ALMA MATER STUDIORUM - BOLOGNA UNIVERSITY

SCHOOL OF ENGINEERING AND ARCHITECTURE

MASTER DEGREE IN AUTOMATION ENGINEERING

THESIS IN MECHANICS OF MACHINES FOR AUTOMATION

MOTORIZED SYSTEM FOR OPENING AND CLOSING A DOMESTIC DISHWASHER

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1 INTRODUCTION

Technological advances that we are experimenting in the last decades have the goal of making our lives easier and comfortable. Simple and ordinary actions that we compute everyday are getting replaced by automatic systems and/or robots: cars that drive autonomously, robots that clean out houses, windows that open and close automatically…These tireless substitutes have the goal of avoiding us strenuous and possibly dangerous actions.

Pursuing this idea, the company Nuova Star spa, in collaboration with the University of Bologna, Alma Mater Studiorum, decided to propose a new concept of automatic system in the domotic branch: a motorized dishwasher capable of opening and closing autonomously.

This application, in addition to simplifying daily actions, has the specific purpose of helping old or disabled people who have difficulty in moving the heavy appliance door. From a market point of view the requests are many and Nuova Star's costumers have repeatedly asked for such solution.

In the past, a similar concept was developed for ovens doors only. However, this solution interested only a very high-end market that looks for novelty and refinement rather than utility of the application itself. So the target was mainly the beauty of the movement of the door, leaving economic savings in background.

In this thesis I will therefore describe the developments and functionalities of a motorized dishwasher whose target is the kitchen of the average user, requiring a modest expense, bringing comfort and innovation at the same time.

2 NUOVA STAR S.P.A.

Founded in 1967, it deals with designing, making and marketing hinges for household appliances; is currently the leader in this sector and sells his products all over the world, meeting every kind of request.
The company is located in Zola Predosa, near Bologna, and has about 150 employees. The individual components of the hinges are made of sheet metal, through mechanical molding carried out with presses that ranges from 130 to 400 tons. In a second department the components are then assembled through lines that are partly automated and partly manual. There are numerous patents held by the company that protect both the geometry of the hinges and the assembly technologies.

Until few years ago the main products of the company were oven hinges. In recent years, however, dishwasher ones are becoming a key product for company expansion.

3 DISHWASHER HINGE

The term hinge stands for a mechanical assembly whose main function is to connect the appliance structure to the door. Each dishwasher has two hinges mounted on its bottom section.

Their fundamental characteristic is the capability of balancing the weight of the door. Thanks to the addition of a mechanical lever system, that returns the motion to a properly selected spring, the weight of the door is gradually balanced. The effort that the user must perform in order to open and close the appliance is minimal. Secondary functionalities are:

- Silence;
- Compactness;
- Resistance to many life cycles;
- Smart assembly;
- Flexibility (means the possibility to adapt to different door weights);

As a first step we need to identify the kind of hinge on which developing the system. Actually 3 models exist:

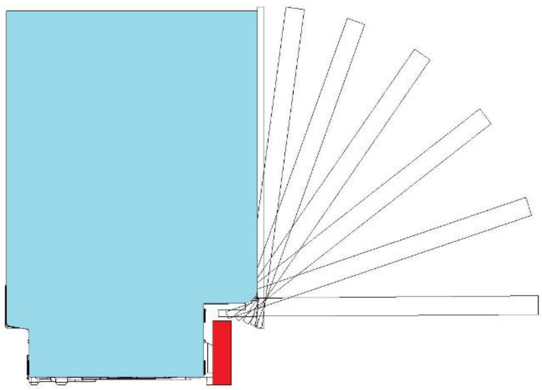
1. FIXED FULCRUM hinge: allows to rotate the door around a fixed point;
2. VARIABLE FULCRUM hinge: allows the roto-translation of the door, the goal is to prevent the crash of the decorative panel fixed to the door against the base usually located below the appliance;
3. SLIDING hinge: development of the fixed fulcrum hinge, in addition to the pure motion of rotation, a secondary mechanism imposes the outside translation of the decorative panel in order to avoid the impact with the bottom base;

We choose to develop the motorized system for the VARIABLE FULCRUM hinge since it represents the worst-case scenario. This means that's the most complicated condition to solve. Once a solution for this hinge is found, it will be quick and intuitive to develop the solution for the two remaining models.

Let us now examine more specifically the operation and components of the variable fulcrum hinge.

The recent design of modern dishwashers requires the presence of a wooden plinth placed in the bottom part of the dishwasher which has the function of covering the supports of the appliance. At the same time, a very protruding decorative panel is attached to the door. Therefore the application of a single rotational motion would lead the panel to impact against the plinth with disastrous consequences for the entire system.

A possible solution is to make the door perform a trajectory specifically designed to translate outside. This short translation is sufficient to avoid contact without being perceptible to the eyes. In the next picture some door reference positions are showed.



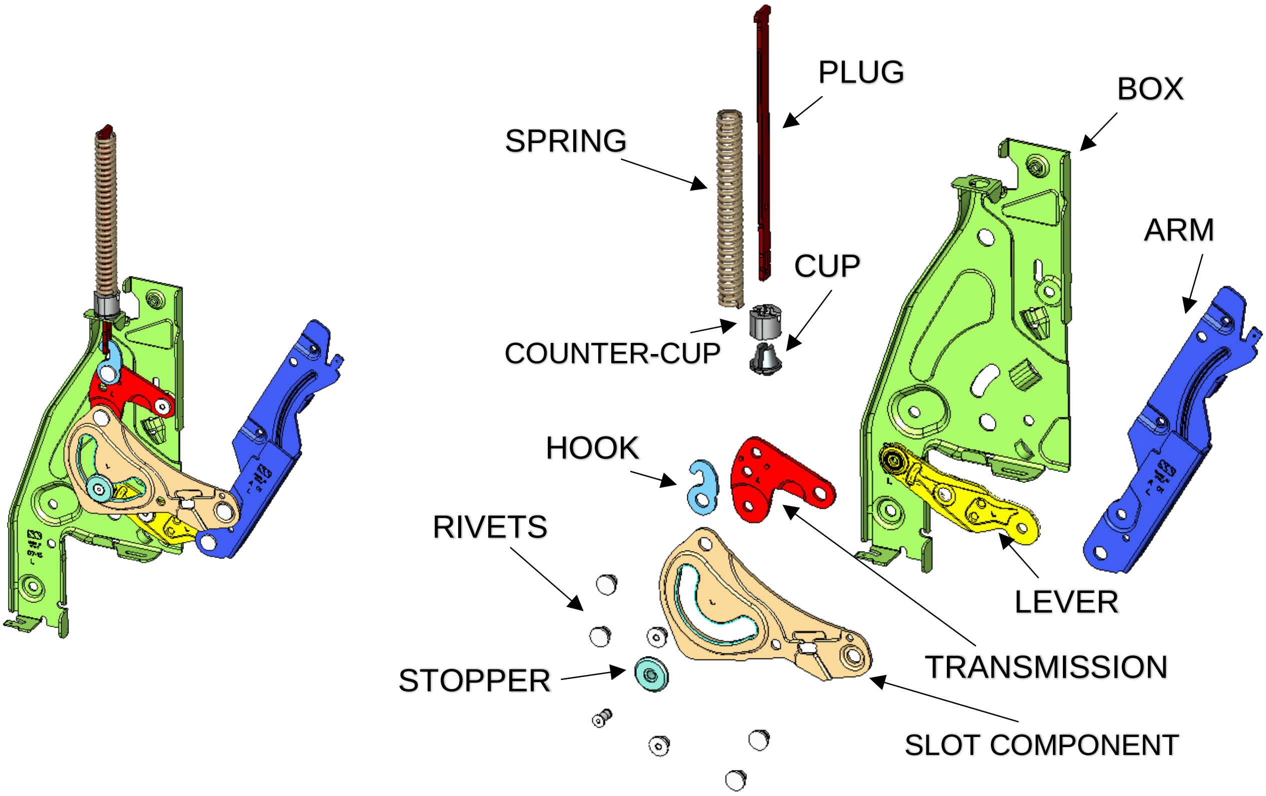
The components that make up this complex mechanism are:

- BOX: represents the frame of the hinge to which all the components are connected, it also has few limits that block the motion between 0° and 88°;
- ARM: acts as a fastening for the door and is fixed to it, has also a U shape that makes it more robust avoiding deformation;
- SLOT COMPONENT: fixed to the arm, it has a well-designed slot that determines the trajectory of the door. It is also a transmission element for the force generated by the spring;
- LEVER: joins stopper, arm and box;
- TRANSMISSION and HOOK: join the slot component to the plug. The particular shape of the hook allows the plug to translate along a single direction without rotating;
- PLUG: acts as a housing for the spring, during the opening phase it compresses and vice versa;

All these previous components are made by Nuova Star using mechanical molding, the material is stainless steel with a thickness between 1.5 and 2.5 mm.

- SPRING: it is the key element of the hinge, it collects the potential energy released by the door during the opening phase and then returns it during the closing phase. This energy exchange allows the user to avoid a huge effort during the closing phase;
- CUP and COUNTER-CUP: used together, they cancel out the action of the spring in the first 15 degrees of opening. By omitting these two components it is possible to observe how the door tends to close on this angular interval. Both components are made by an external company using PA 6.6 30% g.f .;
- RIVETS: commercial components used to connect the previous ones;
- STOPPER: acts as a sliding runner inside the slot. It is made of PA 6.6 30% g.f. in order to avoid noise and vibrations. Moreover its geometric shape generates a strong friction against the component with slot.

The friction generated by the stopper has the function of making the hinge adaptable to many doors. In fact, each door with its decorative panel has a particular weight that changes depending on the customer and on the model of the kitchen. As already mentioned, the fundamental objective of the hinge is to balance, as well as possible, their gravitational thrust. Therefore, each weight-door corresponds to its own ideal spring, capable of perfectly counteracting this thrust but it would be necessary to produce an infinity of hinges different from each other for the thrust generated by the spring. Instead, by generating a strong friction between the stopper and the component with a slot, it is possible to dissipate a part of the gravitational energy of the door. In this way, each spring balances a weight range of approximately 4kg.



4 REFERENCE DISHWASHER

Let’s now consider the reference dishwasher structure. Generally we can split it in three main parts: the washing chamber, the devices area and the door.

- Washing chamber: it’s a thin steel hollow in which dishes are washed, since it must be in watertight, we can’t add holes on it.
- Devices area: placed on the bottom side, it supports the washing chamber. It’s made with plastic and so it’s not too robust. All the hydraulic components of the appliance are fixed to it. For prototyping purpose we use a dishwasher with an empty devices area in order to have more available space;
- Door: it’s the visible component of the dishwasher, it’s also the most influence in hinge set up since its weight and its center of gravity influence al lot the device. In our case we consider a door with a decorative panel which total weight is 11 kg and center of gravity positioned at 350 mm from the bottom edge;

Now the hinge is the only component that links these three sections and it’s also very robust. During the prototype development we have to fix all the new elements to it, otherwise we could introduce structural criticality.

5 REQUIREMENTS

After this brief explanation we now deal with the concept of motorized system. Firstly we need to define functional requirements then we will go on with the design itself.

The fundamental prototype characteristics are:

- Door trajectory should range between 0° and 88°;
- Opening/closing cycle time should range between 6-8 seconds;
- Door motion should be sufficiently fluent, no vibration should be visible;
- No excessive noise should be noticeable during the cycle;
- New design should not compromise the structure of the dishwasher;
- System should work after 100’000 cycles;
- Two push buttons should allow to open or close the appliance;
- If an obstacle impacts the door during a cycle, the actuation must immediately be interrupted;

Secondary characteristics are:

- Reversibility: should be possible to open and close the door manually;
- As an alternative to push buttons, a vocal recognizer sensor should allow to open or close the appliance;
- Total application price should range from 20-40 €;
- If motor is AC then it should have max rated voltage of 220V at 50 Hz, otherwise if DC should require maximum 48 V;

For solving all these requests we need to split the entire job in smaller sections and solve them one by one.

The small sections that we identify are three and represent the usual division computed in machine development:

1. Mechanic: consists in computing kinematic analysis understanding how the single components move and interact, draw the new components simulating and validating them, finally realize them all;
2. Electric: consists in identifying the motor, the driver, all the electronic components and mount everything on the dishwasher;
3. Software: consists in writing the program that the controller has to compute;

6 MECHANIC DEVELOPMENT

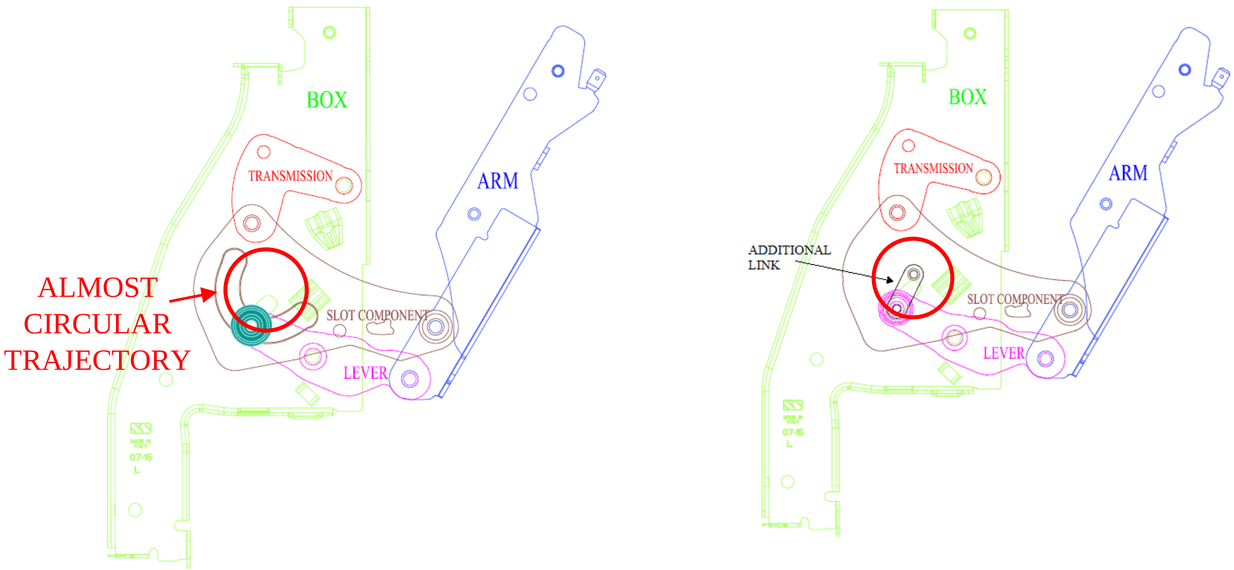
This section is probably the most complicated and requires a lot of studies and trials.

Developing a smart mechanical design is essential for this application. First, we need to think about available space: since it's very limited we need to minimize all components dimension. Then we have to think about price, we want an advanced but cheap application, so material and design should be developed in this sense.

We start computing kinematic analysis, that are focused on identifying the trajectories and the forces between the components. We adopt three different approaches: graphic, software and numerical.

Before introducing them all, we highlight some assumption that we will make. First of all, we don't consider the hook, the plug, the cup, the counter-cup and the spring. Their function is just to apply a vertical force to the transmission element. Thus, instead of considering them all, we just consider the force that they generate on the transmission element.

A second assumption that we will compute regards the slot component. As we can see in the following picture, the slot shape looks like a circle, leading the stopper to rotate around the center. From a kinematical point of view this is equal to have an additional link with two rotational joints; the length of the link is equal to the radius of the circle, one rotational joint is positioned in the center while the other is connected to the lever replacing the stopper.



Further we check the validity of the new model by using Grubler formula for 2D systems:

$$M = \text{degrees of freedom} = 3(n - 1) - 2j_p - j_h$$

Where n is the total number of links in the mechanism, j_p is the total number of primary joints and j_h is the total number of higher order joints. In the model with the additional link we have a total of 7 primary joints, 0 higher order joints and 6 mechanical links.

Thus the total number of *dof* is:

$$M = 3\ (6 - 1) - 2 \times 7 - 0 = 15 - 14 = \mathbf{1\ dof}$$

This is a right and expected result since the door can only rotate around a variable fulcrum. So the additional-link assumption is correct.

An additional consideration is obtained from practical constraints: looking at the hinge itself we can’t actuate few components. For example the lever component is located between box and slot one, so the space for accessing it is very limited. Its actuation requires to redesign all the hinge, this must be absolutely avoided due to high costs.

Other components are not rigid enough. Applying forces to them would bend all the hinge and cycle after cycle it would collapse. For example, the transmission element is designed to sustain just spring vertical force, applying an additional force with a different orientation would damage the entire hinge.

So in the next chapter we will not consider these critical components.

7 KINEMATIC ANALYSIS

7.1 GRAPHIC METHOD

We solve the problem for few significant positions by using Autocad software. The steps are:

- identify the CIR (center of instantaneous rotation) for each couple of elements;
- measure the distances among them that represent the arm lever;
- apply the force equilibrium criterion that states;

$$M_i \times \Omega_i = M_j \times \Omega_j$$

Getting:

$$M_i = \frac{\Omega_j}{\Omega_i} \times M_j$$

Where M_i term represents the torque imposed to element i while Ω_i is the arm lever obtained as :

$$\Omega_i = CIR_{i\text{-}j} - CIR_{j\text{-}frame}$$

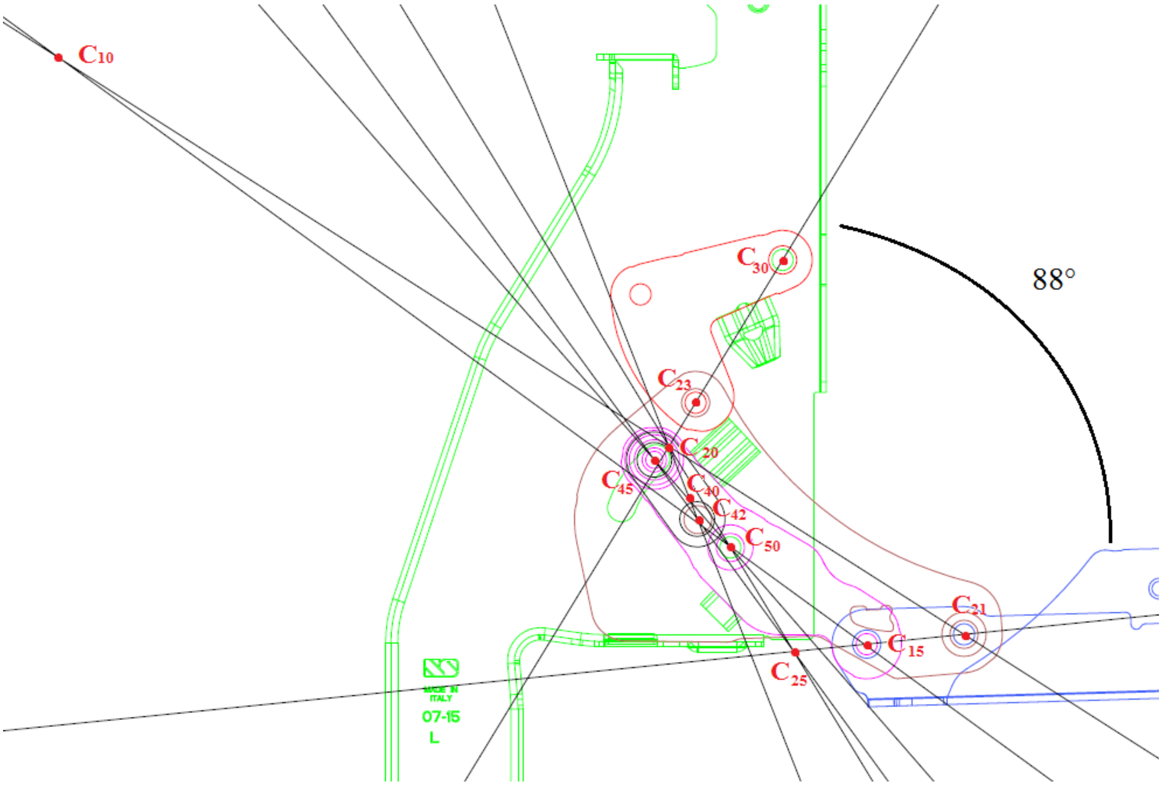
So that $\frac{\Omega_j}{\Omega_i}$ is the transmission ratio.

The target of this analysis is to find the couple of elements that maintain the most constant transmission ratio. We do this study for just few hinge positions because of lack of space.

LEGEND:

COMPONENT	LINK NUMBER
BOX	0
ARM	1
SLOT COMPONENT	2
TRANSMISSION	3
ADDITIONAL LINK	4
LEVER	5

88 DEGREES CASE:



$$M_5 = \frac{\Omega_1}{\Omega_5} \text{ x } M_1 = \frac{C_{15}-C_{50}}{C_{15}-C_{10}} \text{ x } M_1 = \frac{47{,}96\ mm}{744{,}47\ mm} \text{ x } M_1 = \mathbf{0{,}0644} \text{ x } M_1$$

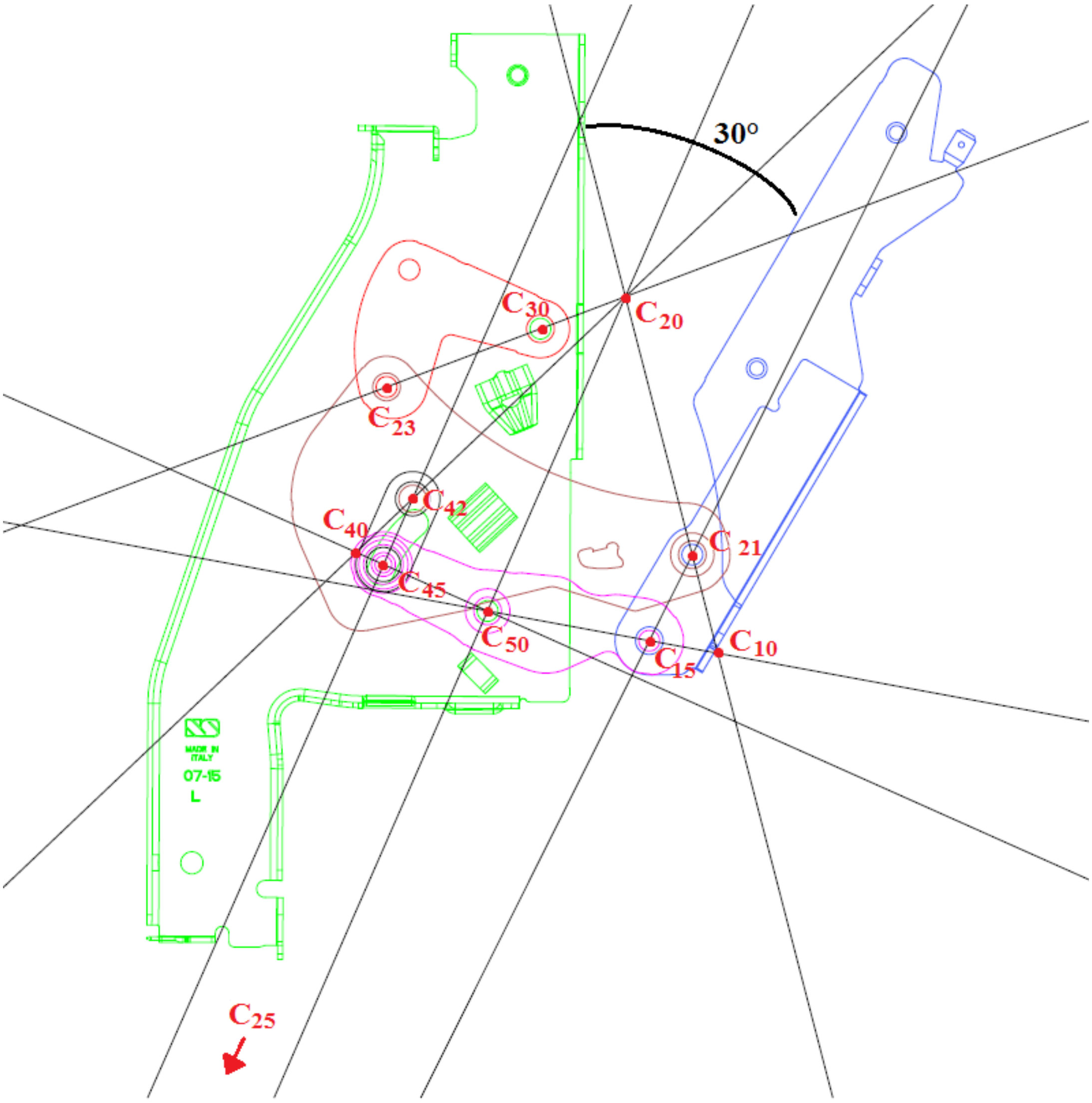
$$M_2 = \frac{\Omega_1}{\Omega_2} \text{ x } M_1 = \frac{C_{12}-C_{20}}{C_{12}-C_{10}} \text{ x } M_1 = \frac{102{,}92\ mm}{728{,}01\ mm} \text{ x } M_1 = \mathbf{0{,}1414} \text{ x } M_1$$

$$M_2 = \frac{\Omega_5}{\Omega_2} \text{ x } M_5 = \frac{C_{25}-C_{20}}{C_{25}-C_{50}} \text{ x } M_5 = \frac{83{,}8\ mm}{38{,}1\ mm} \text{ x } M_5 = \mathbf{2{,}199} \text{ x } M_5$$

$$M_4 = \frac{\Omega_2}{\Omega_4} \text{ x } M_2 = \frac{C_{24}-C_{40}}{C_{24}-C_{20}} \text{ x } M_2 = \frac{7{,}429\ mm}{36{,}507\ mm} \text{ x } M_2 = \mathbf{0{,}2035} \text{ x } M_2$$

$$M_5 = \frac{\Omega_4}{\Omega_5} \text{ x } M_4 = \frac{C_{45}-C_{50}}{C_{45}-C_{40}} \text{ x } M_4 = \frac{33{,}302\ mm}{14{,}88\ mm} \text{ x } M_4 = \mathbf{2{,}238} \text{ x } M_4$$

30 DEGREES CASE:



$$M_5 = \frac{\Omega_1}{\Omega_5} \times M_1 = \frac{C_{15}-C_{50}}{C_{15}-C_{10}} \times M_1 = \frac{47{,}96\ mm}{16{,}12\ mm} \times M_1 = \mathbf{2{,}9748} \times M_1$$

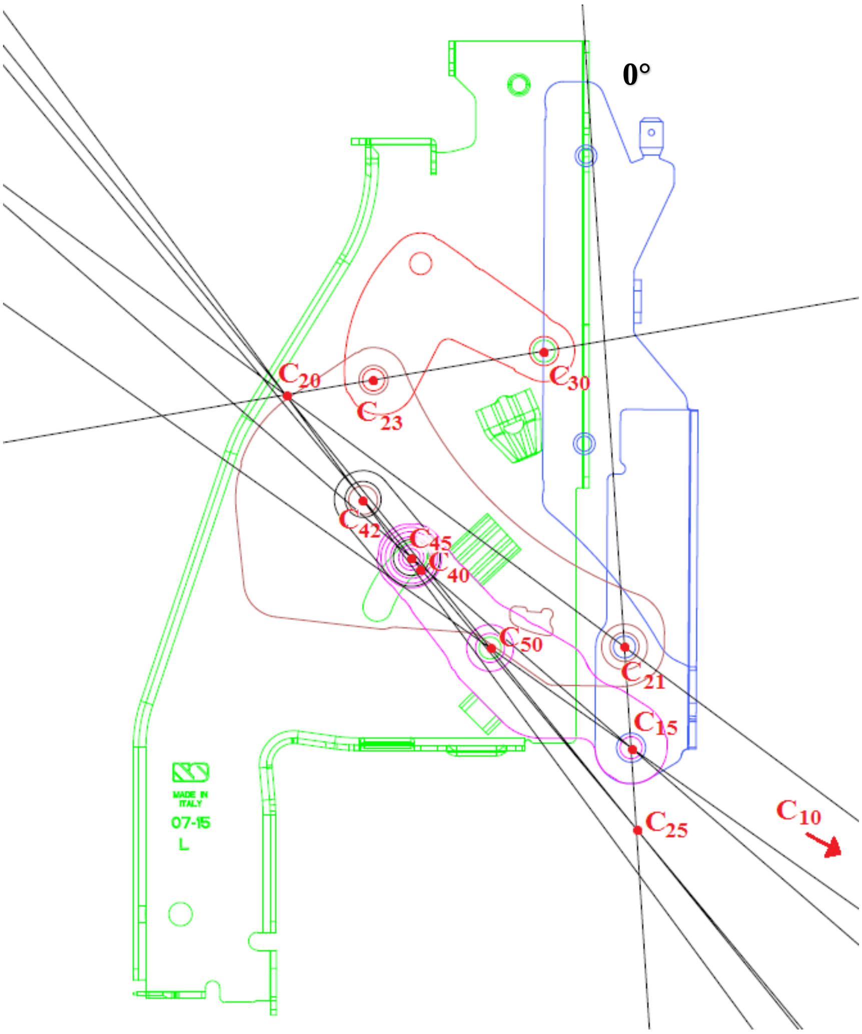
$$M_2 = \frac{\Omega_1}{\Omega_2} \times M_1 = \frac{C_{12}-C_{20}}{C_{12}-C_{10}} \times M_1 = \frac{81{,}60\ mm}{28{,}76\ mm} \times M_1 = \mathbf{2{,}8368} \times M_1$$

$$M_2 = \frac{\Omega_5}{\Omega_2} \times M_5 = \frac{C_{25}-C_{20}}{C_{25}-C_{50}} \times M_5 = \frac{2228{,}3\ mm}{2336{,}69\ mm} \times M_5 = \mathbf{0{,}9536} \times M_5$$

$$M_4 = \frac{\Omega_2}{\Omega_4} \times M_2 = \frac{C_{24}-C_{40}}{C_{24}-C_{20}} \times M_2 = \frac{23{,}1978\ mm}{93{,}675\ mm} \times M_2 = \mathbf{0{,}2476} \times M_2$$

$$M_5 = \frac{\Omega_4}{\Omega_5} \times M_4 = \frac{C_{45}-C_{50}}{C_{45}-C_{40}} \times M_4 = \frac{33{,}302\ mm}{7{,}8638\ mm} \times M_4 = \mathbf{4{,}2348} \times M_4$$

0 DEGREES CASE:



$$M_5 = \frac{\Omega_1}{\Omega_5} \times M_1 = \frac{C_{15}-C_{50}}{C_{15}-C_{10}} \times M_1 = \frac{47{,}96\ mm}{904{,}8\ mm} \times M_1 = \mathbf{0{,}0530} \times M_1$$

$$M_2 = \frac{\Omega_1}{\Omega_2} \times M_1 = \frac{C_{12}-C_{20}}{C_{12}-C_{10}} \times M_1 = \frac{116{,}87\ mm}{922{,}83\ mm} \times M_1 = \mathbf{0{,}1266} \times M_1$$

$$M_2 = \frac{\Omega_5}{\Omega_2} \times M_5 = \frac{C_{25}-C_{20}}{C_{25}-C_{50}} \times M_5 = \frac{155{,}046\ mm}{64{,}662\ mm} \times M_5 = \mathbf{2{,}3978} \times M_5$$

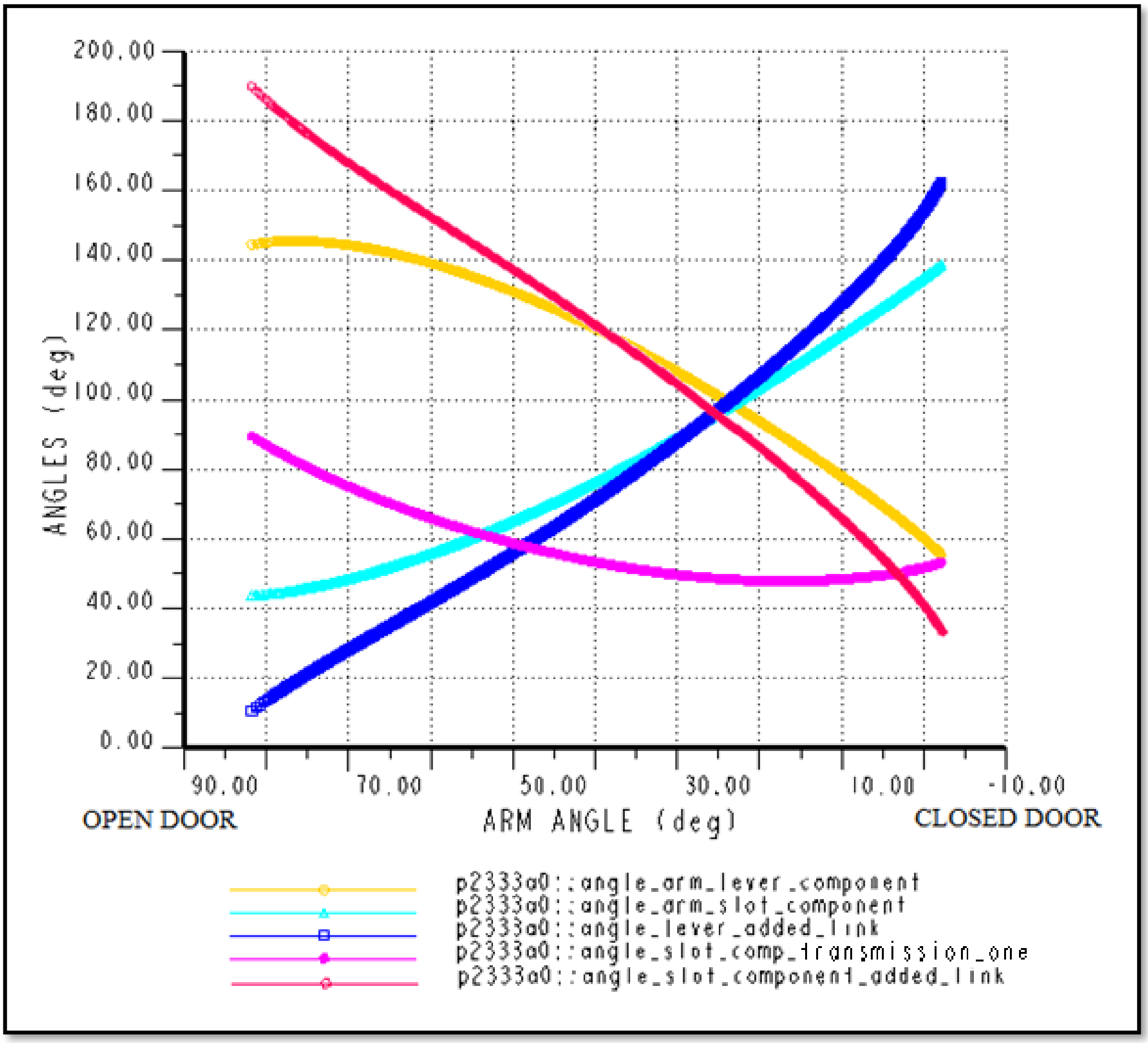
$$M_4 = \frac{\Omega_2}{\Omega_4} \times M_2 = \frac{C_{24}-C_{40}}{C_{24}-C_{20}} \times M_2 = \frac{27{,}648\ mm}{36{,}177\ mm} \times M_2 = \mathbf{0{,}7642} \times M_2$$

$$M_5 = \frac{\Omega_4}{\Omega_5} \times M_4 = \frac{C_{45}-C_{50}}{C_{45}-C_{40}} \times M_4 = \frac{33{,}211\ mm}{6{,}9176\ mm} \times M_4 = \mathbf{4{,}8009} \times M_4$$

7.2 SOFTWARE METHOD

Software method consists in performing a kinematic simulation using the Creo Parametric 2.0. By using the tool called “Mechanism” we simulate the positions of the different components.

We firstly define a reference variable, the angle of the door and we make it changing between 0 and 88 degrees. Then, by plotting all the other variables as its function, we find the following graph:



From this result we observe that the angle between added link and the slot component is the one that changes mostly.

Since the work done is expressed as:

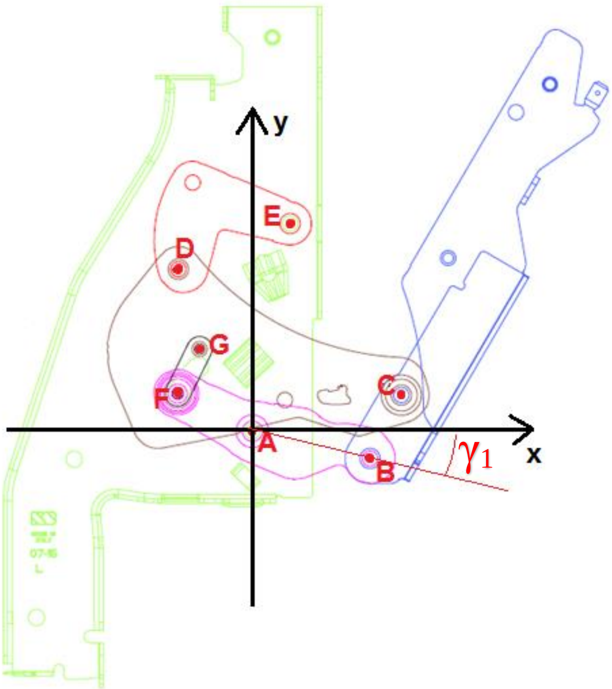
$$\text{Work} = \text{Effort} \times \text{displacement}$$

Having a high displacement means achieving the same work but with less effort. This is the concept of leverage system that we want to exploit in mechanism development.

7.3 NUMERICAL METHOD

A final method is used to validate previous solution by using Matlab. The main goal of this technique is to identify angle behavior between components and, as done for software solution, identify the couple that has the highest displacement.

Starting from the following picture that allows to understand the notation used, we now describe the sections of Matlab code and then we show the results.



1. In the first section the constant mechanism variables are initialized. We also define the input angle (γ_1) that is recursively changed by hand in order to arrange the hinge mechanism in different configurations.
   The program then reconstructs all the possible mechanical patterns by using an iterative procedure. It first finds AB and AF coordinates that just depend on $\gamma_{1.}$ Then it draws a circle centered on F having radius of length FG. It then splits this circle in a large number of equal sections and stores all the coordinates of FG vectors.
   Now from vector sum AG = AF + FG, knowing precisely AF and all the allowable FG coordinates, we get all the possible AG coordinates that again are stored in an appropriate array.
   Till now a first mechanical constraint has been set, we now move to a second one that regards point C.
   We start considering AB vector that depends just on $\gamma_{1,}$ for mechanism geometry point C must necessary lay on a circle centered in B having radius BC, a known value. At the same time point C is forced on the circle centered in G having radius CG, again known, while G coordinates are obtained from previously stored array.
   Therefore we intersect these two circles and get the two point coordinates, that represent vector AC. The command used for the intersection is a Matlab function already implemented in the software:

```
[xc1, yc1, xc2, yc2] = intcirc(AG(1), AG(2), AB(1), AB(2), lGC, lBC);
```

The two intersection coordinates are then stored in two arrays called AC1 and AC2.
We highlight that the number of solutions till here is quite large since for each AG a couple of AC coordinates is found.
To complete the analysis we need to identify an additional constraint that restricts the number of solutions: the link ED. Indeed the next step consists in getting the length of link ED from GC knowledge, then compare it to the length obtained from the cad model. The configurations that equal these two results are the only feasible. But let see more in detail the reasoning.
Slot component is a rigid body with known dimensions, thus the angle DGC is known.
We take GC vector from GC = AC- AG relation, and divide it by its modulus, in this way we obtain its unit vector. We rotate it by angle DCG making use of the rotation matrix. Further we multiply for GC modulus, that's known since it's a rigid body, obtaining GD.
Now ED modulus is known from the rigid body geometry. But at the same time the vector properties allow to define it as ED = AG + GD – AE. Therefore a correct solution should lead to zero difference between the ED obtained from link dimension and from the computed calculations. We identify this difference as an error and we go on by saving the overall geometry that leads to a smaller one.
We highlight that the expected allowable configurations are two since we found two AC arrays.

```
rotm = [cos(aCGD) -sin(aCGD);
            sin(aCGD) cos(aCGD)];

    % Sol1
    GC1 = AC1 - AG;
    GD1 = (lGD/lGC) * rotm * GC1;
    err1 = abs(norm(AG+GD1-AE)-lED);

    % Sol2
    GC2 = AC2 - AG;
    GD2 = (lGD/lGC) * rotm * GC2;
    err2 = abs(norm(AG+GD2-AE)-lED);
```

Finally error is compared to a small value that is chosen arbitrary: if it is smaller, configuration results are saved into an array, otherwise not.

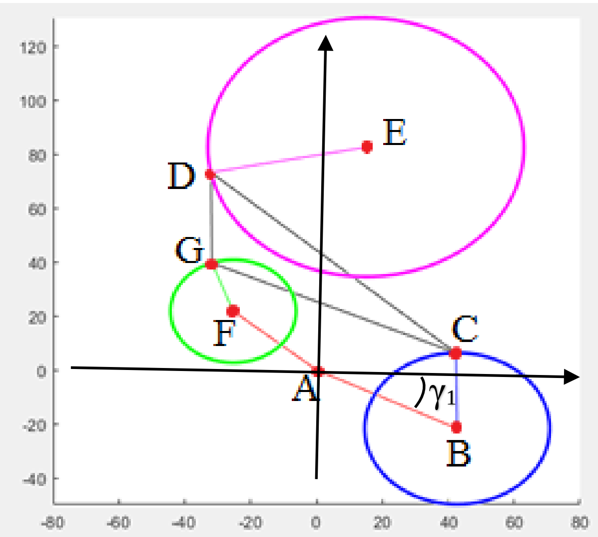
```
  if err1<tol
          res(cnt).AG = AG;
          res(cnt).AC = AC1;
          res(cnt).AD = AD1;
          cnt=cnt+1;
      end
      if err2<tol
          res(cnt).AG = AG;
          res(cnt).AC = AC2;
          res(cnt).AD = AD2;
          cnt=cnt+1;
      end
```

For better understanding we explain that *res* is the vector of feasible solutions while *cnt* is a counter that increases by one for each feasible configuration.

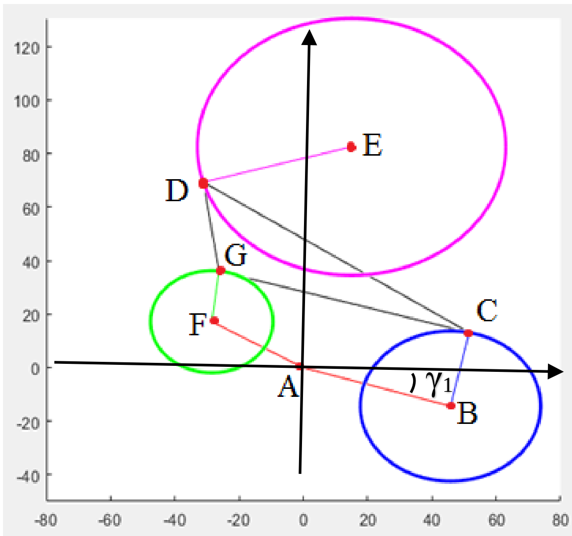
2. In this section angles among links are computed for each feasible solution since we know all vectors coordinates in x and y. For doing this we simply make use of the Matlab command *atan2*: it returns the inverse tangent of x and y that must be real.
   Angles obtained are than saved as variables.
3. In this section all the feasible configurations are plotted in a graph. Different color lines represent the links while circles show the evaluated joint positions.
   By plotting the solutions we see that possible mechanism configurations are more than one. This conclusion is justified by the fact that FG link is completely free to rotate around G. Instead in real case it's constrained by the slot on F joint and allowable configuration is only one.
4. In this last section we plot one by one the feasible configurations allowing to understand better links position. By knowing a little bit the hinge geometry it's immediate to understand the real mechanism solution neglecting the wrong alternative that's due to FG lack of constraint.

Next few resultant feasible configurations are plotted:

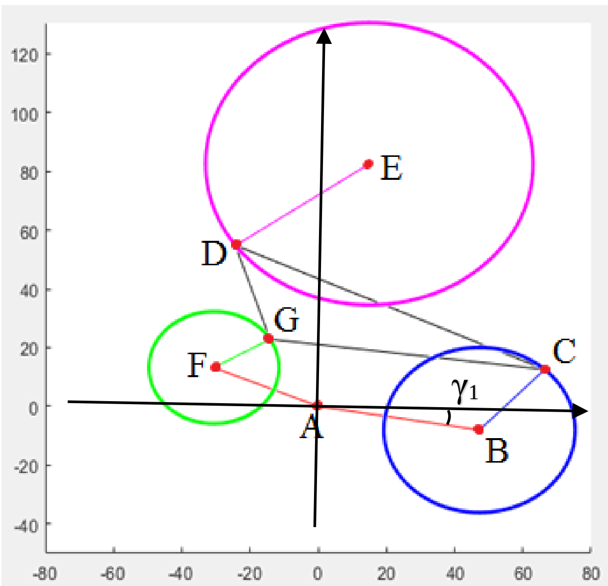
DOOR ANGLE: 0°



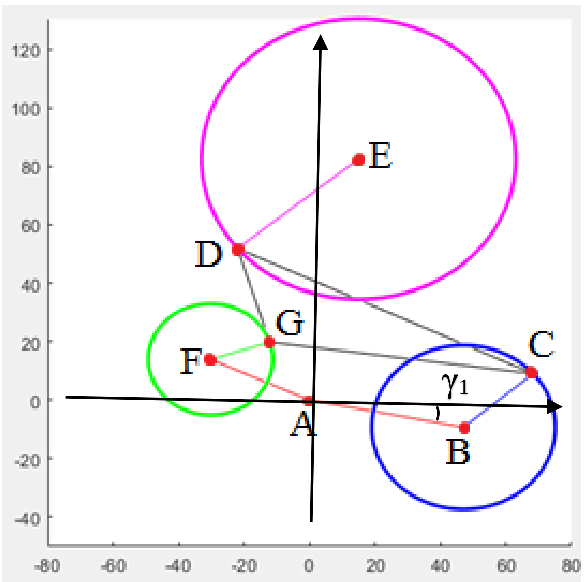
DOOR ANGLE: 20°



DOOR ANGLE: 40°



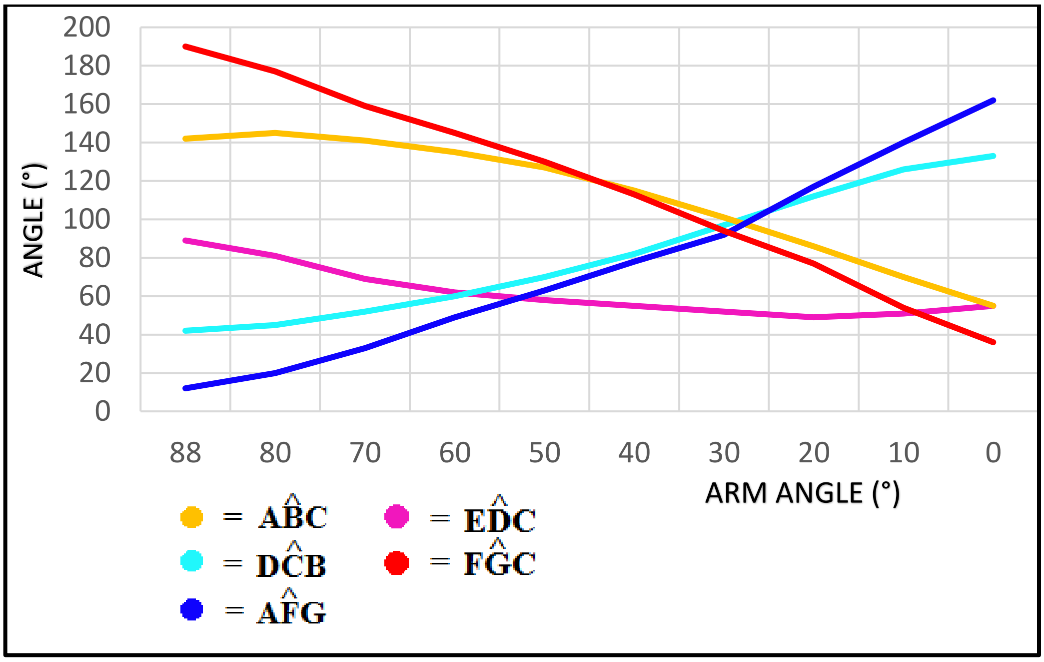
DOOR ANGLE: 60°



Finally a table reporting all main angles displacements:

DOOR ANGLE (°)	ANGLE $E\hat{D}C$ (°)	ANGLE $D\hat{C}B$ (°)	ANGLE $A\hat{B}C$ (°)	ANGLE $A\hat{F}G$ (°)	ANGLE $F\hat{G}C$ (°)
0	55	133	55	162	36
10	51	126	70	140	54
20	49	112	86	117	77
30	52	97	101	92	94
40	55	82	115	78	113
50	58	70	127	63	130
60	62	60	135	49	145
70	69	52	141	33	159
80	81	45	145	20	177
88	89	42	142	12	190

This analysis confirms results of previous software analysis. The angle between added link and the slot component is the one that changes mostly allowing to minimize overall effort.



7.4 RESULTS OF KINEMATIC ANALYSIS

The goal of these analysis is to find the best couple of components on which applying the actuator force. A good couple is the one that maintains a constant transmission ratio with an high angle displacement. In this way we can easily define the motion law of the motor at a fixed gear ratio minimizing the required effort.

Looking at our results we see that two couples of elements maintain an almost constant transmission ratio and a quite wide relative motion: the *added link-lever* and the *added link- slot component*. But applying a force between the elements of the first couple would be complicated. So we will focus on the second couple.

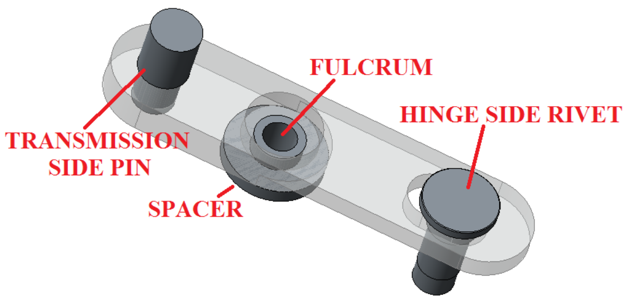
8 DESIGN CONCEPT

Developing a realizable and efficient mechanism is very complicated. The overall idea is quite clear since it's given by the previous analysis: we need to apply a force or a torque between the added link and the slot component. But the main trouble is to find a clever design that leads to simple and compact components. Indeed the dishwasher shape doesn't allow to locate a motor near the hinge since the available space is just 20 mm thick. The only place for the actuator is located in the devices area. Thus it's also required a transmission system that transfers the motion from the actuator area to the hinge.

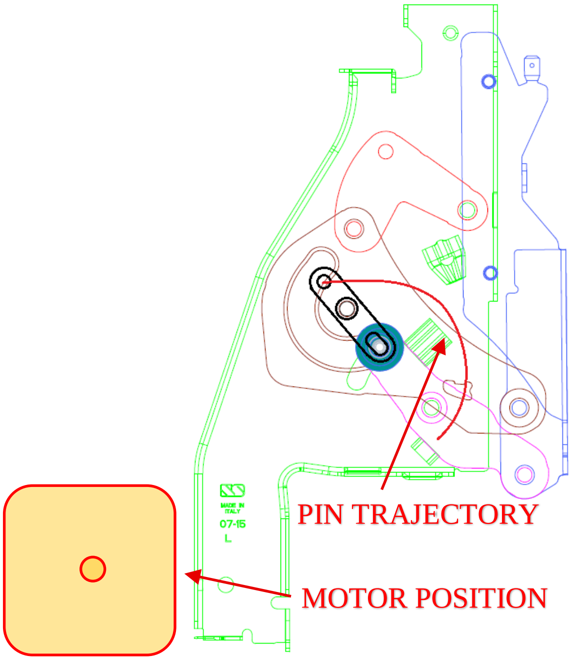
For a first working solution we can take inspiration from Oldham joint principle. From a theorical point of view this joint is used as a coupling device for connecting two shafts. Its main purpose is to transmit power allowing a small misalignment. It's composed of three discs, one coupled to the input, one coupled to the output and a middle disc that it's joined with them using a tongue and groove mechanism. The tongue and groove on one side is perpendicular to the tongue and groove on the other side.

The starting point consists in modifying the added-link design. We increase its length and transform it into an advantageous lever having these 3 junctions:

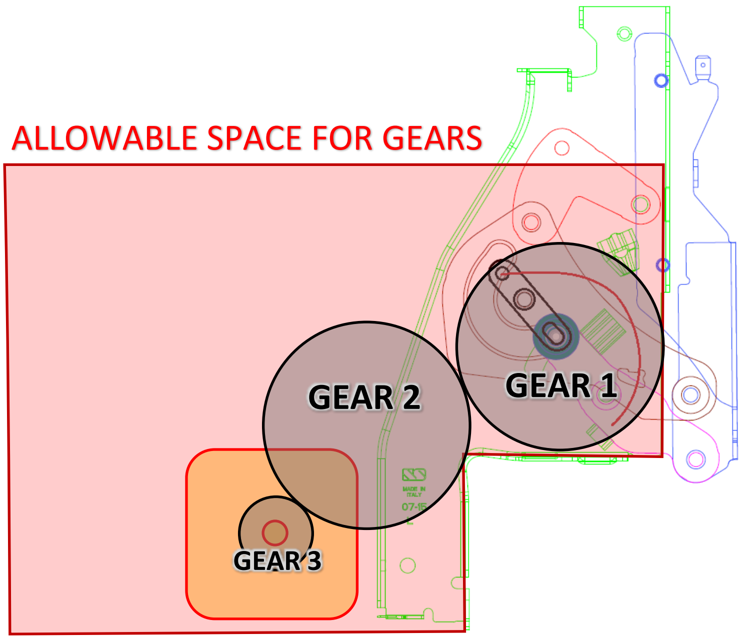
- Fulcrum: located on the slot component, is jointed to it using a pin connection. In order to decrease friction and increase its joint robustness we added a double diameter spacer in between;
- Hinge side: reaches the stopper and the lever component. It's link to both them using a double diameter rivet that allows to maintain the exact thickness among them. During the previous analysis we assumed that the slot was perfectly circular. Actually this is not true since in the last 10 degrees of opening and closing, the slot has an irregular shape. In order to compensate this deviation, we implement a slot also in this new component;
- Transmission side: receives the input motion from the transmission. We use a third pin-slot coupling to make this power exchange, the pin is located on current component while the slot is obtained on the transmission element. The position of this last pin is essential since it defines the leverage ratio. We put it at 15 mm from the fulcrum since it's a good compromise among increasing the ratio and maintain small incumbrance.



So the overall mechanism just developed allows to open and close the arm of the hinge by moving the transmission side pin on the following trajectory:



Going further we develop the transmission system considering the motor position. The first simplest approach deals with using gears. Till now we have no requirements from the overall gear ratio since the motor has not being identified yet. Thus our only constraints regard allowable space and components simplicity. Following these considerations we design three different gears; two gears only won't allow to reach all trajectory points respecting space constraints. In the following picture the gear schematic is presented.



In order to maintain a simple design, we implement a teeth modulus one for all the gears. By fixing their primitive radius we get the number of teeth and the overall gear ratio. For setting the radius we look at the pin trajectory making sure that it always stays inside the gear circle. So the two biggest gears have a primitive diameter of 95mm, thus also 95 teeth. While the last one has a diameter of 24 mm, thus it's 24 teeth.

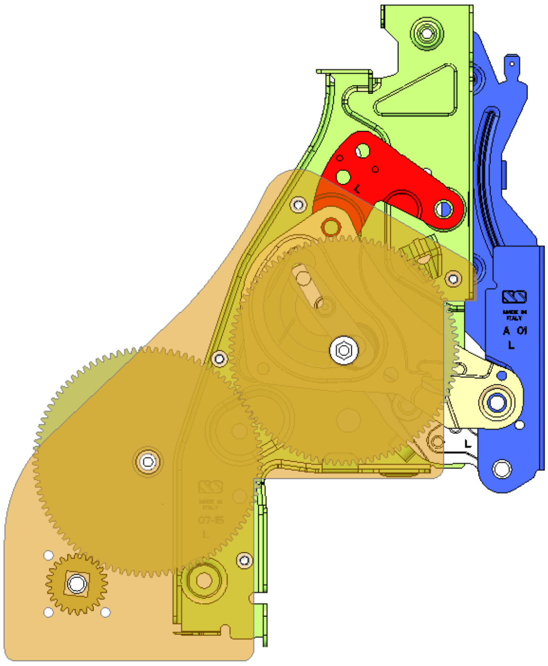
We adopt the same radius for two gears since in this way the manufacturing process is repetitive and fast. The only difference among them is that the gear linked to the transmission pin features a slot, the other not.

In order to draw them we use AutoCAD software that has a proper library to easily design cylindrical gear.

The overall gear ratio is given by:

$$\tau = \frac{T_{gear1}}{T_{gear2}} \text{ x } \frac{T_{gear2}}{T_{gear3}} = \frac{95}{95} \text{ x } \frac{95}{24} = 3.958$$

Finally in order to fix the gears to the hinge we add a layer linked to the box thanks to 4 rivets; each gear is then linked to it by using bushing and screw in order to reduce friction and noises. The obtained model is the following one:



9 SIMULATIONS:

After a roughly model development, we move to simulation environment in order to optimize the set up and estimate the forces. For simulations purposes we use the toolkit Simscape of Matlab 2019a. This simulation environment allows to make dynamic simulation using a block programming language. Further, we install a toolkit in Matlab and on Creo Parametric 2.0, our 3D cad software, that allows to automatically import in Simscape the designed mechanism.

This toolkit is very useful and permits to save a lot of time, all the components are quickly exported keeping their shapes and positions, anyway it requires to redefine all the joints.

So we export all the hinge in the simulation environment. For simplicity and for making faster simulation we just consider one dishwasher side. We put all the door weight on this side, we will eventually divide by two the loads that must be split between the two hinges. For example in the following pages we will describe 2 different simulations, the first one is focused on finding right spring of the hinge, so in this case the obtained load will be divided by two since there’s a spring for each hinge. Instead in a second simulation we will focus on motor and gears load, in this case we will not divide by two since the actuated system is put just on one hinge.

Another clarification concerns the hinge friction. We have seen that the stopper is used to make a high interference with the slot component. In this way the hinge balances many doors that have a weight ranging between 4 kg. Actually in our application the door weight is assumed to be fixed and equal to 11 kg. In order to minimize the motor effort we must remove all possible friction elements included the one generated by the stopper. Thus also in the simulations we will consider almost zero friction among the components.

9.1 SPRING SIMULATION:

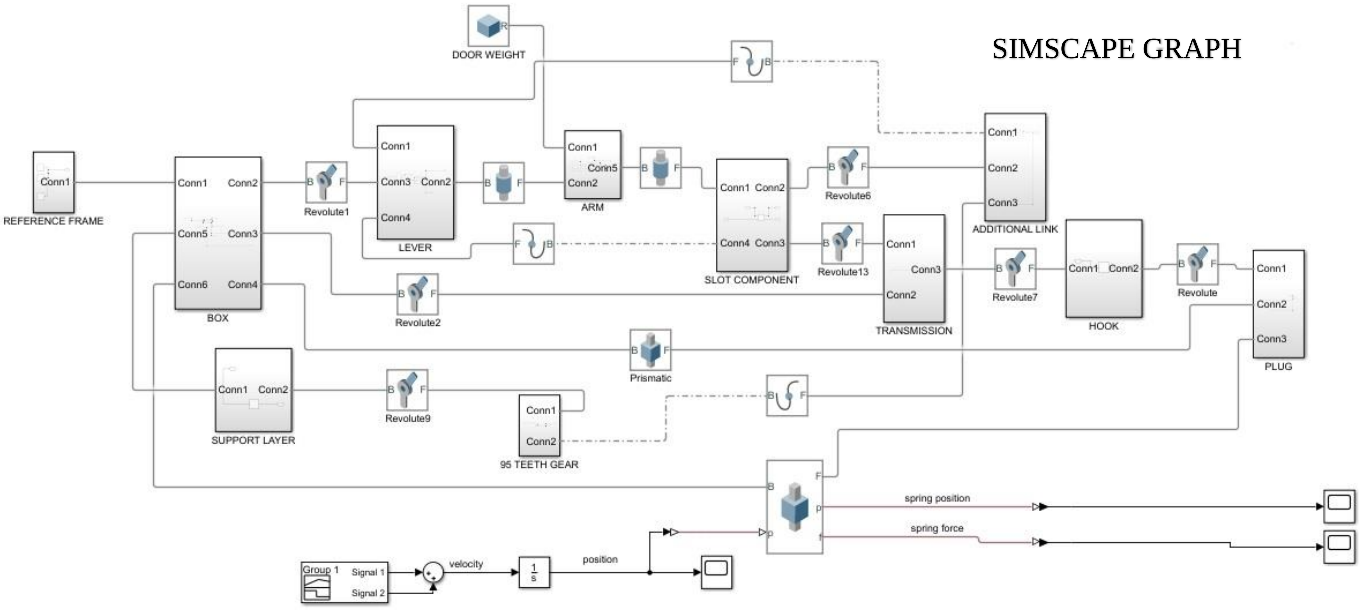
In order to minimize motor load, we must optimize the spring. Its goal is to store the potential energy of the door during the opening phase and release it during the closing phase.

For computing this kind of simulation we start from the model of the hinge. It’s made by two kind of files, one is a matlab file which defines all components properties (physical and graphical). The other is a Simulink file which represents all the parts as blocks without connecting them. We start by defining each weight and each center of mass in the matlab file. We then move to the Simulink file and we connect all the links with the appropriate joints.

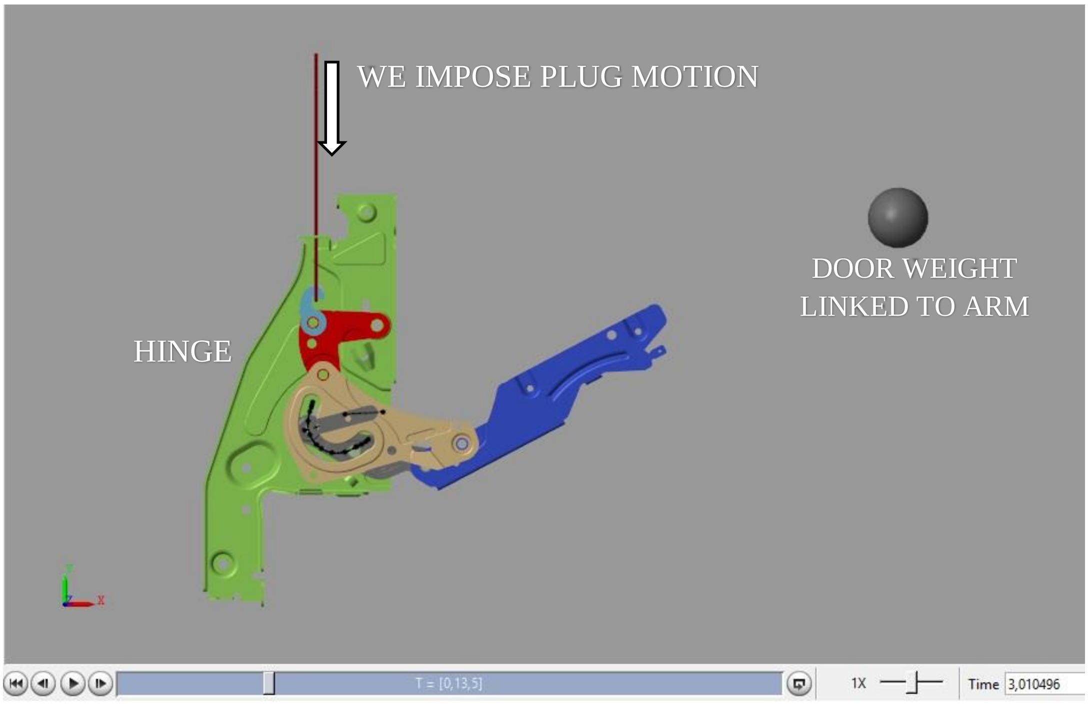
By running the Simulink file we are able to check the model validity. A simulation window should appear revealing all the assembly and the connections between the pieces. If everything matches we can go ahead applying a law of motion to the system.

Having the model we decide which component has to be moved during the simulation. In our case we want to measure the force that acts on the plug. Avoiding frictions, this is mainly due to door weight.

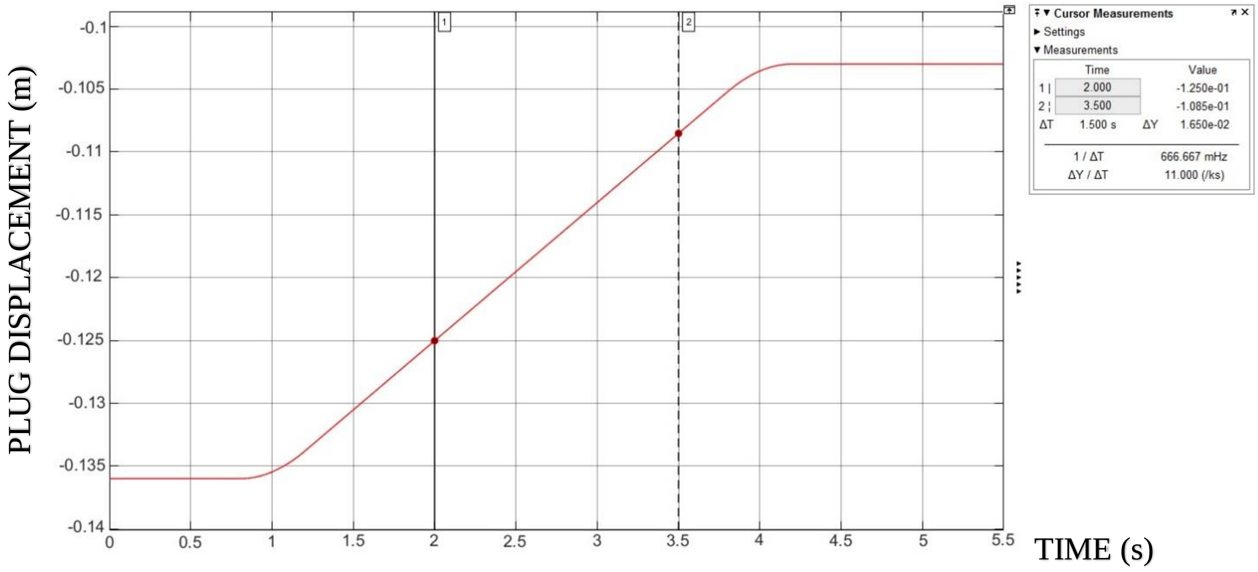
To measure it, we move the plug down sensing meanwhile the required effort. From program point of view, making this test is quite simple, we need to focus on the cylindrical joint that connects the plug with the box. Here we impose a proper law of motion that makes the component going down, this also leads the door to open; meanwhile we measure the vertical force that’s achieved in the same joint. The results are showed in the following graphs:

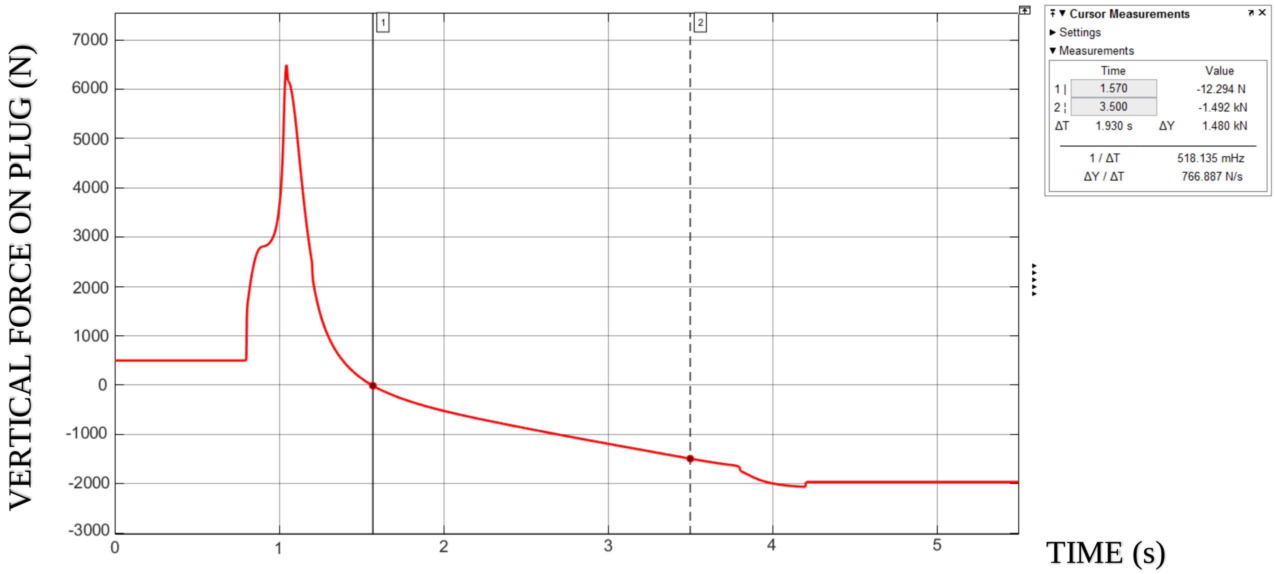


SIMSCAPE GRAPH
DOOR WEIGHT
REFERENCE FRAME
BOX
Revolute1
LEVER
ARM
SLOT COMPONENT
Revolute6
Revolute13
ADDITIONAL LINK
TRANSMISSION
Revolute7
HOOK
Revolute
PLUG
Revolute2
Prismatic
SUPPORT LAYER
Revolute9
95 TEETH GEAR
spring position
spring force
velocity
position
Group 1
Signal 1
Signal 2

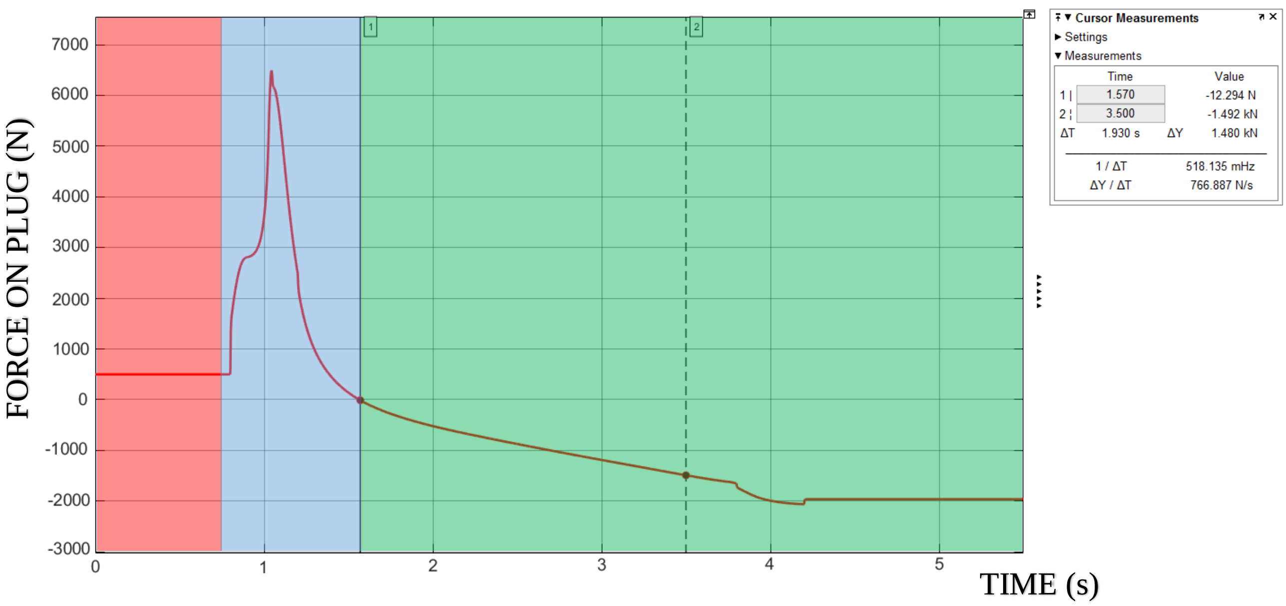


WE IMPOSE PLUG MOTION
HINGE
DOOR WEIGHT
LINKED TO ARM
T = [0,13,5]
1X
Time 3,010496





From this last graph we obtain a force curve that perfectly balances the door:



We identify three parts:

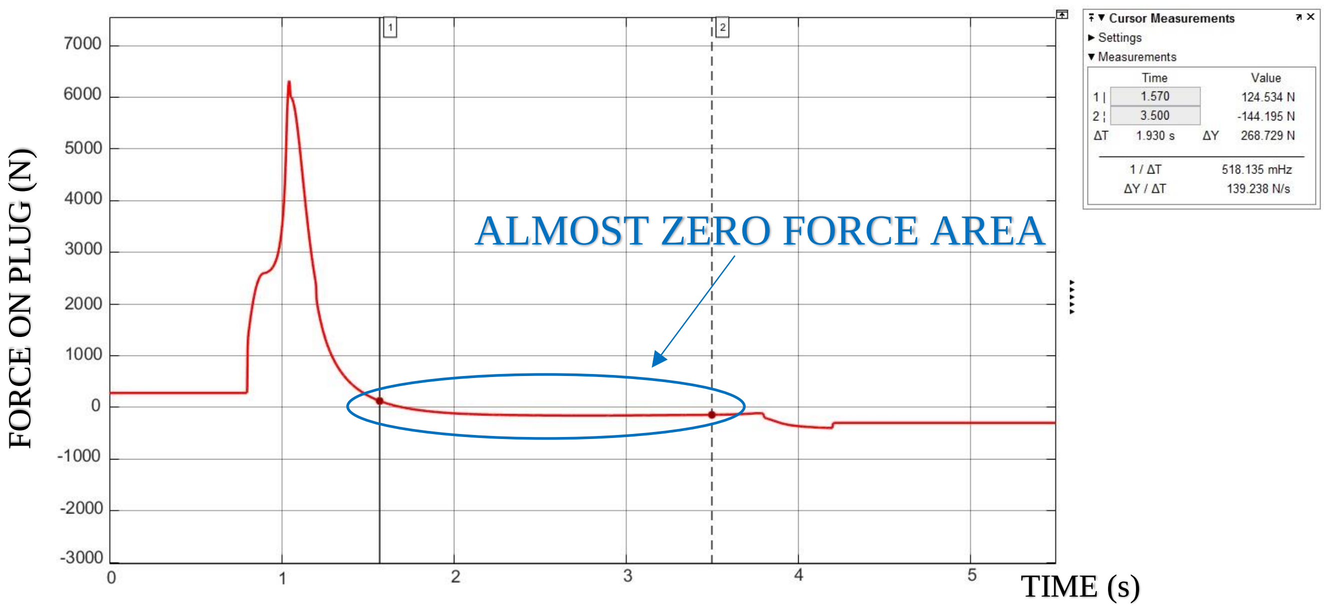
1. RED ZONE: till 0.8 seconds the force is constant and equal to 498 N. This represents the effort that the plug receives from the door weight at 0 degrees. Due to hinge geometry, the door tends to push against the hinge trying to reach negative angle. In real case the end strokes of the dishwasher delete this force, so by mounting hinges and putting the door at zero degrees, we see that it doesn't move;
2. BLUE ZONE: from 0.8 seconds till 1.57 seconds the door tends to close itself due to the variable fulcrum behavior. Till 27 degrees the door center of gravity is moved up, thus instead of releasing energy the door requires it. This is the reason why the spring has a positive modulus;
3. GREEN ZONE: after 1.57 seconds the door behavior becomes more linear and the door weight tends to open it, the spring force is negative since it's opposite to door action;

In order to define the spring we focus on the green zone and we identify two total force values:

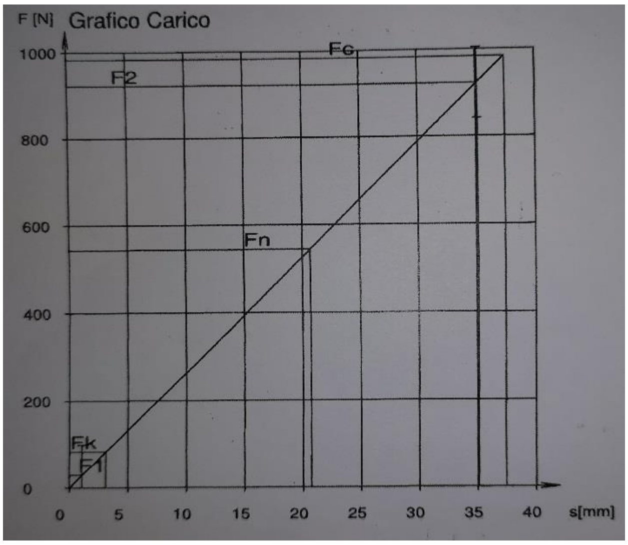
TIME (s)	ABSOLUTE POSITION (mm)	DISPLACEMENT [136mm - ABS. POS.] (mm)	TOTAL FORCE (N)	HINGE FORCE (N)
2	125	11	-522	-261
3.5	108	28	-1492	-746

Since the hinges are two we need to split total force among them in equal parts. In this way we get the single hinge forces.

Finally in order to check the functionality of the spring we did an additional simulation. We put it in position and simulate again the door motion. The obtained plug force is plotted in the following:



Now the required effort for opening and closing the door is minimized. From these results we are able to order the springs from our supplier. We receive a sample with the following characteristic:



L = length   s = displacement   F = force

L0 : 138.10 mm
L1 : 137.00 mm → s1 : 1.10 mm → F1 : 28.89 N
L2 : 103.00 mm → s2 : 35.1 mm → F2 : 922.0 N
Ln : 117.40 mm → sn : 20.7 mm → Fn : 543.7 N
Lc : 100.67 mm → sc : 37.4 mm → Fc : 983.2 N

L3 : 125.00 mm → s3 : 13.1 mm → F3 : 305 N
L4: 108.00 mm → s4 : 30.1 mm → F4 : 790 N

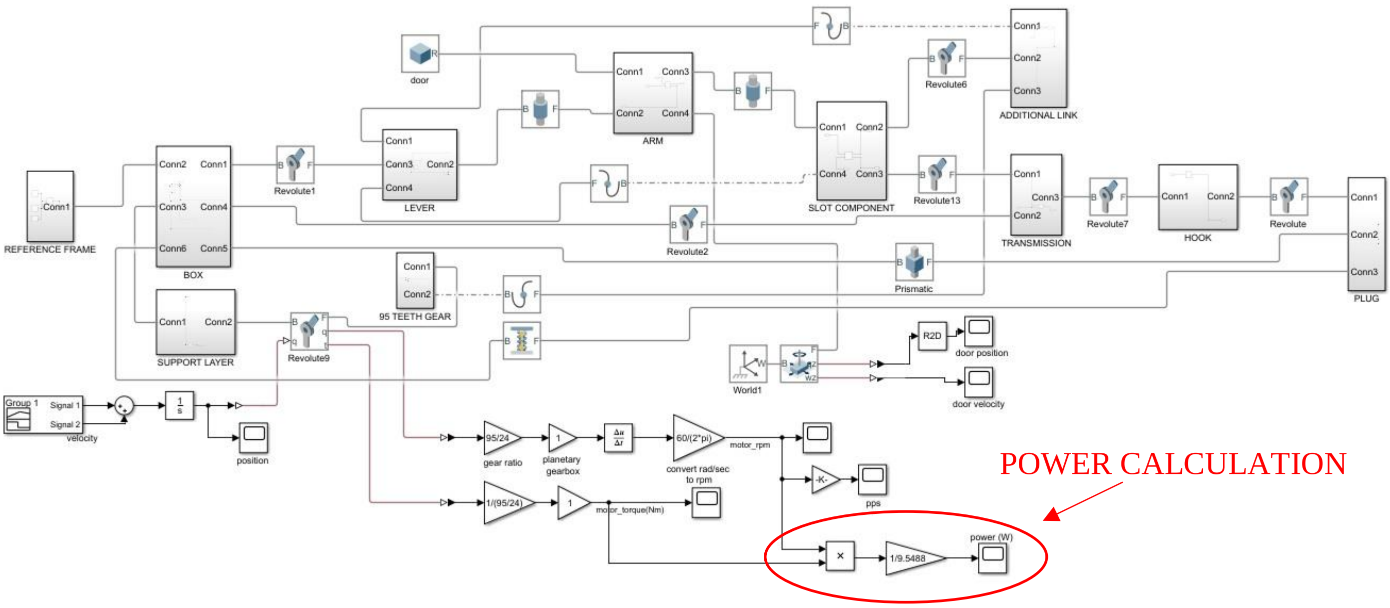
VALUES ARE A LITTLE BIT HIGH BUT IN REAL CASE WE HAVE A LITTLE BIT OF FRICTIONS.

9.2 POWER SIMULATION:

After defining the hinge characteristics, we focus on our particular application. In order to size the motor we have to know the power for moving the system. Again we use the Simscape environment looking for the motor power. We use the previous model but we also add the springs and a little bit of friction in the joints in order to simulate the worst-case scenario that accordingly require the maximum power.

Knowing the power it's possible to select a motor from the datasheets. Then by looking at its torque curve we can define the overall gear ratio. We highlight that some gears have already been defined during transmission design. As already shown, their teeth number defines their gear ratio that's equal to 3.958. In our simulation we model them as a simple gain that will divide the required torque and multiply the required speed. In this way we avoid the inertia and the friction introduced by the transmission elements since they are negligible.

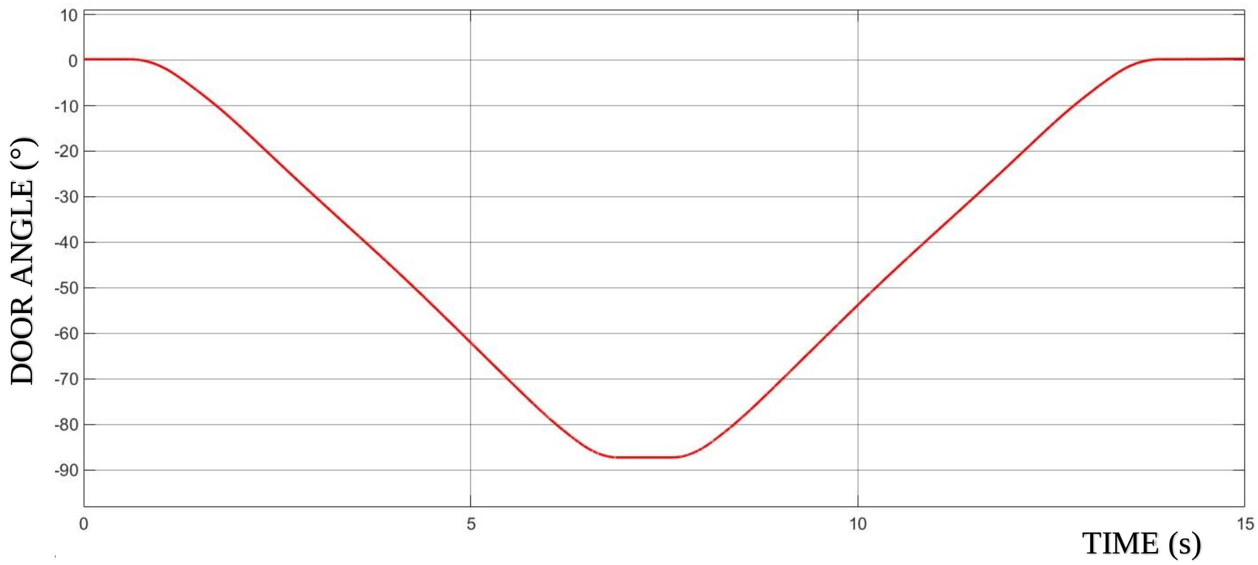
The obtained Simscape model is:

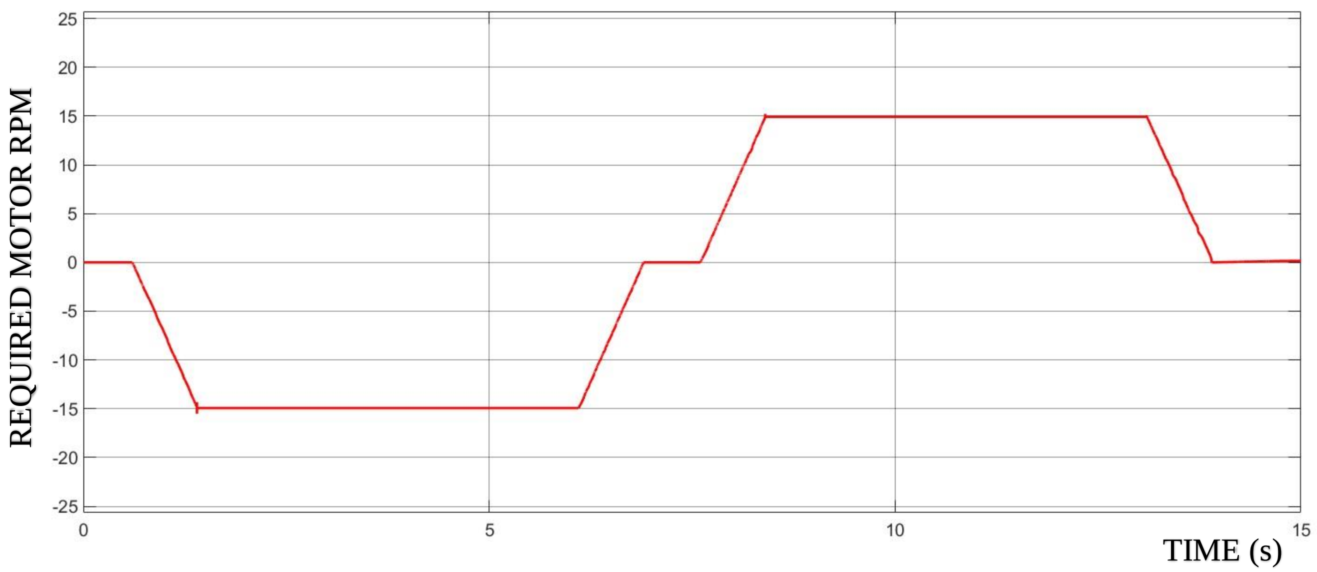


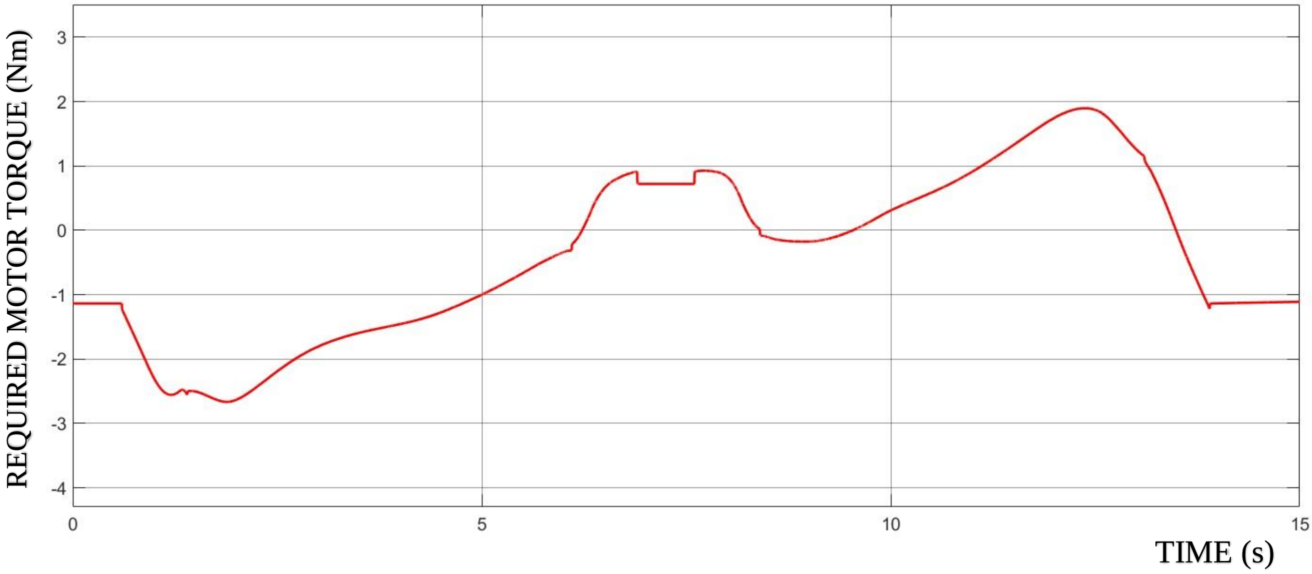
Where the power is obtained from the following relation:

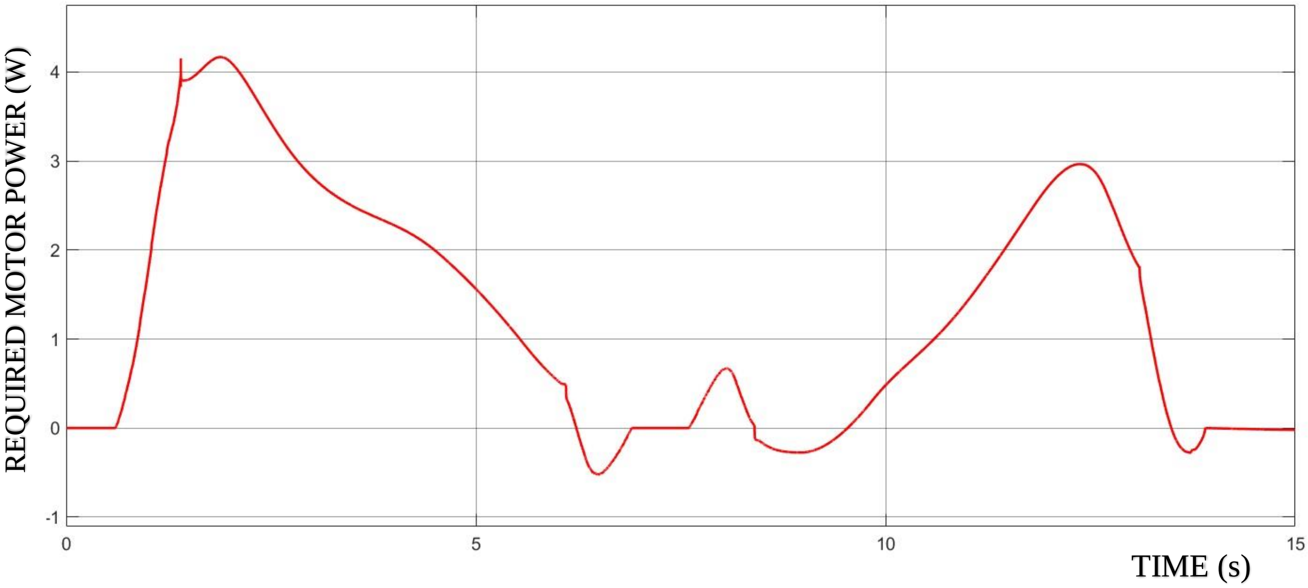
$$\text{Power (W)} = \frac{\text{Torque (Nm) x Speed (rpm)}}{9.5488}$$

We run the simulation making the plug moving up and down thus opening and closing the door, the following plots are found:









From these results we get the fundamental variables handled by the geared motor output shaft. We start from the power peak that's 4.2 Watt. For safety margin we consider a value of 6 Watt. Instead for torque and speed we consider respectively 3 Nm and 18 rpm values: with respect to the plots results we maintain again a safety margin.

Converting these two values into power we obtain:

$$\text{Power (W)} = \frac{3\ (\text{Nm})\ \text{x}\ 18\ (\text{rpm})}{9.5488} = 5.655\ \text{W} < 6\ \text{W}$$

Their product is lower than the considered power, so they respect the constrains.

A brief look to these parameters allows to understand that an additional reduction ratio is fundamental. Actually just few motors reach such an high torque at a low speed, but are very expensive and bulky (ex. frameless motors).

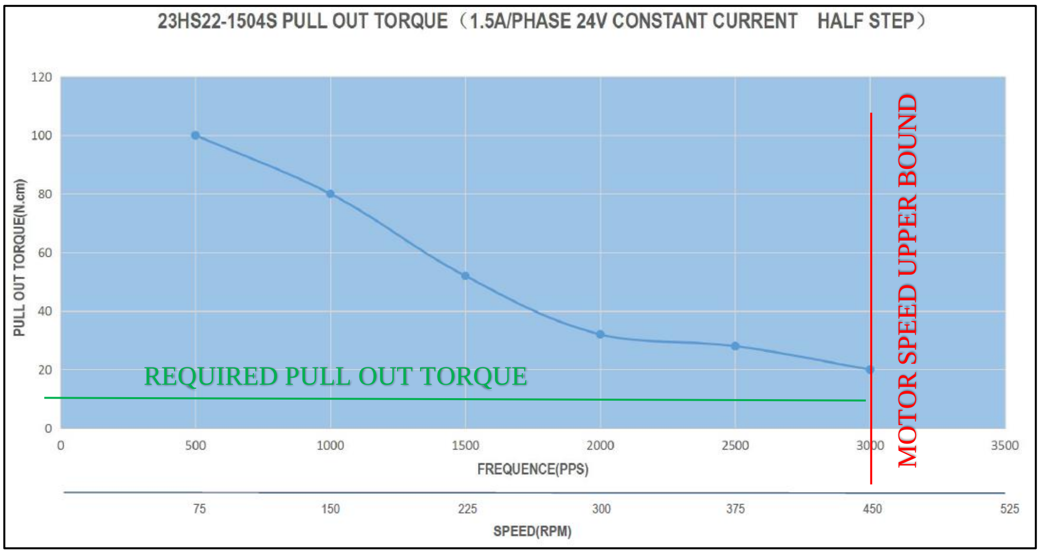
Starting from these parameters we identify the right one and an appropriate reduction ratio.

10 MOTOR SELECTION:

We firstly choose the kind of motor. Thinking about requirements: we don't need a feedback loop, we need something very cheap that's able to generate a high torque with respect to its dimension.

Another important constraint is power supply: we would like to use common AC voltage at 220V and 50 Hz so that motor is easily connected to house plug. Even if few AC configurations are feasible, they're very expensive. It's much cheaper to use a stepper motor supplied by a rectifier. So we focused on this kind of actuators. Among all the possible configurations we decide to adopt the most common and commercial one. Therefore we identify the Nema 23 family that shares the same frame of 56 mm. Then we analyze one after one the Nema 23 motors looking for a valuable torque curve.

We find a stepper motor which identity code is 23HS22-1504S and which torque characteristic is:



During the simulations we use 6 seconds time interval for closing and opening. In the initial requirement we find an interval of 6-8 seconds for each operation so we consider actual rpm value as an upper bound.

For computing calculations we need to fix a certain gear ratio that can be obtained from the following formula:

$$\text{Gear ratio } (\tau) = \frac{\text{Motor speed upper bound (rpm)}}{\text{simulation speed (rpm)}} = \frac{450 \text{ rpm}}{18 \text{ rpm}} = \mathbf{25}$$

Where the motor speed upper bound is obtained from the datasheet.
Now we check if the toque generated by the motor is enough:

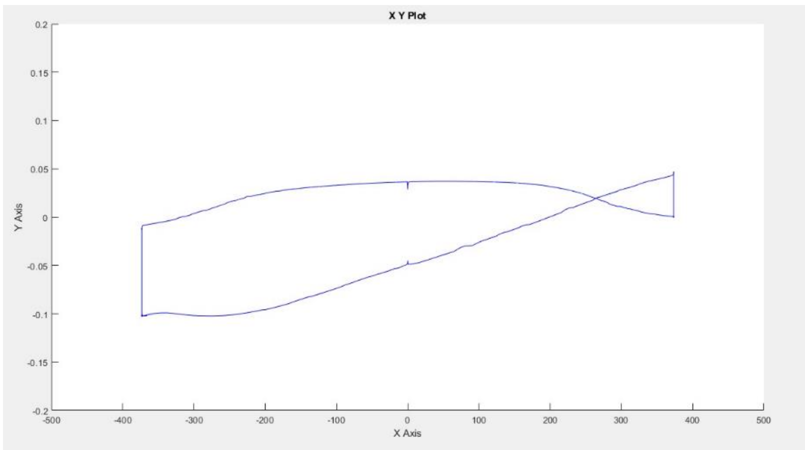
$$\text{Required pull out torque (Nm)} = \frac{\text{Motor torque upper bound (Nm)}}{\text{Gear ratio}} = \frac{3 \text{ Nm}}{25} = \mathbf{0.12\ Nm}$$

Again the motor torque upper bound is obtained from datasheet.

So, just from these rough calculations, we understand that the selected Nema 23 motor supplies the requirements.
In order to be sure of the results we develop a more detailed simulation. Its goal is to show that the motor characteristics are sufficient during all door motion. For doing this we need to simulate and plot the minimal motor torque curve as a function of the required speed. From this simulation we get a closed curve that has the same X and Y axis of the datasheet one. So we just need to overlap and compare the two plots. We check that the obtained closed curve lays on the bottom side of the datasheet one. If this condition is respected than the motor and the gear ratio work well. If not, we could try to change the gear ratio until the condition is respected, otherwise we need to change the motor.

So we update the Simulink model by inserting the 25 gear ratio and plot the results obtaining:

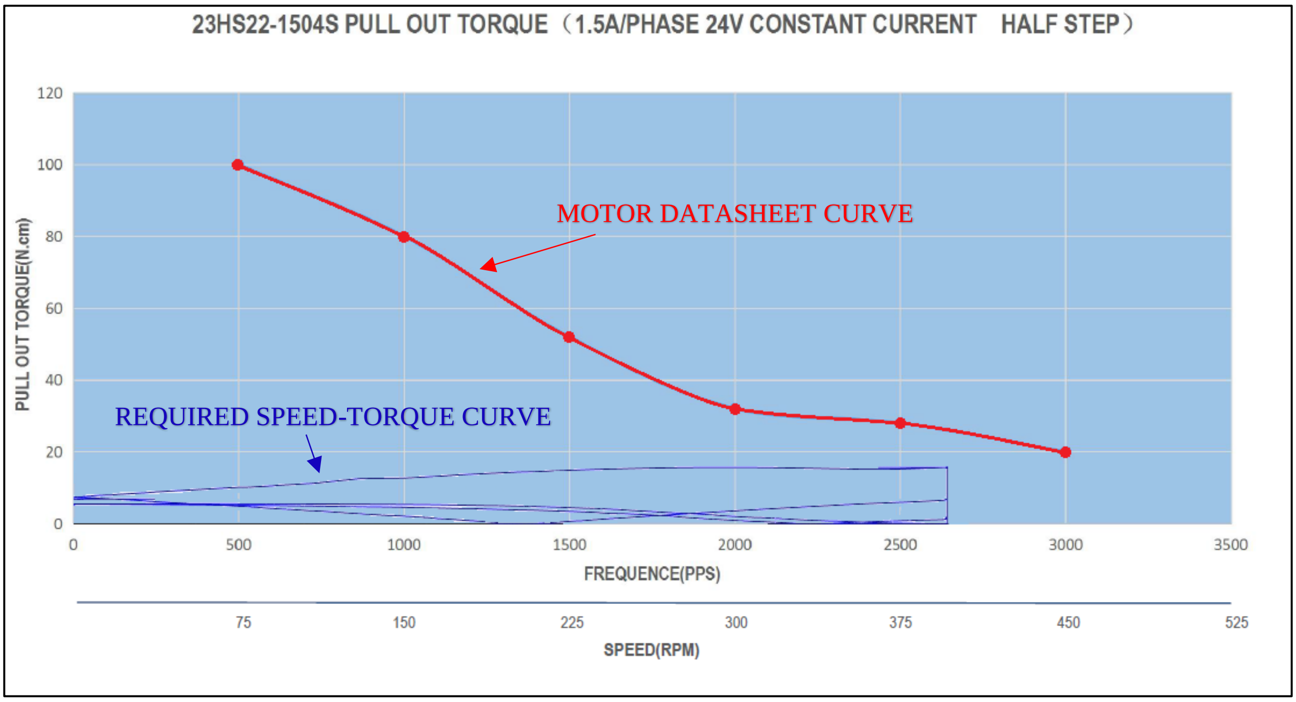
REQUIRED MOTOR TORQUE (Nm)



REQUIRED MOTOR SPEED (rpm)

Now negative velocity and negative torque are transformed into positive, we are just interested in absolute values.

The obtained curve is fit and merged into the datasheet curve. Final results are compared:



As we can see from the plot, the motor characteristic is far above the requirement. The chosen motor and gear ratio fit well.

Here we specify few electric characteristics of the motor from the datasheet:

Bipolar/Unipolar	Bipolar
Holding Torque (Nm)	1.16
Inductance (mH)	13
Phase Resistance (ohm)	3.6
Rated Current (A)	1.5
Step Angle (°)	1.8

11 GEARBOX:

11.1 PLANETARY LAYOUT:

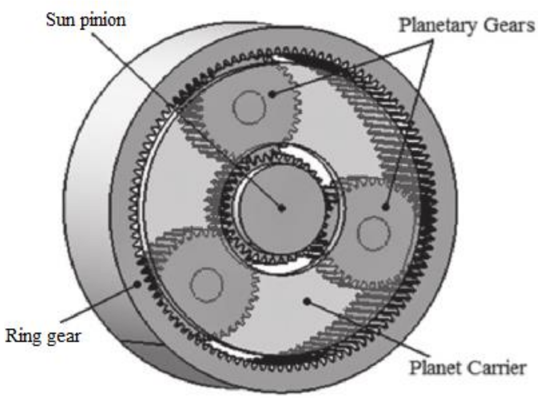
We have already selected the motor for the application. Doing this we have also identified the required gear ratio that's 25. Now we need to focus on this value for the gearbox design.

Since the space is limited, the forces are quite high and the noise must be unremarkable, we decide to adopt a planetary gearbox. Looking on website or asking to our suppliers we find that a 25 ratio planetary gearbox cost at least 100 euros. This solution is too expensive and we can't afford it.

Thus we design and print a very cheap planetary gearbox having 2 internal stages.

Let's now briefly introduce how they work. Each gear train can be divided into four main constituents:

1. RING GEAR: it also represents the housing of the reducer and integrates internal teeth. In the majority of the cases this component is fixed to the frame.
2. SUN PINION: located in the center of the ring gear it's also coaxially arranged with respect to the output. It's attached to a clamping system in order to provide the mechanical connection to the motor shaft.
3. PLANETARY CARRIER: it's main function is to sustain the planetary gears avoiding friction and assuring alignment between components. It also represents the output shaft of the gearbox.
4. PLANETARY GEARS: are mounted on the carrier and roll between the sun pinion and the ring gear. Their number is usually three each stage but also higher. As their number increases, the distribution of the load increases too and therefore the torque that can be transmitted is higher.



A planetary gearbox is extremely efficient, its advantage, compared to a single spur gear, lies in this load distribution. It is therefore possible to transmit high torques at high efficiency with a compact design.

Now by fixing the ring gear the maximum reduction ratio is achieved; the value is given by:

$$\text{Planetary gearbox ratio } (\tau) = 1 + \frac{\text{Ring gear teeth}}{\text{Sun pinion teeth}}$$

So considering our desired ratio of 25 we have:

$$\tau - 1 = 24 = \frac{\text{Ring gear teeth}}{\text{Sun pinion teeth}}$$

Supposing 10 sun pinion teeth we will obtain 240 ring gear teeth. Actually this difference is huge, it would lead to a large gearbox diameter. An alternative solution is to develop a two stages reducer in which the length of the ring gear is increased allowing the location of two consecutive carriers with gears. As a consequence the overall planetary ratio is given by the product of each stage ratio. By considering two stages and a total gear ratio of 25, the obtained single stage ratio is:

$$\text{Single stage ratio } (\tau_{st}) = 1 + \frac{\text{Ring gear teeth}}{\text{Sun pinion teeth}} = \sqrt{25} = \mathbf{5}$$

Further:

$$\tau_{st} - 1 = \mathbf{4} = \frac{\text{Ring gear teeth}}{\text{Sun pinion teeth}}$$

Again supposing 10 sun pinion teeth we will obtain 40 ring gear teeth. Now this difference is acceptable, reducer diameter will be small enough.

Now we will mathematically dimension the single stages.

11.2 SIZING:

Let's start from motor output shaft datas. We have already seen that in worst case scenario the motor will face a speed of 450 rpm and a torque equal to 0.12 Nm. Knowing that the first stage ratio is equal to 5 we will obtain an output torque equal to 0.6 Nm (0.12 Nm x 5) and a speed of 90 rpm (450 rpm /5).

We deal with calculating the teeth number of the sun pinion. The minimal number can be obtained from the rack pinion model having a pressure angle of 20° and is:

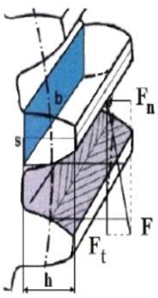
$$Z_{min} = \frac{2}{\sin^2 \theta} \approx 17$$

We adopt this value in order to optimize dimensions.

For defining the modulus we use the Lewis method that is based on tooth flexion. We consider a tooth base dimension that's equal to

$$b = \lambda \times m$$

where m is the tooth modulus.



Now we consider the PA 6.6 30% g.f. material that has a tensile strength of 150 N/mm^2. This material is our target for producing the gearbox since it's very cheap and allows to make component using injection moulded technology.
Thus knowing that the admissible tension is equal to:

$$\sigma_{adm} = \frac{tensile\ strenght}{4}$$

we obtain:

$$\sigma_{adm} = 37.5 \text{ N/mm}^2$$

Now we suppose the gear length (λ) been 20 mm. Thus:

$$\lambda = \frac{b}{m} = 20$$

We start now a procedure that allows to find the modulus. We have to go on with calculations until the result converges.

$$\text{Vp}_{sun} = \frac{\omega_{sun} \times r_{sun}}{2} = \frac{\omega_{sun} \times m \times z_{sun}}{2}$$

Where ω_{sun} is the angular velocity of the sun, such that:

$$\omega_{sun} = 450 \text{ rpm} = \frac{450 \times 2 \times \pi}{60} \text{ rad/sec} = 47.12 \text{ rad/sec}$$

While Vp_{sun} is the peripheral velocity of the sun gear, r_{sun} is the radius and Z_{sun} is the sun teeth number that's equal to $Z_{min} = 17$.

We now use Lewis formula:

$$m = \sqrt[3]{\frac{10.9 \times torque_{eff}}{n°\ plan.gears \times \lambda \times kd \times \text{Vpsun}}} = \sqrt[3]{\frac{10.9 \times 0.18 \times 1000}{3 \times 20 \times kd \times 47.12 \times m \times 17}} =$$

Where

$$torque_{eff} = 1.5 \times torque = 0.18 \text{ Nm} = 0.18 \times 1000 \text{ Nmm}$$

And kd is

$$\text{kd}_{sun} = \sigma_{adm} \times \frac{3}{(3+\text{Vp}_{sun})} = 37.5 \times \frac{3}{(3+\frac{\omega_{sun} \times m \times z_{sun}}{2})} =$$

$$= 37.5 \times \frac{3}{(3+\frac{47.12 \times m \times 17}{2})}$$

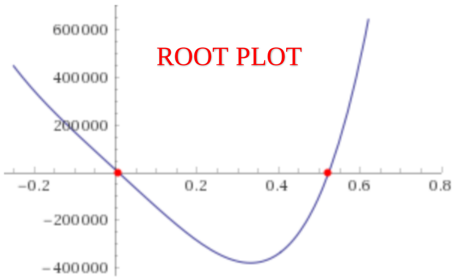
By substituting we obtain the following 4th order equation:

$$10\,813\,950 m^4 - 1\,571\,562\, m - 11\,772 = 0$$

Which results are:

$m \approx 0.0074907$

$m \approx 0.52323$



We consider the positive result getting a modulus of 0.52323, we approximate it to 0.75 for safety reasons. This result will be adopted for the design of all the first stage teeth.

Using this smaller modulus allows to decrease the ring gear diameter. In this way the overall gearbox will be smaller.

Now we will find the number of teeth of the planetary gears and the ring gear. For doing this we use the constitutive relation of the reducer:

$$i = \frac{1}{\tau_{st}} = \frac{Z_{sun}}{Z_{sun} + Z_{ring}} = \frac{17}{17 + Z_{ring}} = 0.2$$

From this we get

$$Z_{ring} = 68$$

But for allowing the assembly of the gearbox we need the following relation to be an integer number:

$$\frac{Z_{sun} + Z_{ring}}{\#\ planetary\ gears} = \frac{17 + 68}{3} = 28.3333 \notin \mathbb{Z}$$

To solve assembly trouble we decrease by one the number of ring teeth getting:

$$Z_{ring} = 67$$

And:

$$\frac{Z_{sun} + Z_{ring}}{\#\ planetary\ gears} = \frac{17 + 67}{3} = 28 \in \mathbb{Z}$$

Now we calculate the number of planetary gear teeth:

$$Z_{ring} = Z_{sun} + 2 \times Z_{plan}$$

By substituting Z_{ring}= 67 and Z_{sun}= 17 :

$$Z_{plan} = \frac{67-17}{2} = 25$$

Since the number of ring teeth is 67 and not 68, we need to calculate the new gear ratio:

$$\tau_{st} = 1 + \frac{\text{Ring gear teeth}}{\text{Sun pinion teeth}} = 4.941$$

Now having this data we use the software Autocad Mechanics in order to draw the reducer gears. In this way we are sure that the teeth match perfectly and we don't need to calculate their internal and external rounding.

Finally we need to check the feasibility of the second stage maintaining all the gear teeth adopted for the first stage. For doing this we evaluate the Lewis formula hoping to find the same 0.75 modulus:

$$m = \sqrt[3]{\frac{10.9 \times torque_{eff}}{n° \; plan.gears \times \lambda \times kd \times \text{Vpsun}}}$$

With

$torque_{eff}$ = 0.12 Nm × 4.941 × 1000 = 592.9 Nmm

$\lambda = 20$

$$\omega_{sun} = \frac{450}{4.941} \text{ rpm} = \frac{\frac{450}{4.941} \times 2 \times \pi}{60} \text{ rad/sec} = 9.5373 \text{ rad/sec}$$

Z_{sun}= 17

$$Vp_{sun} = \frac{\omega_{sun} \times m \times z_{sun}}{2} = \frac{9.5373 \times m \times 17}{2}$$

$$kd = \sigma_{adm} \times \frac{3}{(3+Vp_{sun})} = 37.5 \times \frac{3}{(3+\frac{9.5373 \times m \times 17}{2})}$$

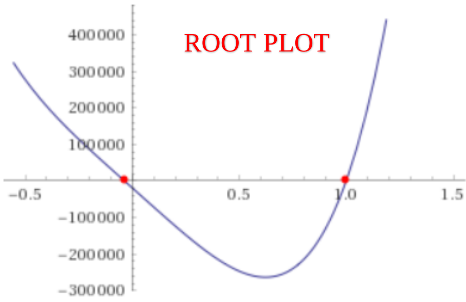
By substituting we obtain the following 4[th] order equation:

$$547222 \, m^4 - 523\,923 \, m - 19\,388 = 0$$

Which results are:

$m \approx 0.037003$

$m \approx 0.99764$



The obtained and required modulus is almost equal to 1. This result is expected since in the second stage we have a torque that's 4.941 times higher, the teeth receive much higher stress. In order to maintain the same modulus we design the second stage sun using C40 steel that has a

$$\sigma_{adm} = 200 \text{ N/mm}^2$$

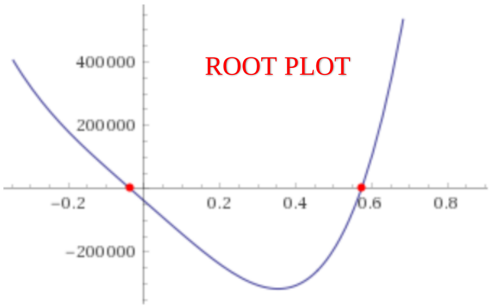
Doing again all calculations we get a final equation of:

$$5\,836\,824\ m^4 - 1\,047\,783\ m - 38\,775 = 0$$

Which results are:

$m \approx -0.00480$

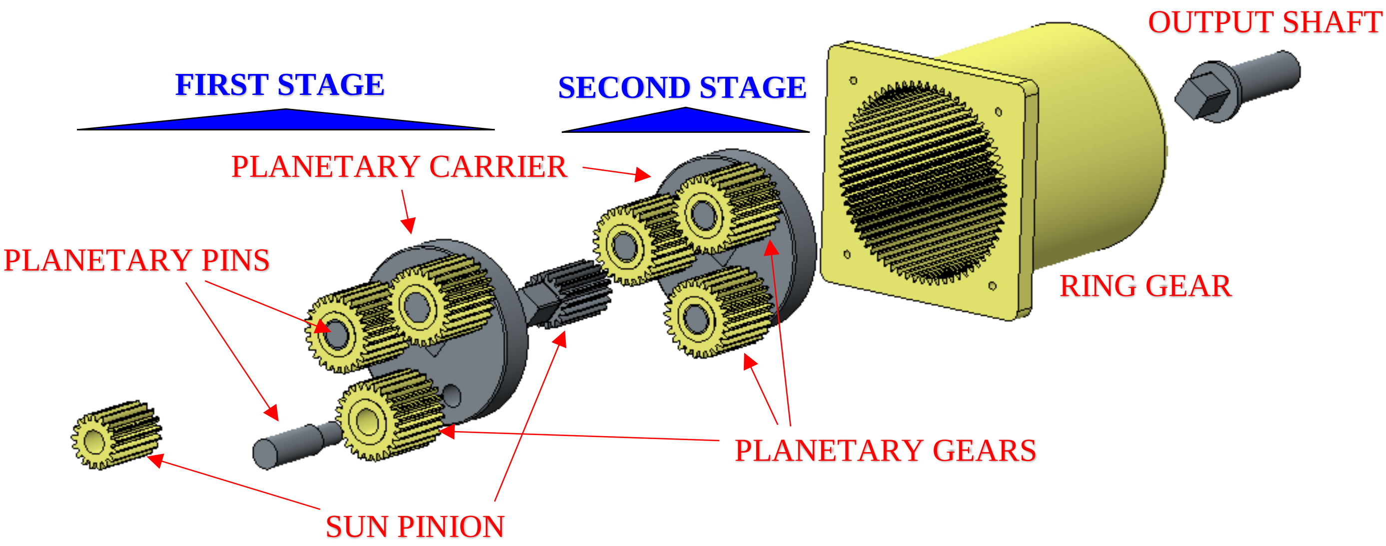
$m \approx 0.57240$



From the results we see that the C40 sun gear makes the second stage feasible since the allowed modulus are higher than 0.573, so 0.75 works well.

11.3 DESIGN:

After sizing the planetary gearbox we design it using Creo Parametric. We focus on its simplicity making interlocking components that are blocked in position by the motor.



As we have seen in the previous chapter, the first stage gears are made with PA 6.6 30% g.f. while the second stage ones should be in C40.

Actually making the ring gear in two different material isn’t possible, we should make two different parts that are than joint together. This doesn’t make sense since it would cost too much and increases assembly difficulty.

A better solution is to maintain the second stage sun gear in C40 and make its planetary gears and the ring gear using a more performing material Akroloy PARA ICF 40 black that has:

$$\sigma_{adm} = 87 \text{ N/mm}^2$$

In this case, from Lewis formula we obtain a modulus of

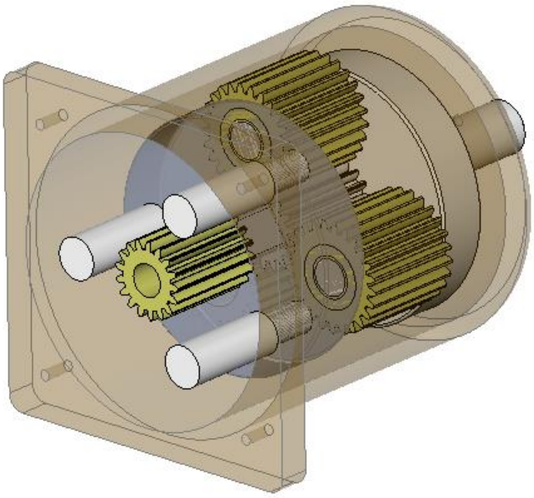
$$m \approx 0.7622$$

that is acceptable since sufficiently close to 0.75. We also highlight that the used σ_{adm} considers the gearbox constantly rotating. Instead in our case we would stress it just for few seconds.

Contrarily the second stage sun gear is maintained in C40 in order to create a robust connection with the first stage planetary carrier.

In order to make the reducer work efficiently we must be very precise in holes and tolerances: all the holes have a H7 tolerance, while shafts have a g6. A possible slack between gears could lead to peak of forces on a single tooth and thus to damage it. Instead we want everything perfectly matched so that all the gears work simultaneously.

The overall assembled gearbox is shown in the following, first stage planetary gears are omitted for better understanding while the first stage sun gear has to be intended connected to the motor.



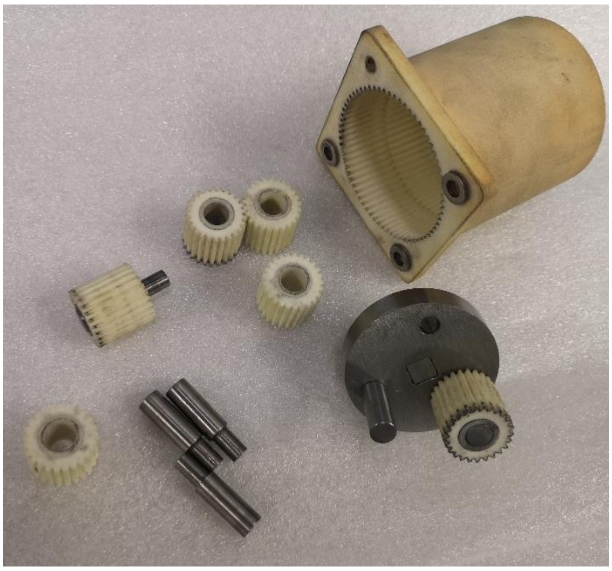
After these calculations we make a first gearbox prototype. As we said ring gear, first stage gears and second stage planetary gears are made using plastic while other components with C40. Since the metal parts are quite simple, Nuova Star decided to produce them by its own. They are made by milling and lathing procedures.

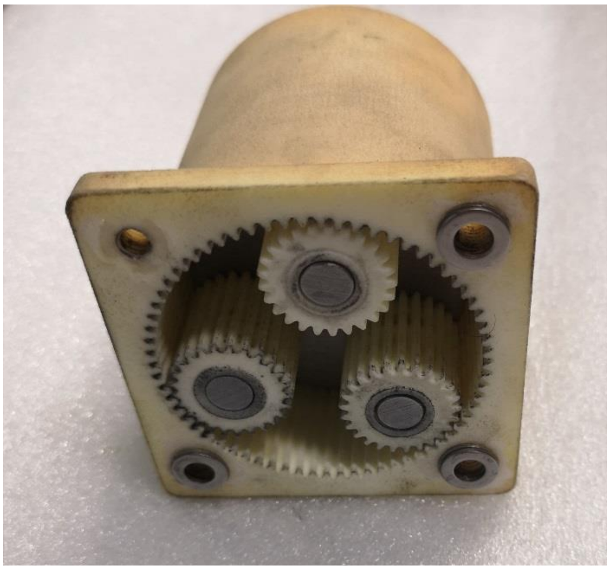
Plastic components are much more complicated since they require a mold, this means a huge initial investment. For this reason we decide to 3D print these parts to avoid the mold requirement. Our supplier make them using a particular material called Iglidur J that has a similar resistance to PA 6.6 30% g.f. : at 20° the admissible load is

$$\sigma_{adm} = 40 \text{ N/mm}^2$$

Actually the second stage requires a σ_{adm} much higher but since we are just making a prototype for computing only few cycles and for short intervals, we decide to make a first trial using this very cheap material.

Finally, after receiving all the parts, we assemble and try the reducer: it works well and looks quite robust. In the following two pictures, one with disassembled configuration the other with mounted gearbox.





12 TRANSMISSION VALIDATION:

After motor and gearbox design we compute a step-back checking if the forces and the angular speed of the transmission gears require bearing or not. For solving these calculations we start from the Simulink model. As we have seen the reducer output shaft rotates at a speed of 18 rpm stressed by a 3 Nm torque. A first calculation is done to evaluate if between the shaft and the frame support a bushing is enough or if a more expensive bearing is necessary.

We deal with the evaluation of bushing, if it's feasible for our application we can conclude this study, otherwise we will study the bearing solution.

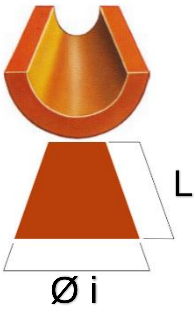
Key parameters in its sizing are two:

- Radial shaft speed;
- Specific load;

During working condition these two values are related by the following equation:

$$K = P \times V$$

Where K is a tabulated constant that depends on material, for example it's 1.549×10^6 N/(m × sec) for sintered bronze and 1.225×10^6 N/(m × sec) for sintered iron. Instead P is the pressure in N/m^2 that acts on the area obtained by projecting the internal bushing surface on a plane (see the following picture). Finally V is the linear peripheral speed of the shaft expressed as m/sec.



So we consider a bronze bushing having internal diameter hole of 6 mm, external one of 12mm and a thickness of 3.5mm. We obtain:

$$Projection = 6 \text{ mm} \times 3.5 \text{ mm} = 21 \text{ mm}^2 = 21 \times 10^{-6} \text{ m}^2$$

$$Maximal\ force = \frac{torque}{radius} = \frac{3\ Nm}{0.003\ m} = 1000 \text{ N}$$

$$P = \frac{Maximal\ force}{Projection} = \frac{1000\ N}{21 \times 10^{-6}\ m^2} = 47.62 \times 10^6\ \frac{N}{m^2}$$

$$V = \frac{\pi \times diameter}{60} \times angular\ speed = \frac{339.3}{60}\frac{mm}{min} = 5.65 \times 10^{-3}\ \frac{m}{sec}$$

$$K = 1.549\times10^6 \ \frac{N}{m \times sec} > 0.269 \times10^6 \ \frac{N}{m \times sec} = P \times V$$

This result show that the bronze bushing sustains the stress.

An additional validation is computed for the other two transmission gears. Since their reduction ratio is 3.958 they are subjected to a torque of 11.87 Nm and an angular speed of 4.547 rpm. Instead of using a bronze bushing we will use an iron one having diameter hole of 10 mm, and thickness of 3 mm. Following results are obtained:

$Projection$ = 10 mm $\times$ 3 mm = 30 mm^2 = 30 $\times$ 10^{-6} m^2

$Maximal\ force = \frac{torque}{radius} = \frac{11.87\ Nm}{0.005\ m} = 2374$ N

$P = \frac{Maximal\ force}{Projection} = \frac{2374\ N}{30 \times 10^{-6}\ m^2} = 79.13 \times 10^6 \ \frac{N}{m^2}$

$V = \frac{\pi \times diameter}{60} \times angular\ speed = 2.381 \ \frac{mm}{min} = 2.38 \times 10^{-3} \ \frac{m}{sec}$

$$K = 1.225 \times10^6 \ \frac{N}{m \times sec} > 0.188 \times10^6 \ \frac{N}{m \times sec} = P \times V$$

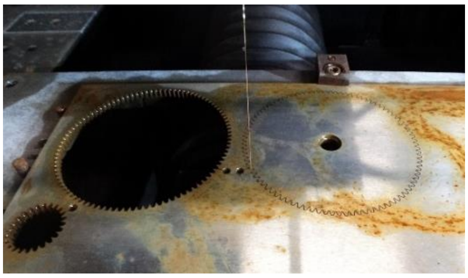
So also in this case the bushing is enough.

Anyway in these calculation we evaluate mechanical resistance of components. A possible trouble, hard to evaluate, comes from noise constraints. Having iron shaft that rotates on a sintered iron bushing could create noises. Till now we will go on without considering this eventuality, if after many cycles noise arises we will assess the use of bearings.

13 HINGE AND TRANSMISSION CONSTRUCTION:

Computed all the simulations and validated the whole model we can proceed with hinge and transmission construction. Since all the additional hinge components and all the transmission parts are steel made, Nuova Star decide to make them using internal machines.

Unless transmission gears, all the parts are quite simple and require just lathe and milling machine. Instead gears require a very high precision and are made using EDM (electrical discharge machining) machine.



TRANSMISSION GEARS PRODUCTION

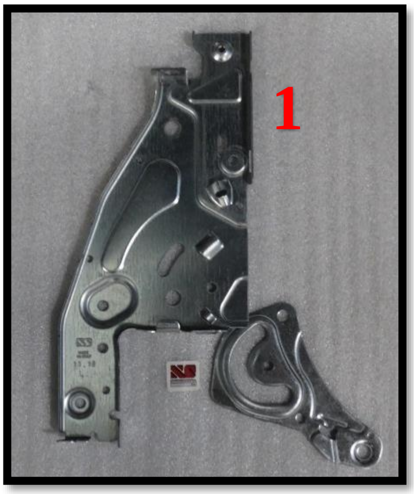
As we have already shown the motorized application is a plug-in of the variable fulcrum hinge. Starting from that we do the following:

1. Add holes in the box that will be used to fix the whole transmission. Also add a hole in the slot component in order to fix the additional link.
2. Assembly the first group of the hinge: slot component, hinge transmission, hook, rivets;
3. Assembly the second hinge group: box, lever, arm, rivets;
4. Decrease stopper diameter by 0.4 mm using lathe, in this way it will not make friction with the slot component;
5. Connect the two hinge groups using: stopper, additional link, spacer and pin;
6. Insert the spring in the plug and hung them to the hook;

Right now the hinge is computed and can be fixed to the dishwasher. We can also mount the combined hinge on the other side of the appliance. Finally the door can be mounted. With this configuration we obtain a manual variable fulcrum hinge with a perfectly balanced door.

Going further we mount the transmission-drive separately and finally we couple them to the hinge. Let’s analyze the steps:

7. Mount the motor on the gearbox;
8. Take the support layer, mount the bushing and the 95 teeth gears on it;
9. Connect the 24 teeth gear to the gearbox output shaft and screw it to the support layer;
10. Finally match the hinge pin to the 95 teeth slot and screw the transmission assembly to the hinge box;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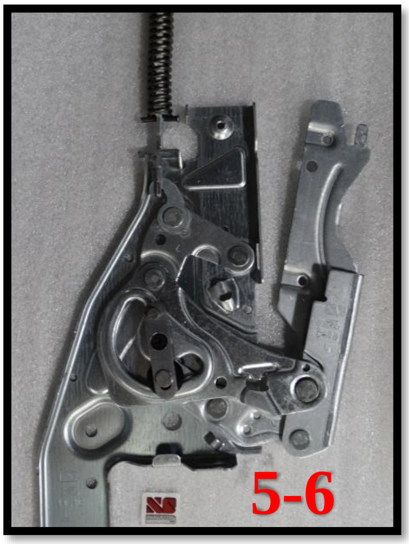
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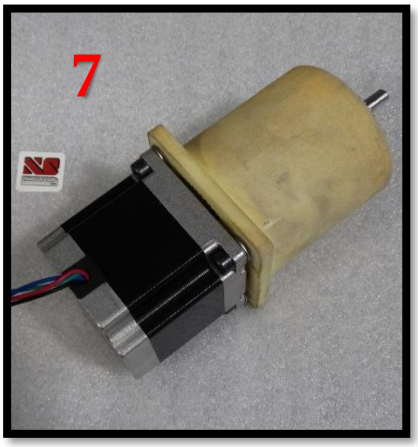
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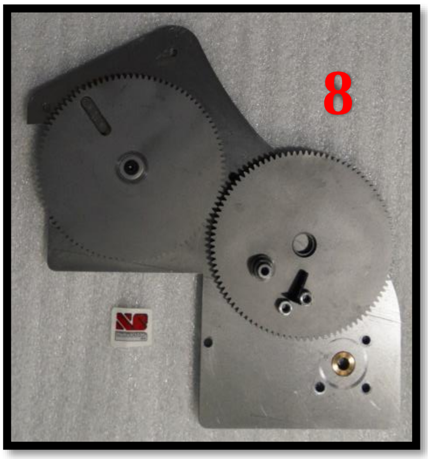
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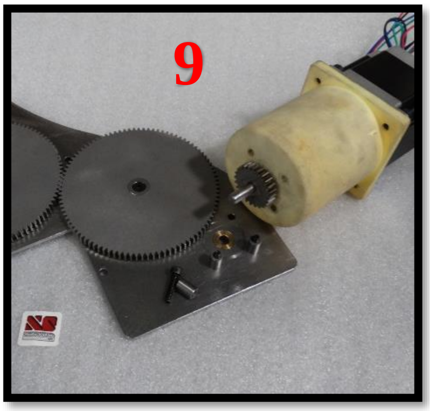
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14 PROBLEM SOLVING:

Everything matches and a first mechanical prototype is assembled. Before going on with electronic and software development we check if the mechanism works well.

We disassemble the motor and using a tool that simulates the output shaft of the motor, we make the gearbox moving slowly. Even if this kind of actuation doesn't represent the final law of motion we can already find mechanical criticality. From a first trial we see that the designed system works well and the door moves without too much effort unless in the last 5 degrees of closing procedure.

We go on doing analysis, we disassemble the gearbox and we measure concretely the torque that acts on the last transmission gear using a torque wrench. The measurement are not much precise and have an uncertainties of ± 0.8 Nm but at least they give an idea of the torque behavior:

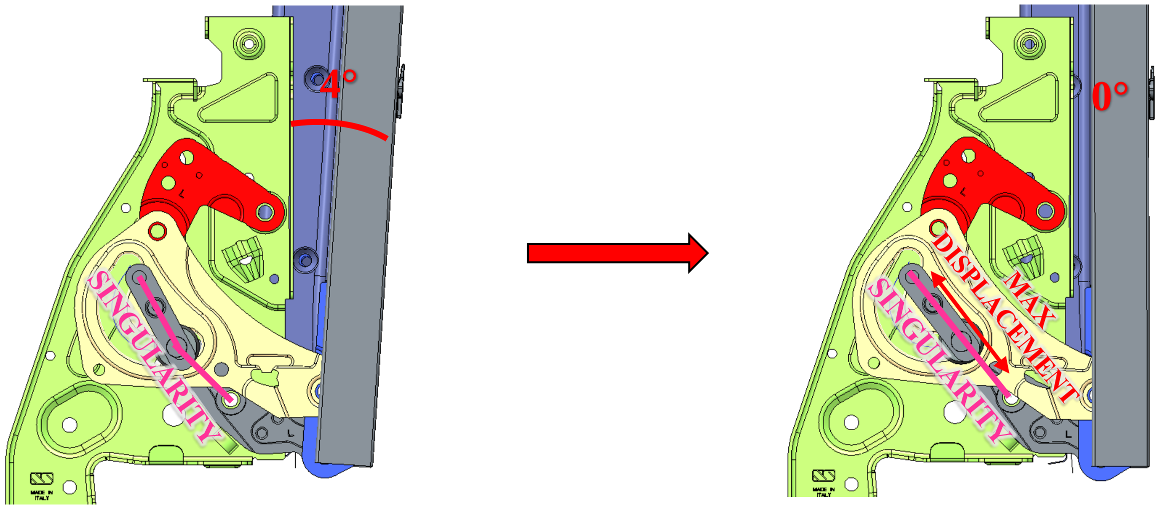
OPENING		CLOSING	
OPENING INTERVAL (°)	TORQUE (Nm)	CLOSING INTERVAL (°)	TORQUE (Nm)
0 - 5	2.3	88 - 70	-0.6
6 - 13	2.9	69 - 60	-0.4
14 - 45	1.7	59 - 48	-0.7
46 - 64	1.3	47 – 38	-1.1
65 - 70	1.0	37 - 30	-1.3
71 - 75	0.8	29 - 22	-2.1
76 - 81	0.6	21 - 12	-1.2
82 - 83	0.3	11 - 7	-2.8
84 - 88	0.1	6 - 4	-6.8
---	---	3 - 1	-12

A comparison with the simulation results is computed: the measured values behave as the Simulink motor torque unless in the last degrees of closing procedure. Other discrepancies are mainly due to frictions, that are roughly modelled in the simulation, and to measurement errors.

Now we focus on the abnormal requirement that's observed in the last 10 degrees of closing procedure. Hopefully it's due to some mechanics uncertainties and thus by reviewing some components we can delete it. If this peak can't be avoided we would need to study again all motor and gearbox supposing a 12 Nm torque request.

At this stage we remember that the slot component was modelled with a circular slot, instead in the steel part it has a different shape especially in the last degrees. This may be a cause that generate the incongruency.

Thus we focus our attention on this supposition and we try to work out by looking at the 3d cad model.



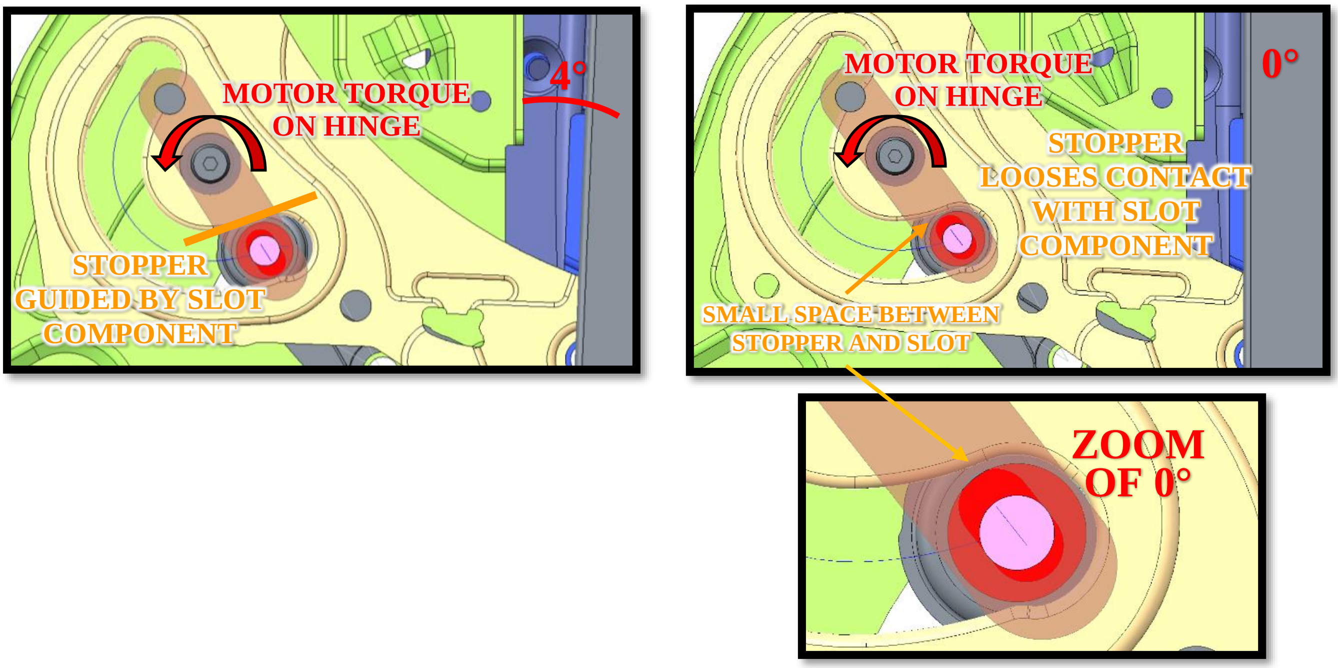
As we can see at zero degrees the hinge approaches a singularity position: the additional link and the lever are aligned. Thus a big link rotation leads to a small rotation of the door. Following the energy conservation principle we see that the effort necessary to rotate the link is quite low but requires more displacement to be computed.

This configuration is particularly useful but at the same time dangerous. Having all components perfectly aligned, as in our case, means that we have reached a maximal displacement configuration. For closing further the door we should increase the distances between the link fulcrum and the lever one. But in our case this is not possible since we have already reached the limit.

Now suppose to have a mechanical gain between some components that leads the door to stay a little bit opened even if the max displacement position has been reached. Following the previous consideration it wouldn't be possible to close further the door.

Instead in a forward-looking design we should suppose the existence of mechanical gain thus the mechanism should approach the singularity position without ever reaching it.

In the following pictures the stopper and the rivet are sectioned in order to show the contact with the other parts. The pin section is pink colored while the stopper one is red.



As we can see, in the last degrees the stopper losses contact with the slot. This condition occurs just in the software files where we represent the link between the stopper and slot component as a point (center of the stopper) on a line (slot construction line) constraint.

In real case the stopper is forced to collide against one of the slot wall. In this way a dangerous mechanical game is introduced. It’s quite small, more or less 0.8 mm, but the singularity condition amplify it. As a consequence the door requires a huge motor torque for reaching zero degrees position.

For solving this problem we should stay a little bit further from singularity configuration. But at the same time we don’t want to change the slot shape for the following reasons:

1. its change leads to a big review of the die, thus a huge expense;
2. a slot that impact against the stopper in the last degrees introduces a huge friction. As we said the mechanism is almost in singularity position and the added link is aligned with the lever, thus high forces are developed between stopper and slot side introducing frictions;
3. after many life cycles the stopper wear out against the slot, so the door progressively losses the ability of reaching 0 degrees. Changing the slot is not a robust solution;

An alternative answer to this mechanism fault consists in changing the additional link.

In order to avoid misunderstanding we remember that the additional link is composed of a pin that matches the 95 gear, a fulcrum and an additional small slot (do not confuse this with the bigger one of the slot component). A double diameter rivet slides in this small slot, passes through the stopper and finally reaches and catches the lever. Thus the rivet and the stopper are connected rigidly, constraining the first or the second element is the same.

We focus now on the rivet and we identify the position that it must reach at zero degrees. Since, as said, it’s linked to the stopper, this zero rivet position corresponds to the zero position of the stopper. But differently from it, the rivet is made by steel, it’s very robust and doesn’t produce much friction. Also the added link is made with steel and it’s robust too. As a consequence the 2nd and 3rd drawback previously found are avoided.

Talking about expenses, since rivet and link are new components, we should anyway make new dies for production purposes. Thus changing them doesn’t have impact on costs yet.

Finally we decide to modify the added link slot. Considering the already found rivet position at zero degrees we create an end stroke there. During closing procedure the pin and the stopper are forced in the right position by the additional link slot.

Anyway, for introducing this functionality we have to shift a little bit the link fulcrum. This change is required in order to move out a little bit from singularity condition. But changing the fulcrum will also require a variation of 95 teeth gear slot, its length is thus increased in order to cover all pin trajectory.

The obtained overall design stays far from the singularity position and is correspondingly able to reach -2 degrees door position. In this way we can compensate mechanical gain that appears after many life cycles.

Final design and functionality are presented in the following pictures:

Completed this conceptual design, we updated the real components and we mount everything on the dishwasher. We compute again the measurements using the torque wrench and we obtain excellent improvements:

OPENING		CLOSING	
OPENING INTERVAL (°)	TORQUE (Nm)	CLOSING INTERVAL (°)	TORQUE (Nm)
0 - 5	2.2	88 - 70	-0.6
6 - 13	2.9	69 - 60	-0.4
14 - 45	1.8	59 - 48	-0.7
46 - 64	1.2	47 – 38	-1.1
65 - 70	1.0	37 - 30	-1.3
71 - 75	0.8	29 - 22	-2.1
76 - 81	0.6	21 - 12	-1.2
82 - 83	0.2	11 - 0	-0.5
84 - 88	0	---	---

15 ELECTRONIC DEVELOPMENT:

Finished the mechanical design and assembled the real prototype we can now move to a different environment: electronic development.

We have seen that the motor has already been identified and simulated. We now need to identify the appropriate electronic components for moving it respecting the required functionality.

As first thing we need to fix the controller that we are going to use for monitoring the driver.

16 MICROCONTROLLER:

Generally a microcontroller unit (MCU) is a compact integrated circuit designed to govern a specific operation in an embedded system. A typical microcontroller includes a processor, a memory (flash, RAM, ROM, EPROM) and a I/O peripherals on a single chip.

In industrial applications MCU are often used especially in fast prototyping application. Among all the brand the most used and reliable is the famous Arduino. This brand represents a family of boards with an open source electronic. They stand out for the flexible and easy to use hardware and software.

Many boards exist but we choose to use the Arduino Mega. With respect to the other it integrates the processor ATmega2560, has many I/O interface (54 digital, 16 analog), a max frequency of 16 MHz and works at a nominal tension of 5 V.

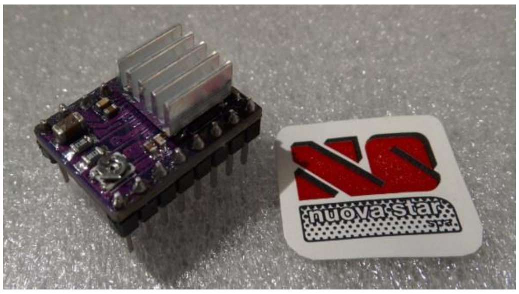
We are not going further with Arduino description, we just say that is very easy to program since it’s possible to connect it to a PC using a USB cable. The brand provides a GUI to program and load codes written in C++.

17 DRIVER:

In previous chapters we already identify the motor able to move properly the mechanism. We have seen that's a stepper motor and that generates a quite high torque at a slow rotational speed.

In order to actuate this stepper we decide to use a dedicated driver that can sustain a high voltage. This driver is able to receive signals from controller and transform them into current impulse that are sent to the motor.

As we have seen for the controller, a huge number of stepper driver exists. Among them all we identify a very cheap one that is produced by the Pololu company, it's a breakout board that incorporates the Texas Instrument driver DRV8825.



Its characteristic are impressive: it has an adjustable current limiter, over current and over temperature protection. It can reach a microstep resolution of 1/32 step operating from 8.2 V till 45V.

In order to make it work a deep knowledge is required, it's base on two half bridges, each one controls one phase. From datasheet we see that it's able to sustain up to 2.5 A per phase. This value is reached if the driver is cooled down properly. So we put a heat sink on it in order to disperse heat and sustain more or less 2.2 A.

For controlling the H-bridges two signals are requested:

- STEP: each rising edge makes the H-bridge trigger and as a consequence the motor computes a step or a microstep depending from the mode;
- DIRECTION: identify the required rotation direction so the proper couples of MOSFET activated or deactivated;

Now other 3 signals (MODE0, MODE1, MODE2) are used for defining the stepping behaviours:

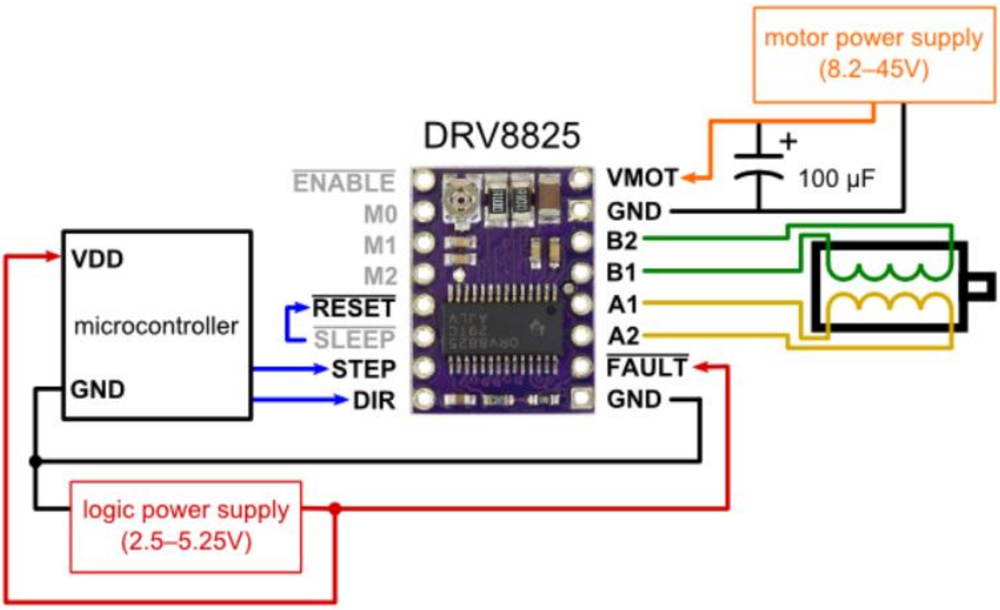
MODE2	MODE1	MODE0	STEP MODE
0	0	0	Full step (2-phase excitation) with 71% current
0	0	1	1/2 step (1-2 phase excitation)
0	1	0	1/4 step (W1-2 phase excitation)
0	1	1	8 microsteps/step
1	0	0	16 microsteps/step
1	0	1	32 microsteps/step
1	1	0	32 microsteps/step
1	1	1	32 microsteps/step

All these functionalities refers to Texas Instrument chip, actually Pololu goes further by inserting this chip on a board and surround it with useful electronic components. Among them the most influencing ones are the current limiters that have a direct impact on motor behaviors.

Two kind of them are mounted, one is fixed while the other is adjustable.

- FIXED CURRENT LIMITER: 2 sense resistors of 0.1 Ω allow to compare a reference current value for each phase current. Pololu sets this reference value to 2.2 A thus this is actually the maximum phase current reachable value. If an higher current occurs, the driver is shut down and an output fault signal is generated;
- ADJUSTABLE CURRENT LIMITER: To achieve high step rates, the motor supply is typically much higher than would be permissible without active current limiting. For instance, a typical stepper motor might have a maximum current rating of 1 A with a 5Ω coil resistance, which would indicate a maximum motor supply of 5 V. Using such a motor with 12 V would allow higher step rates, but the current must actively be limited to 1 A to prevent motor damage. Trimmer potentiometer on the board is used to set this current limit.

Now let's focus on the I/O ports, we first present an overall scheme than we describe their function:



- STEP: connected to the internal driver, it triggers the H bridge;
- DIR: identifies the motor rotation direction;
- SLEEP: when is low voltage it makes the driver entering a low-power sleep mode, instead turning it high will wake up the driver;
- RESET: if driven low it initialize the logic and disables the H-bridge outputs;
- M0, M1, M2: are used to select the stepping mode;
- ENABLE: if driven high it will disable all the board, actually leaving it disconnected makes the board working;
- GND: is connected to zero voltage;
- FAULT: pulls up whether a thermal shutdown or an overcurrent protection occurs;
- A1, A2, B1, B2: are the 4 wires coming from the bipolar motor phases;
- VMOT: is the supplied voltage of the power circuit, it must range between 8 and 45 V;

18 ELECTRONIC SETUP:

Before connecting the electronic to the dishwasher we must setup the driver current adjustable limiter. Avoiding this operation could lead to two opposite effects:

- If the current is too high the board broke up for overheating;
- If the current is too low the motor loses constantly steps;

Thus we must arrange properly this limit. For doing this we firstly define a motor reference voltage that we set to 24 V.

We take an electronic rectifier able to supply the required DC voltage and thanks to a breadboard we joint V^+ and ground with a 100 μF in order to stabilize eventual sharp requirements of current.

Right now our aim is to power up the motor without making it rotating. In this way the maximal allowable phase ΔV can be measured using proper instrument. It's thus possible to convert this value to a current corresponding one and tune it by arranging the adjustable limiter.

Following these guidelines we connect the driver I/O properly thanks to the breadboard: four phases with the motor wires, power supply to the DC rectified channel, other I/O to the microcontroller. We avoid the STEP and DIR connections such that the motor doesn't rotate.

Now using a multimeter we measure the voltage between one phase and the ground. Then by using a screwdriver we rotate the trimmer potentiometer. The phase voltage is shown on multimeter display.

From datasheet we obtain this formula that relates phase max current (I_{max}) with the measured voltage (V_{ref}):

$$I_{max} = 2 \times V_{ref}$$

Actually we estimate an I_{max} value of 2.2 A. Thus we turn the trimmer till a V_{ref} of 1.1V is found.

Completed this setup we are now able to assemble electronic to the dishwasher.

19 ASSEMBLY THE ELECTRONIC:

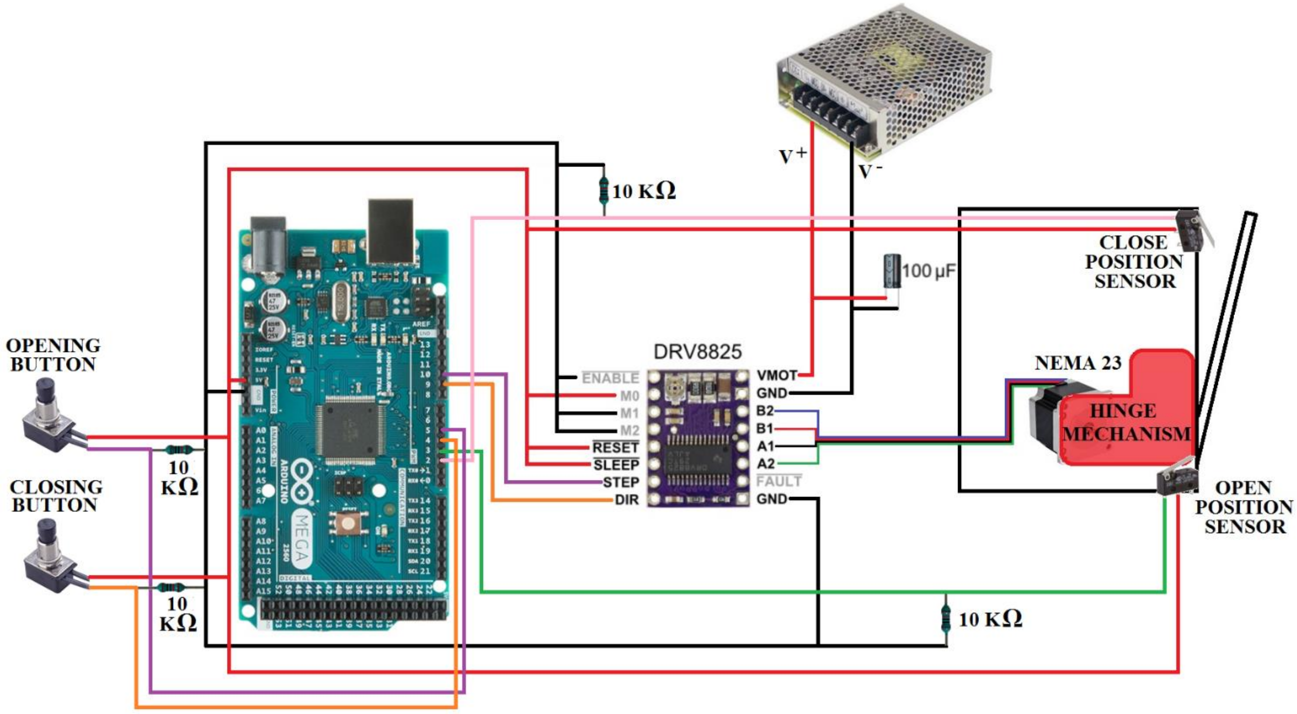
For prototyping purposes we decide to mount all boards on top of dishwasher, in this way it's easier to make I/O change and identify eventual dangerous fault. Obviously in the final products all boards will be merged on a single one mounted near the motor in the bottom part.

The overall electronic model consists of a motor, its driver, the power supply and the controller. Actually we are missing some additional components that allow to receive inputs from the users and from door end strokes.

Thus we add two push button and two switches. The first components allow to receive opening and closing requests from the user. While the second components allow to feel the 0° or 88° position of the door. Both them are classified as micro switches and work with the normally open logic: as soon as the user or the door touch them, the circuit closes and a signal is sent as output.

We then assemble everything on the dishwasher, for linking all components we use a breadboard with jumper wires. In the following picture a circuit diagram is shown, it represents the used elements with the adopted connections.

We highlight that the Arduino board is connected to the PC, in real application is alimented by a dedicated 5 V rectifier.

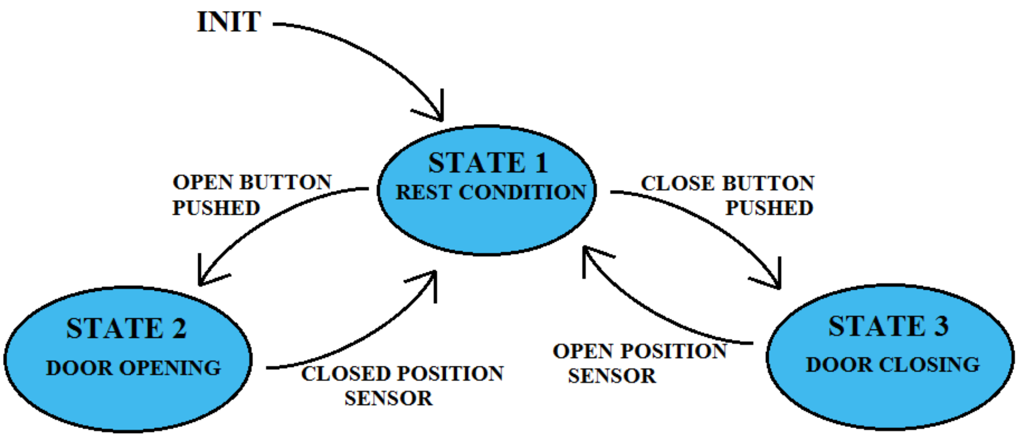


These schematic represents the essential design for respecting the main characteristic of the system. Till now we don't add other components but we go strictly to program writing in order to validate the overall concept.

20 PROGRAM WRITING:

Linking the Arduino board to the PC allows to program it using the high level language C++.

Before writing code we decide to create a state diagram:



The scheme is quite simple and very functional, by implementing it we obtain a system able to open or close the door as soon as the open or close button are pushed respectively. The motion is interrupted as soon as an end stroke switch feels the door presence.

This logic doesn't allow to interrupt the door motion in the middle by pushing a button. Anyway it's a good starting point from which developing more complicated functionalities.

Now talking about coding itself, we use Arduino libraries that implements a precompiled series of commands for controlling properly the stepper.

Following the state chart the code can be split in 4 parts:

INIT: the Arduino.h library and the driver one (DRV8825.h) are called, then everything is initialized: first input buttons, second the end stroke switches, third the driver. Finally the stepper speed profile is defined, this allows to choose a trapezoidal law of motion setting the acceleration and deceleration rates.

INIT

```
#include <Arduino.h>
#include "DRV8825.h"
DRV8825 stepper(MOTOR_STEPS, DIR, STEP, SLEEP, 1, 0, 0);
// Motor steps per revolution. In full step we have 200 steps for revolution, we use half step.
#define MOTOR_STEPS 400
#define RPM 450
#define DIR 9
#define STEP 10
//end stroke sensors
#define SENS_CLOSE 2
#define SENS_OPEN 3
//variable declaration
int state=1;
int stepper_on =0;
//set pin button number
const int buttonPin_close = 4;
const int buttonPin_open = 5;
//variable pin button
int buttonState_close = 0;
int buttonState_open = 0;
```

1. STATE 1: in this state we impose the motor to stop and we read cyclically the push button signals. The state changes to value 2 if the open button is pushed otherwise to value 3 if close button is pushed.
2. STATE 2: motor is fed and opens the door following the previous defined trajectory. It has to continue till open position switch sends a signal. Since the motor control algorithm has blocking behavior we are forced to define switch signal as interrupt. Its trigger changes the state to 1 immediately.
   Having sensors as interrupts is acceptable since reaching a mechanical stroke-end must stop the driver.
3. STATE 3: the functionalities are similar to state 2 but differently from it we change the motor rotation direction and look for close position switch.

```
void loop() {
      while (state==1){
           stepper_on=0;
           stepper.stop();
           //read the state of the pushbutton value:
           buttonState_close = digitalRead(buttonPin_close);
           buttonState_open = digitalRead(buttonPin_open);
           if(buttonState_open == HIGH && digitalRead(SENS_OPEN) == LOW) {
              state=2;
              }
           else if(buttonState_close == HIGH && digitalRead(SENS_CLOSE) == LOW) {
              state=3;
              }
          }
        while (state==2){
                  //motor opens
                  if (stepper_on==0){
                   stepper.rotate(10*360);
                   stepper_on=1;
                   }
          }
        while (state==3){
                      //motor closes
                      if (stepper_on==0){
                       stepper.rotate(-10*360);
                       stepper_on=1;
                   }
        }
   }
//interrupt 1 definition
void blink1() {
   while(state==3){
   stepper.stop();
   state=1;
    }
   }
//interrupt 2 definition
void blink2() {
   while (state==2){
   stepper.stop();
   state=1;
   }
  }
}
```

STATE 1

STATE 2

STATE 3

INTERRUPTS

Now this program is very limited since uses a blocking behavior command: *stepper.rotate( )*. This code line is fundamental since lets the motor rotate of an angle defined between the parenthesis. Anyway for making the acceleration trajectory, it forces the MCU to continuously calculate the step frequency. In this way the obtained motor trajectory is very precise but the MCU is not able to make any other computation.

Thus the main functionality is maintained but if we want to implement the capability of blocking the door in the middle by pushing a button, the program becomes critical.

Finished this first simple program we upload it to Arduino board.

21 FIRST TEST:

First test has the object of verifying mechanism functionalities and acquire a tangible feeling of door motion. Thus we immediately run the program with success and by pushing a button the door starts to move.

Results are quite promising since the mechanism answer is instantaneous and the motor overcomes power request. Mechanical components are robust enough and the door moves from 0 till 88 as requested. Even after 40/50 cycles mechanism works well and the components don't show any damage, neither the gearbox.

Downside are a quite jerky motion especially in opening phase. Instead we would like a smoother and more silent behavior for both the phases.

Another problem comes from motor heat up. After few cycles the increasing temperature of the motor makes it loosing maximum torque. As a consequence during closing trajectory it looses few steps and creates a loud noise. Anyway this downside don't compromise overall functionality since the motor is still able to close the door.

Final trouble come from the high reduction ratio introduced by the gearbox. With this configuration, that has a total ratio of 96.61 (3.958 x 24.41), we don't achieve reversibility. This last term means the possibility of opening and closing the door manually even if the motor is powered.

Instead motorized application should work without excluding the normal operation way. Especially in failure situation the manual use is essential, for example if a blackout occurs when the dishwasher is closed, we must be able to open it.

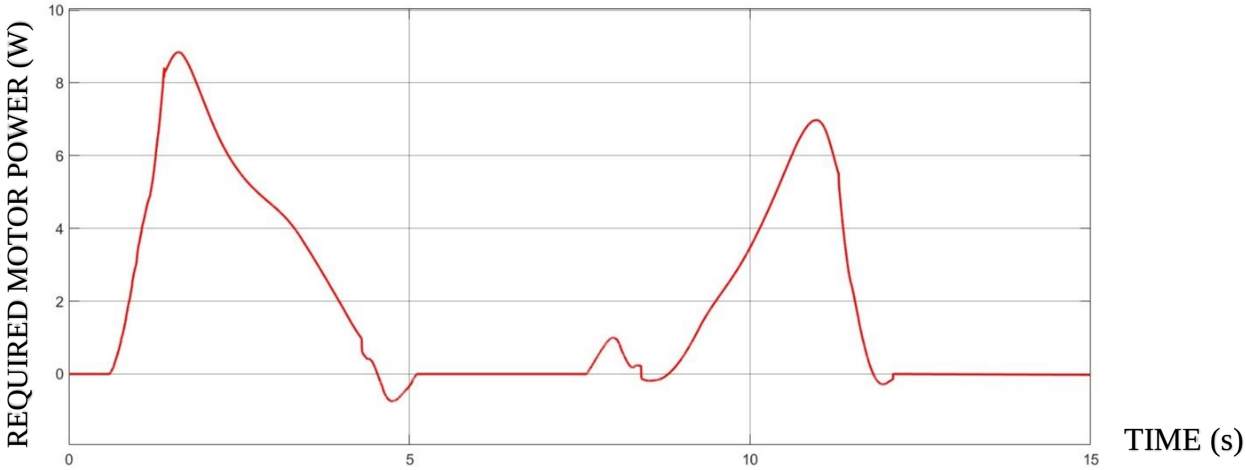
Before making further adjustments, we let our customers evaluating the prototype. The innovative concept leaves them speechless but they all agree on having faster cycles and reversibility option.

22 PROTOTYPE UPDATE:

Following these guidelines we update the prototype. We have seen that motor selection was computed ranging between 6-8 seconds cycle time. In order to reduce the speed we must review the motor choice and the reduction chain.

So we consider a new time requirement equal to 4.5 seconds. As we already seen for motor selection, we start by calculating power requirement. We use the same Simulink model deleting the gearbox blocks in order to evaluate its output shaft requirements.

By launching the simulation we obtain the following power plot:



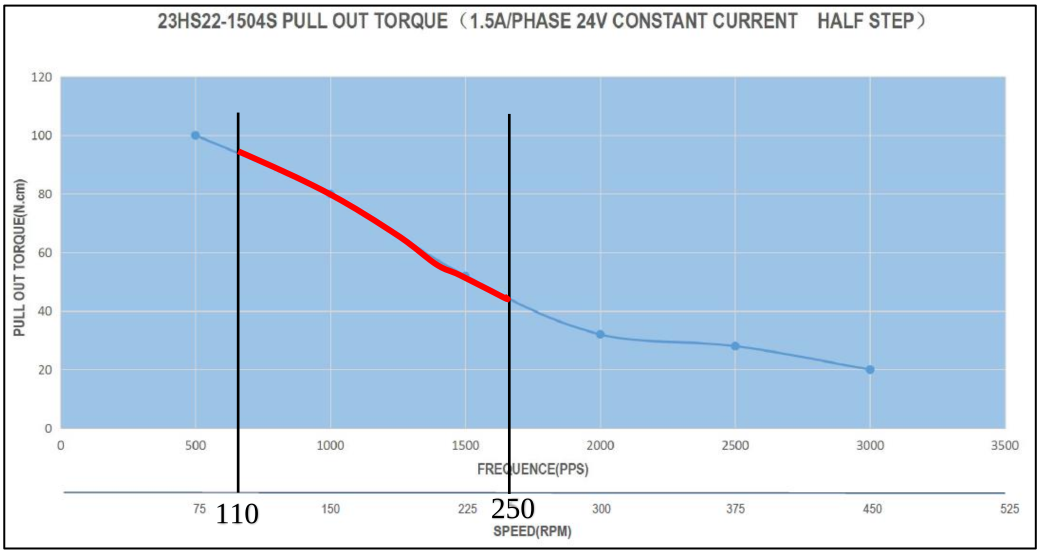
From the plot we find a peak value of 9 W that maintaining a safety margin becomes 10.5 W.

As peak rpm we obtain 22 rpm that with a safety margin becomes 23 rpm. Concerning torque we get a peak of 3.8 Nm that again with a safety margin becomes 4.2 Nm.

Converting these two values into power we obtain:

$$\text{Power (W)} = \frac{4.2\ (\text{Nm}) \times 23\ (\text{rpm})}{9.5488} = 10.116\ \text{W} < 10.5\ \text{W}$$

Their product is lower than the considered power, so they respect the constraints.
Now we move to motor curve and we highlight in red the section that generates required power:



For obtaining the limits value we consider the couple of speed and torque that respect power constraint:

$$\textbf{Power (W)} = \frac{\text{Torque x speed}}{9.5488} < 10.5\ \text{W}$$

Thus inside the red line the speed-torque couple generates enough power.

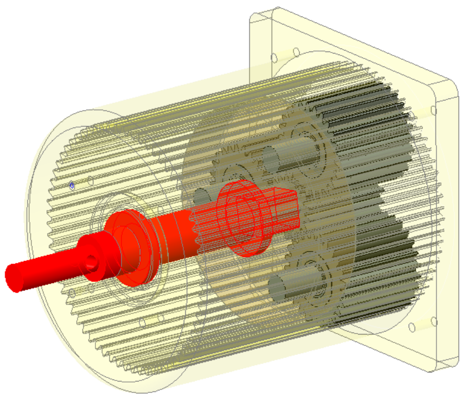
Now we consider the speed ratio among the motor required speed (23 rpm) and the allowable one. We obtain 2 values that represent the gearbox minimum and maximum values:

$$\tau_{min} = \frac{110\ (\text{rpm})}{23\ (\text{rpm})} = 4.78$$

$$\tau_{max} = \frac{250\ (\text{rpm})}{23\ (\text{rpm})} = 10.87$$

But the mounted planetary gearbox has a total ratio of 24.413 split between two equal stages having a ratio of 4.941 each. Thus the intuition comes from this values: if we skip one gearbox stage a ratio between the maximum and minimum value is reached.

Taking the 3d file we understand the possibility of eliminating the second stage by constructing a new and longer output shaft. The obtained component has the following shape:



This particular shape has the function of maintaining the mounted stage in the original position. The alternative solution for maintaining the same output shaft would be to redesign the ring gear and print it again. This requires few time and is much more expensive.

Thus the overall gearbox ratio becomes 4.941. We now compute a final check to validate it. The maximum motor speed is:

Max speed (ω_{mmax}) = Required speed(rpm) $\times$ Gear ratio = 23$\times$ 4.941= 113.6 rpm

Now we check if the corresponding motor torque is enough:

$$\text{Required pull out torque (Nm)} = \frac{\text{Motor torque upper bound (Nm)}}{\text{Gear ratio}} = \frac{4.2\ \text{Nm}}{4.941} = 0.850\ \text{Nm}$$

This torque value is enough since at 113.6 rpm the motor generates 0.94 Nm.

Now focusing on the gearbox we check if it resists. We consider first stage setup so that:

σ_{adm} = 37.5 N/mm^2

$$\lambda = \frac{b}{m} = 20$$

$$Vp_{sun} = \frac{\omega_{sun} \times r_{sun}}{2} = \frac{\omega_{sun} \times m \times z_{sun}}{2}$$

Where ω_{sun} is the angular velocity of the sun, such that:

$$\omega_{sun} = 113.6 \text{ rpm} = \frac{113.6 \times 2 \times \pi}{60} \text{ rad/sec} = 11.90 \text{ rad/sec}$$

We use Lewis formula:

$$m = \sqrt[3]{\frac{10.9 \times torque_{eff}}{n° \, plan.gears \times \lambda \times kd \times \text{Vpsun}}} = \sqrt[3]{\frac{10.9 \times 1.275 \times 1000}{3 \times 20 \times kd \times \text{Vpsun}}} =$$

Having

$$torque_{eff} = 1.5 \times torque = 1.275 \text{ Nm} = 1.275 \times 10^3 \text{ Nmm}$$

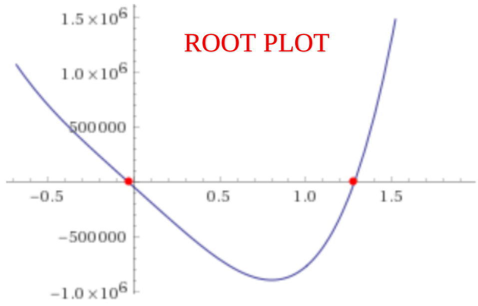
And kd is:

$$kd_{sun} = \sigma_{adm} \times \frac{3}{(3+Vp_{sun})} = 37.5 \times \frac{3}{(3+\frac{\omega_{sun} \times m \times z_{sun}}{2})} = 37.5 \times \frac{3}{(3+\frac{11.90 \times m \times 17}{2})}$$

By substituting we obtain the following 4th order equation:

$$682\,763 \, m^4 - 1\,405\,732 \, m - 41\,693 = 0$$

Which results are:



m ≈ -0.02966

m ≈ 1.2819

Thus the PA 6.6 is not enough robust for sustaining a 0.75 gear modulus.

For solving this problem we make it using C40 that has:

$$\sigma_{adm} = 200 \text{ N/mm}^2$$

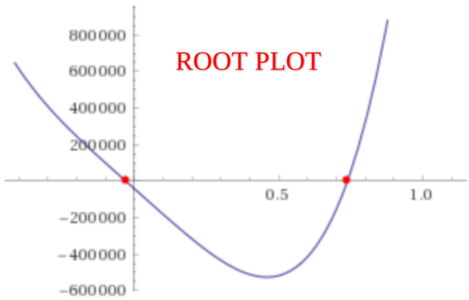
Doing again calculations we get:

$$3\,641\,400 \text{ m}^4 - 1\,405\,732 \text{ m} - 41\,693 = 0$$

Which results are:

m ≈ -0.03240

m ≈ 0.73828



So the C40 sun gear works well for the one stage reducer.

Anyway since we want to make a fast and cheap trial we decide to assembly the gearbox with the available components thus we maintain the ring gear and the planetary gears in Iglidur J material hoping that they will not wear out too soon. We're aware that the final gearbox teeth must be realized in C40 steel.

Finally we compute a brief validation of transmission bushing.

Following the "Transmission Validation" chapter, the following calculation are computed for the reducer output shaft bushing:

$$Projection = 6 \text{ mm} \times 3.5 \text{ mm} = 21 \text{ mm}^2 = 21 \times 10^{-6} \text{ m}^2$$

$$Maximal\ force = \frac{torque}{radius} = \frac{0.85 \times 4.941\ Nm}{0.003\ m} = 1400 \text{ N}$$

$$\text{P} = \frac{Maximal\ force}{Projection} = \frac{1400\ N}{21 \times 10^{-6}\ m^2} = 66.66 \times 10^6\ \frac{N}{m^2}$$

$$\text{V} = \frac{\pi \times diameter}{60} \times\ angular\ speed = 7.223\ \frac{mm}{min} = 7.22 \times 10^{-3}\ \frac{m}{sec}$$

$$K = 1.549 \times 10^6 \frac{N}{m \times sec} > 0.481 \times 10^6 \frac{N}{m \times sec} = P \times V$$

So the bronze bushing applied to the 24 teeth transmission gear is enough.

Doing the same calculation for the 95 teeth gear we get:

$$K = 1.225 \times 10^6 \frac{N}{m \times sec} > 0.202 \times 10^6 \frac{N}{m \times sec} = P \times V$$

The iron bushings sustain the effort.

23 FINAL TEST:

Completed the feasibility studies we assemble the new mechanism with the modified gearbox. We then change the software by decreasing the target speed and we load it on the MCU.

The subsequent tests show a remarkable improvement, the stepper motor works better at lower speed. We see that it's more silent and during closing procedure it doesn't lose any step. The speed increment is appreciable and decreases the user waiting time in front of the appliance.

Another fundamental achievement is reversibility, since we decrease the reduction ratio by 4.941, we can now open and close the door manually. This function is essential for safety condition, it allows to open and close the appliance even if a shutdown occurs.

Anyway a small downside is the lack of fluency during opening cycle, especially between 40 and 50 degrees the door vibrates a little bit. Since the main functionality of the appliance are anyway respected we will not further review the mechanism. In the future, if the application will attract customers, we will modify the software and eventually some components in order to compensate vibrations. The main goal is the realization of a soft and smooth motion that's appreciable from a visual point of view.

Next a picture of the final test prototype:

With this last development we consider our practical work completed. Anyway we will show some possible implementation that improve the application quality.

24 FURTHER IMPLEMENTATION:

24.1 GEAR CASCADE:

Completed the overall development we get a mechanism that has a total gear ratio of

$$\tau_{tot} = 3.958 \times 4.941 = 19.56$$

Now focusing on planetary gearbox ratio (4.941) we see that's quite small. Actually it's just a little bit higher than the gear one (3.958).

We also notice that the gearbox is a quite complicated component. Its made by several parts and its mounting procedure requires time and increases the costs. For having a better estimation we ask the opinion of a Nuova Star supplier. In this analysis we assume the production of 10 000 reducers every year and we don't consider the initial investment due to dies. The valued cost ranges between 10-12 euros each gearbox.

So a great improvement in term of costs and size can be obtained by merging the gearbox inside the cascade gears. For implementing this change we are forced to modify the transmission shape since reaching a ratio of 19.56 with just one gear pair would be unfeasible.

We decide to develop this concept using two consecutive gear pairs. Talking about numbers we see that coupling a 80 teeth gear and a 18 one leads to a gear ratio of

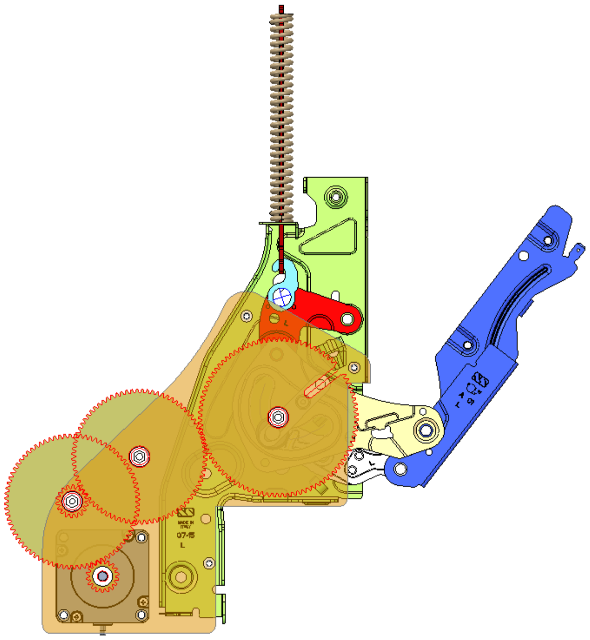
$$\tau = \frac{80}{18} = 4.44$$

By inserting two consecutive gear coupling the obtained ratio becomes:

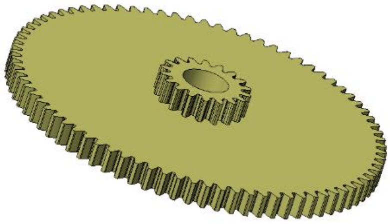
$$\tau_{cascade} = (4.44)^2 = 19.753$$

That's almost equal to the previous τ_{tot} value.

Following this results we design the 3d model on which the gear position is defined graphically by respecting the gear tangency and the structural constraints. The obtained model is the following:



Where the intermediate gear is realized with a double teeth gear. If this component is plastic made than it's feasible, if it's steel made we should make it with sintering technology or otherwise we should review its design.



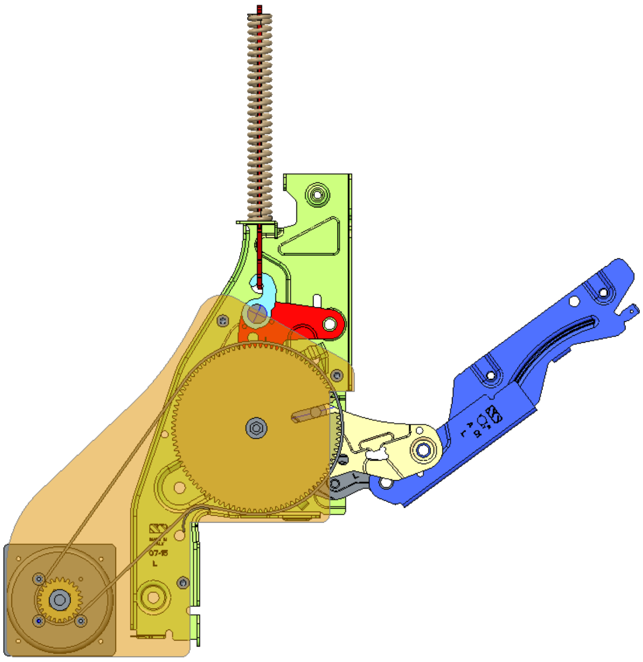
Right now we are not going on with gear cascade development since the intention of this chapter is just to present the concept. Anyway this is probably the best further development that we will follow.

24.2 BELT TRANSMISSION:

A particular alternative solution consists in using belts instead of gears. A big disadvantage of having gears is that they require a high precision during assembly procedure and all the sustain holes must be perfectly parallel or otherwise the gears make a lot of noise and eventually ruin the teeth.

In our case, especially in huge volume case, this precision is very hard to maintain. So a valuable alternatives are timing belts that are more flexible since they are elastic and can compensate misalignment.

But a huge disadvantage of belts transmission is that's not possible to reach high reduction ratio, thus in this scenario we are forced to maintain a gearbox. In the following picture a conceptual design is presented.



24.3 NON BLOCKING ALGORITHM AND GYROSCOPE:

Target goal of this thesis is developing a new working concept. With the final prototype we reached a starting point for further studies. Indeed looking back to primary functionalities we see that few of them have not been faced yet.

We will now focus on safety requirement that ask the system to stop immediately as soon as an obstacle interferes with the door, this characteristic is fundamental and must be integrated for marketing purposes.

A possible way to reach it is by current absorption analysis. We firstly need to create an expected current absorption curve that is estimated from simulations. Then a feed-forward control is implemented: the motor current demand is constantly measured and compared to the threshold expected curve, as soon as the demand overcomes the expectation, the motor is stopped since an external activity occurs.

But for computing this process we must have a feedback from the motor that allows to constantly measure the current absorption.

With a DC motor or an AC motor having this information is quite simple, but stepper motor are just controlled in open loop. As we have seen the MCU just control the stepper using STEP and DIR I/O, it hasn't an analogic signal that measures the current absorption. Consequently we have no possibility of controlling it.

Alternative solutions consist in obtaining information from additional sensors. We now describe a solution that make use of a gyroscope-accelerometer, a small sensor that measures both linear and angular accelerations. We refer in particular to MPU 6050, a very cheap and efficient sensor. It uses a I2C protocol to communicate with the MCU and the processed signal gives the acceleration values.

So, by positioning one of these sensors on top of the door, we are able to measure how it's moving. Ideally during normal door trajectory, after a short interval, the sensor measures a constant angular velocity.

Thus an impact against the door would change this value creating an abrupt deceleration. The processed signal is read by the MCU and since it's different from the expected zero deceleration, the motor is immediately stopped and brought back to waiting state.

The main trouble of this implementation is that requires a data processing. The sensor just gives back binary numbers that represent the acceleration, not an interrupt signal that's low or high. Thus the binary value it's analyzed recursively by Arduino board requiring a little bit of computational time.

But actually during opening and closing cycle the MCU enters blocking algorithm that spends all processing capability in driver control occupying 100% of available time . This doesn't allow to make gyroscope data computation.

In order to make it feasible we need to replace the blocking algorithm with a non-blocking one sacrificing the motor control ability, thus trajectory smoothness. In future application we will try to match a non-blocking algorithm with trajectory smooth requirements, finding a good compromise.

24.4 VOCAL SENSOR:

Another great implementation consists in changing the input modality. Till now we use two simple buttons that are uncomfortable and unaesthetic. Right now automation tries to substitute people effort and movement with technology. Instead of push buttons we want to insert a vocal sensor able to hear voice commands. In this way opening, closing or stopping the door is controlled without any effort.

A possible trouble in this sensor implementation is that a cheap board is not reliable, it just detects 70% of vocal commands. An alternative and more efficient board that detects 97% of commands costs 50 euros and is not feasible for our cheap implementation.

Probably we will implement a prototype using the expensive sensor just for marketing purposes, then we will let the customer decide among cheap buttons or expensive vocal sensor.

24.5 INDUSTRIALIZATION:

A final step consists in industrialize the appliance. Conceptually this means to adapt all the parts to high volume production. For decreasing the costs we must leave prototype environment entering the production one.

Indeed we have to make steel component using stamps, sintering or eventually adopt commercial components. Concerning plastic parts we have to make them feasible for injection molding technique. In this last case we should avoid sharp design and perpendicular or discontinuous parts. Then think more in general about sintering or molding, using less material means making cheap parts, so many components should be lightened and simplified.

A final consideration regards life cycles tests. After developing and creating provisional components we should test their resistance to normal wear out. If their life span is too short we must select more robust materials or make stronger shapes.

24.6 PATENT:

Obtained prototype represents an innovation for domestic appliances. Many Nuova Star competitors should be interested in this kind of product. Right now this application is just reserved for a niche market but in few years automated appliances will be ordinary in our houses. It’s thus necessary to protect the invention.

Following this purpose in the next few months a patent will be written and presented to European patent office. This document will be effective and will protect also the possible implementations.

25 CONCLUSIONS:

The goal of this thesis is the development of a new dishwasher concept: a motorized door that opens and closes automatically.

To better deal with the work load we split the design in 3 parts: mechanics, electronics, software.

We start from mechanics considering the already produced variable fulcrum hinge. We study deeply its kinematic by computing analysis in three different ways: graphic, software, mathematic. All the results let us understand the optimal components on which applying efforts.

We than create a first design solution which is based on a simple additional link that generates an effective leverage mechanism.

Since the dishwasher allowable space for motor is located far from the hinge, we design a transmission system based on gears. Gears coupling is also used to introduce a reduction ratio of 3.958.

The model obtained is exported to Simulink and two simulations are computed. The first one has the goal of identifying the optimal springs, the second one focuses on required power for opening and closing the door.

From this last simulation we are able to select a stepper motor and the gearbox required ratio that's equal to 25. A final overall simulation is performed in order to check that the motor satisfies the requirements.

We than move to gearbox design. Since we want to minimize cost and size we develop and realize a two stages planetary gearbox by ourselves. For prototype purposes we make ring gear, planetary gears and an input sun gear using 3d printed plastic material, while other components are realized using C40. Actually we know that this material is not robust enough for production purposes but our intention is just to make few demo cycles thus we assumed it sufficient.

Further validations are computed for transmission system bushings, we obtain that they sustain the speed and the transmitted load without requiring bearings.

Computed these studies we realize all the components and assemble the mechanism. Using a torque wrench we measure the torque required and we compare the real results with the simulations ones. A big discrepancy is found in the last degrees of closing procedure, a torque peak is measured thus we review the design.

By studying the 3d drawing we find a mechanical play that was neglected in the simulations. By redesigning some parts we are able to delete it and the new torque wrench tests show that the adopted adjustments work well.

Further we move to electronic environment. Since stepper motor was already selected we identify a driver controlled by a simple MCU, the Arduino Mega.

We assemble everything using breadboard, jumper wires and rectifier; we also add micro switches to create electronic end strokes and buttons to acquire user commands.

In order to make the driver working well we adjust its active current limit: a trimmer potentiometer placed on the board. This setup was fundamental otherwise the motor could lose constantly steps or vice versa burn out.

Completed the electronic we write a simple C++ code. It’s based on 3 states:

1. Door is stationary and waits for user inputs, if open button is pushed moves to state 2, instead if close button is pushed state changes to 3;
2. Door opens till 88° where end stroke switch is reached, then goes back to state 1;
3. Door closes till 0° where the other end stroke switch is reached, then goes back to state 1;

By uploading and running it in Arduino we obtain a first working prototype. It works well and we let our costumer seeing it, as a response they suggest us to increase opening and closing speed. We thus reduce the cycle time and by doing simulation we obtain the new requested power. Luckily our mounted motor is able to generate that but just on a specific speed interval.

By doing a reverse reasoning we identify a gearbox ratio that allows to stay in that speed range. We then observe that our planetary gearbox is composed by two stages, each single stage has a ratio that belongs to the required interval. Thus for obtaining the right ratio we just need to skip one planetary stage.

In this way we just need to review one single component instead of all the gearbox; in a short time we update the appliance and compute the final tests.

Further improvements are still necessary, but till now a new working concept has been developed.

26 BIBLIOGRAPHY:

- William L. Cleghorn, (2005), *Mechanics of Machines;*
- E. Funaioli, A. Maggiore, U. Meneghetti (2005), *Lezioni di Meccanica applicata alle Macchine;*
- V. van der Wijk (2014), *Methodology for analysis and synthesis of inherently force and moment-balanced mechanisms: theory and applications;*
- Singiresu S. Rao (2011), *Mechanical Vibrations;*
- G. Legnani, M. Tiboni, R. Adamini, D.Tosi (2016), *Meccanica degli azionamenti;*
- V. Parenti Castelli, *Notes from course of Mechanics of Machines for Automation M 1, academic year 2016-2017.*
- R.Vertechy, *Notes from course of Mechanics of Machines for Automation M 2, academic year 2016-2017.*

27 ACKNOWLEDGEMENTS:

... to my father.

I've always admired the devotion,
the responsibility,
the commitment
that you show in ordinary life.
If I've reached this goal,
it's because of your magnificent example.

Bologna, July 2019,

Samuele Pellegatti