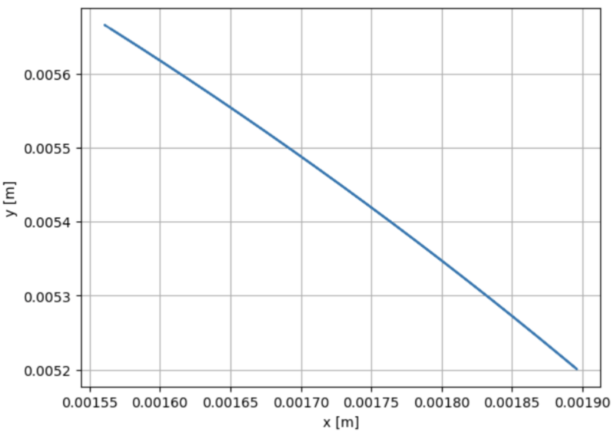


Figure 18: Movements of the center of mass of the pantograph - during actuation

The movement as expected is non-linear. From the plot, it can be observed that the movement is very small and the position is close to the fixed base. Thus, low exported moments due to gravity are expected on the fixed base.

3.3 Budget Analysis

In this section, the revised budget explained in Table 3 will be analysed. This is being done to check if the final design is within the designed budgets. A list has been given as an overview:

- Space 40%: This designed budget has not been kept within the limits, but there is a good reason for this. As explained in this final part of the report, a final design change has been applied in which a mirrored pantograph is added to each actuator. 2 extra pantographs and the actuators being on top of eachother means that assigned space has been increased to around 70%, but this has been thoroughly discussed with the system engineers.

- Thermal 0%: The budget value for the heat production in this subsytems is kept within the limits as the compliant mechanism will not produce any heat. Heat production is expected in the drivetrain and anti-crash design groups.

- Gas 0%: This value has also been kept within the limits, as the pantograph and the other components will not produce gasses. A investigation for the use of carbon fibre has been done, but it was concluded that, if fully cured, carbon fibre will not outgas.

- Forces 45%: Due to the double pantograph's use, this system's exported force has decreased. According the simulations that have been run, the exported forces in y-direction will fully be compensated if the masses have been leveled out exactly. If this has not been done correctly, there will be a slight exported force. The exported force due to the inertia of the pantograph in x-direction is fully compensated due to the double pantograph design. That is why it can be concluded that this system is well within the budgeted value, specifically is has been reduced to less than 10% for the entire system.

- Deviation 30%: It is difficult to estimate the actual value for this budget, but parasitic movement simulations have been run. Though the mover has probably some extra movement due to the way it has been connected to the pantograph, this will be discussed in the Risk Evaluation in subsection 3.4.

- Start up time 0%: The compliant mechanism does not have a start up time, it can be actuated straight away. That is why this system is within this limit.

3.4 Risk Analysis

3.4.1 Parasitic Movement in Pantograph

The parasitic movement in the pantograph could cause for some problems in the accuracy of the blades. While the parasitic movement is estimated on around 1mm and the distance between the two blades is 10mm, this could still be a small problem. Especially when kept in mind for the risk mentioned in the next subsection. But overall, the parasitic movement will not be expected to be a huge problem but it should be kept in mind.

3.4.2 Moment in Mover due to Pantograph connection

The mover, blade and the blade bracket is quite a long assembly. The only connection point for this assembly is on the two pantographs on both sides. This means that if this connection is not exactly in the middle of the blade assembly, the mover can position itself on an angle. While this system has been designed to have the centre of mass roughly on the connection point, the system has a risk to create moments due to the exported force and huge accelerations. This risk is estimated to be great, but it could be solved by adding reinforced folded leaf springs in the xy-plane. These compliant mechanisms have a big range of motion and can carry the loads in the ou-of-plane direction. However, in this paper no emphasis has been put on the mechanism because it is considered fairly easy to implement and there is room for such a mechanism.

3.5 Evaluation of Main Assumptions

In addition to the necessary specifications and boundary conditions supplied by ASML, a few additional assumptions were made in response to comments on our detailed design report, discussions with the system engineer, and meetings with other subsystems. Here is a list of these presumptions:

3.5.1 Assumptions Regarding Actuation

Multiple assumptions regarding the actuation provided were also made.

- It is presumable that the force delivered by the actuator will be linear and consistent, with little chance of variation affecting the mechanism's final movement. This is still valid due to the sophisticated actuator that has been selected.
- The parasitic motion is limited and kept as low as possible by the linear guide attached to the base. This assumptions is also still valid and investigated by the drivetrain subgroup. The parasitic movement of the actuator base is counteracted by the design of a compliant mechanism with a high out-of-plane stiffness.
- The moment in the z-direction will be limited by the close positions of the actuators towards the top. This assumptions has been changed because the actuators are not located next to each other anymore. This means that the lower actuator has an arm in the z-direction to bring the bottom blade to the correct height. So this will create a moment, but it is expected that this moment will not exert the 400N of budgeted exported force. And the actuator base could be balanced to account for this moment.

3.5.2 Assumptions Regarding Mechanism

- The compound linear guide that was initially planned to provide the required linear motion in the y-direction ended up needing to be modified because it would have required the entire design area, hence making it infeasible
- Initially, it was thought that the mechanism would be balanced by adding a counterweight that would move in the opposite direction from the blades. However, as described in subsection 3.1, two pantographs will now be attached to the actuator and will be mirrored to one another.
- It has also been assumed that the performance of the mechanism will not be affected by temperature changes brought on by the high speed movement of the mechanism. As a result, the designed mechanism won't deform or become out of alignment because the thermal forces are minimal or nonexistent in comparison to the forces generated by the actuator. This assumption still has to be remained valid as it is difficult to estimate how the heat produced by the actuator and the anti-crash system.

4 Conclusion

In this paper, a potential system in the ASML EUV Lithography System has been developed. This system would be a machine that moves two blades in vacuum that block the light of the laser in a scanning motion. In the beginning this system was divided into four subsystems of which the blade mechanism become the focus of this subgroup. The design goal: Design a independent linear blade motion mechanism in the y-direction with a balancing mechanism to even out the exerted forces.

The project started of with the development of the conceptual design report. Before concrete steps could be taken, there was a brainstorming session with all the groups about the design specifications and the budget every group has to take into account. The first step was to identify the functional requirement the mechanism has to fulfill, which ended up with 2 crucial functions. For every function a literature study has been performed to come up with state of the art solutions. After the design specifications were concrete, three concepts where retrieved from the morphological chart. With this being the conceptual design report, the concepts where not detailed yet and even a new mechanism would be introduced in the next report. The selection of the final concept was mainly based on the evaluation and not a weighted table which is the general step taken in such a project. This decision has been made because there were only two functional requirements.

In the detailed design phase, the bases of the final system was introduced: the pantograph. Because of the complex integration of this mechanism in this system, extra time has been taken to explain how the mechanism works. With the help of the chosen mechanism and kinematic models, the optimized dimensions for the model were chosen. The conclusion was made that the base of the actuator needed to be linearly guided and this task was given to the drivetrain subgroup. Unfortunately, because a new concept was introduced in the beginning of this phase, the guidance of the mover of the actuator had not been thought of. This would lead to another design change in the next section. Nonetheless, a performance analysis could be performed on the pantograph as this would not be influenced due to the later design change.

In the critical design report, one final change has been introduced in the system. While the previous design had the two actuator next to each other in the middle of the design space with one pantograph each, the mover would still have an extra degree of freedom. Multiple solution have been evaluated and in the end a second pantograph per actuator was added. This meant that the actuator could no longer be located next to each other, but rather on top of one another. Multiple new performance estimations have be done including a FEM analysis vs the PRBM results and a parasitic displacement analysis. Together with this, multibody dynamics simulations have been made to estimate the exported forced during normal functioning of the mechanism. The results confirmed what the mechanism had been designed for: the exported forces created by the fast acceleration of the blades was being counteracted by the actuator base.

In conclusion, the mechanism is a complicated but viable solution. The budget exported forces was 45% of the 400N specified by ASML, but according to the multibody simulations this could be decreased to less than 10% in total. The desired actuation stiffness of the pantograph has been reached to make sure the actuator can work. The only drawback of this mechanism is the existence of parasitic movement which brings the accuracy of the mover in danger. Even though the mover is being held up by two pantographs, there could be undesired movement. This however, can be solved by adding reinforced folded leaf springs but the focus has not been put on the mechanism due to the complexity of the already existing system. If this would be achieved, the subgroup believes this mechanism could be viable and is optimal considering the lower mass and price due to the removal of a counter masses and potential other actuators.

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A Appendix

A.1 Team Member Contributions

**Except System Engineer means that Pepijn worked on the paper of the system engineers while the rest was working on the conceptual and critical report due to distribute the work.*

Week 7 - Q3	**Hours**	
Tasks	*As a Group*	*Individual*
Group Meeting	2	
System Engineers Meeting		1 - Pepijn van Kampen

Week 1 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen

Week 2 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen
Concept generation	2 - Except System Engineer	

Week 3 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting (2x)	4	
Making Morphological Chart		1 - Tom Vis
Individual work on Report 1	2 - Except System Engineer	
System Engineers Report		2 - Pepijn van Kampen

Week 4 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen
Individual give Feedback	1	
Individual Draw Concepts	1	

Week 5 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen
Work on the CAD file		8 - Tom Vis
Work on the CAD file		2 - Riccardo Girolamo
Work on simulations		7 - Riccardo Girolamo
Work on dynamics		5 - Joost Bankras
Additional work on Report 2		2 - Rutwik Patil
Individual work Report 2	2	

Week 6 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen
Individual give feedback	1	
Additional CAD work		2 - Tom Vis

Week 7 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Report		2 - Pepijn van Kampen
Reviewing dimensioning of the mechanism		2 - Rutwik Patil
Additional CAD work		4 - Tom Vis
Additional CAD work		3 - Pepijn van Kampen
Additional Performance Estimations		5 - Riccardo Di Girolamo
Additional Performance Estimations		6 - Joost Bankras
Individual work on Report 3	2 - Except System Engineer	

Week 8 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Lecture Meeting	2	
Group Meeting	2	
System Engineers Meeting		2 - Pepijn van Kampen
Individual Give Feedback	1	

Week 9 - Q4	**Hours**	
Tasks	*As a Group*	*Individual*
Rewrite Report 1		4 - Tom Vis
Rewrite Report 1		1 - Pepijn van Kampen
Check Changes Report 1		1 - Rutwik Patil
Check Changes Report 1		1 - Riccardo Girolamo
Rewrite Report 2		2 - Riccardo Di Girolamo
Finalising Perfomrance Estimation		6 - Riccardo Di Girolamo
Check Report 2		2 - Joost Bankras
Check Report 2		1 - Pepijn van Kampen
Rework Final Report		7 - Tom Vis
Rework Final Report		1 - Rutwik Patil
Rework Final Report		1 - Pepijn van Kampen
Individually Check Final Report	1	

Pepijn van Kampen	**Tom Vis**	**Riccardo Di Girolamo**	**Joost Bankras**	**Rutwik Patil**
67 hours	75 hours	72 hours	62 hours	55 hours