Diplomarbeit DIPL-186

Design and analysis of the Leonardo da Vinci's automatic automobile

von
Eduardo Sanz Regano

Betreuer: Prof. Dr.–Ing. Prof. E.h. P. Eberhard

Universität Stuttgart
Institut für Technische und Numerische Mechanik
Prof. Dr.–Ing. Prof. E.h. P. Eberhard

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Chapter 1

Introduction

This thesis involves the modeling, analysis and simulation of the mechanism. This mechanism is the first auto designed by Leonardo da Vinci. The design of this mechanism is obtained from [1] [2], figure 2.1.

Figure 1.1: Leonardo da Vinci's automobile [2].

The first step was the compression about how function these mechanism and following defined the dimensions. Then the diferents parts would be designing with the program CATIA. The next task would be assembling the mechanism to do with this the kinematics analysis.

The mechanism can be separate in two differents mechanisms, one of this is the propulsion mechanism, and the other one is the direction mechansim. Both are powered with the same motor.

Chapter 2

Mechanism

In this chapter is describe the automobile, the mechanisms and the joint, to proceed with the design and the analysis.

2.1 Propulsion mechanism

This mechanism consist of two motors with dynamic and kinematics equivalents, each motor is on one wheel, figure 2.1. This thesis will be only analyzed with one type of motor, what leads to a symmetric system.

To transmit the movement between the motor and the wheel, it has a group of gears with a one to one transmission ratio. This gear train consist of two spur gears with a one to four transmission ratio, and two bevel gears with a four to one transmission ratio. this gives in total a one to one transmission.

2.2 Direction mechanism

This mechanism received the movement from one of the motor, figure 2.2.This movement is transmitted to the direction wheel by a cam disk and a follower, the cam disk is placed in the motor shaft, and the follower is in charge of transmitting movement to the wheel direction shaft. This will result in a swinging movement, which will be transformed in a discontinuous movement using a ratchet. With it the wheel will revolves gradually.

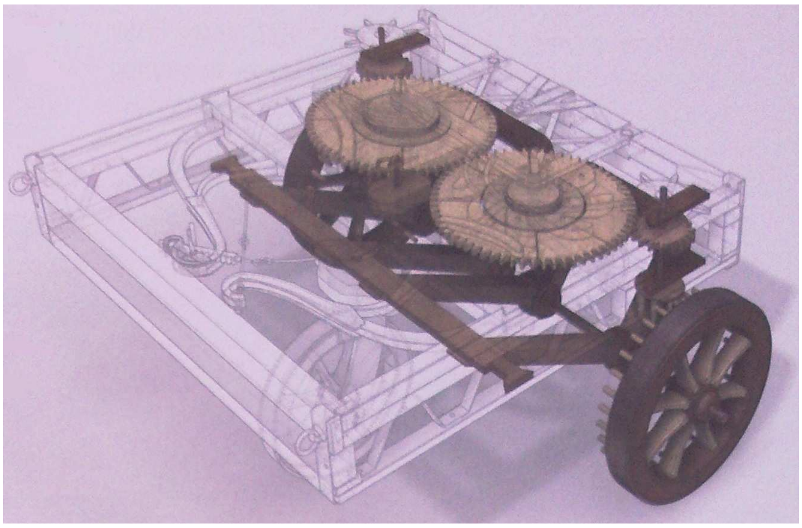


Figure 2.1: Propulsion mechanism [1].

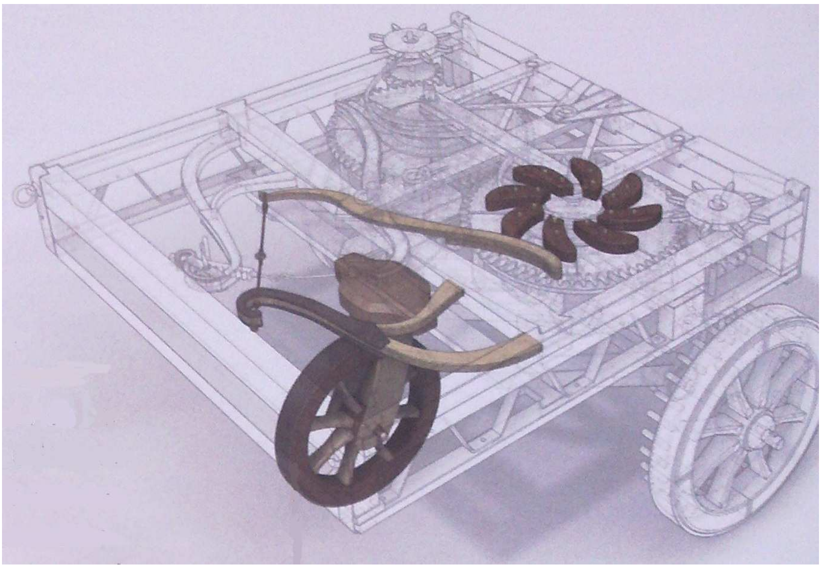


Figure 2.2: Direction mechanism [1].

2.3 Ensemble

When the mechanisms work simultaneously, the trajectory described is curve. This could be change, if the cam disk has different number of cams.

Chapter 3

Modelling of the mechanism

3.1 Introduction

To design and model the mechanism was used the software CATIA, which was developed to design, manufacture and computer aided engineering. it can do complex parts in three dimensions. This software is directed to the production, is a CAM program. CATIA is in CAE applications as it prepared to analyse products. this software has a lot of modulus, for this project were used four of them: *Part Design*, *Assembly Design*, *Generative Shape Design* and *DMU Kinematics*.

In this chapter is described the modelling and design of the different mechanisms parts, as the assembly.

The scale used is the appropriate for this model. The measurements were estimated keeping the equipment in a working order, as the Leonardo da Vinci's documents do not have measurements. This car was never constructed.

3.2 Modelling the mechanism

3.2.1 fixed part, chassis

Could be considered the base of the mechanism, it will be the fixed part to which all other parts are going to be referenced and assembled.

To design the chassis was onlly used the *Part Design* modulus. This modulus can easily and precisely create parts and allows adaption of the design requirements, sketching in a defined plane.

The first step defines the place of the first pair of gears, the spur gears. the length between shafts will be defined from the gears geometry, figure 3.1.

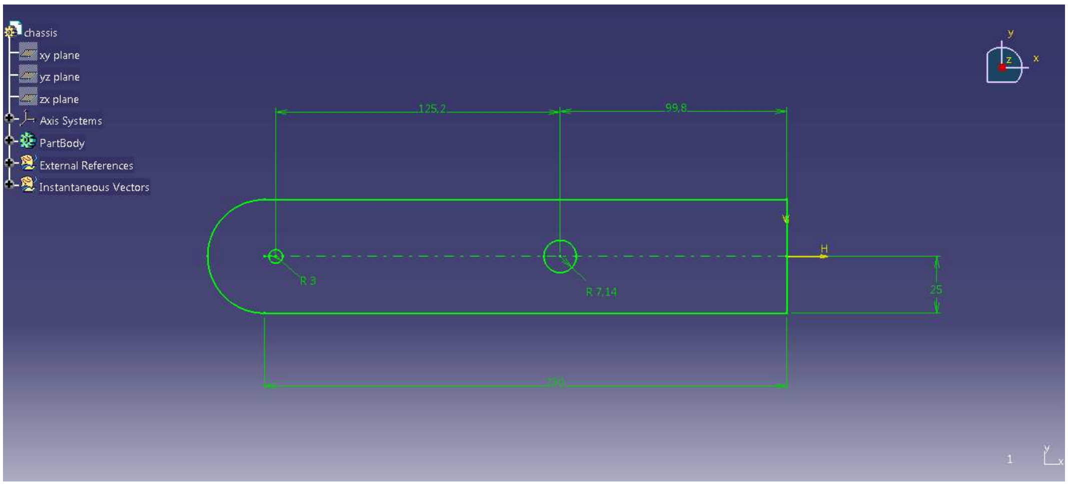


Figure 3.1: Position spur gears shaft.

The next step is the bevel gears zone. the smallest is assembled in the small plane gear shaft and the biggest is assembled in the same shaft of the motor wheel, figure 3.2.

The last sted is the follower position and the guide wheel shaft, figure 3.3.

3.2.2 Spur gears

The spur gears are the first links to the gears train, the big one is the driver and the small one is the driven gear.

Both gears are designed in the same way, using the *Generative Shape Design* and the *Part Design* modulus. The *Generative Shape Design* modulus permits modelling surfaces in three dimensions. In these gears is only to define the teeth geometry, sice the *Part Design* capacity is not enough to that end. When the geometry is finished, the design will be completed with the modulus part design.

To do the gear design is necessary to know some common parameters for both gears to keep the gears in a working order.

The gears should keep the transission ratio constant in all the work, equation 3.1.

$$\frac{\omega_1}{\omega_2} = \frac{\boldsymbol{Z}_2}{\boldsymbol{Z}_1} \tag{3.1}$$

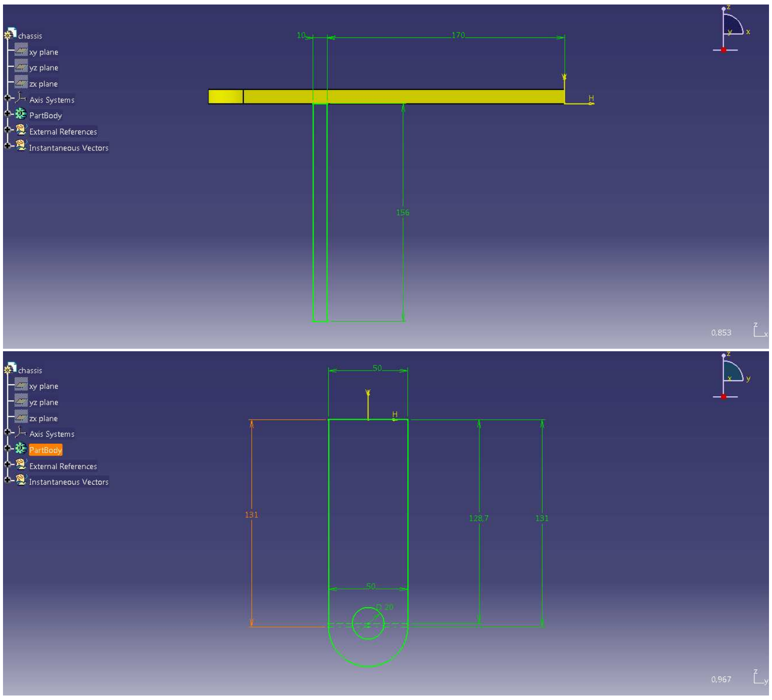


Figure 3.2: Position big bevel gear shaft

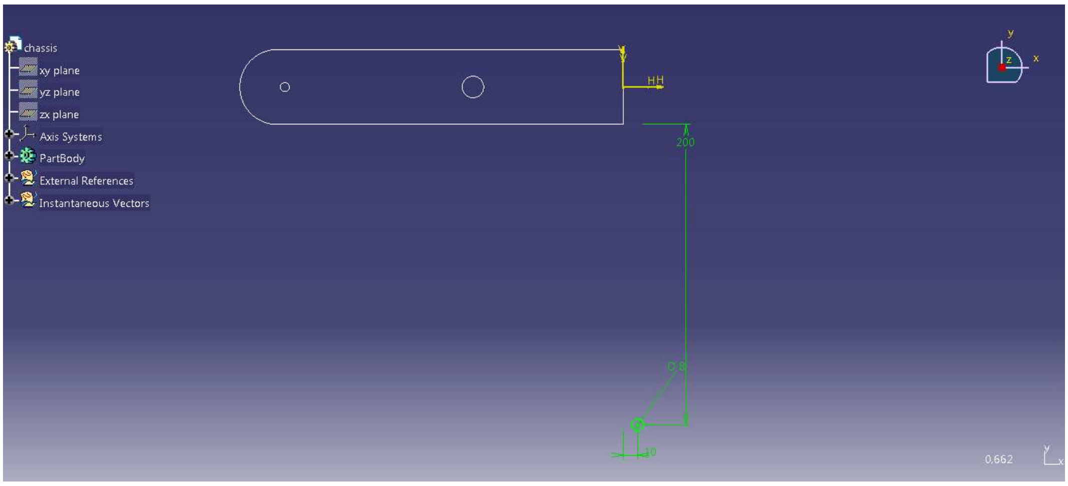


Figure 3.3: Position guide wheel shaft.

This relation reamins for the involute curve from the pitch circle.

To normalize the gears geometry, was defined a parameter, the modulus, equation 3.2.

$$\boldsymbol{m} = \frac{2 \cdot \boldsymbol{r}}{\boldsymbol{Z}} \tag{3.2}$$

As can be read in the equation 3.2 the gear diameter is dependent of the modulus and the number of tooth. Therefore the distance between shaft is equations 3.3 3.4.

$$\boldsymbol{a} = \boldsymbol{r}_1 + \boldsymbol{r}_2 \tag{3.3}$$

$$\boldsymbol{a} = \frac{\boldsymbol{m}}{2} \cdot (\boldsymbol{Z}_1 + \boldsymbol{Z}_2) \tag{3.4}$$

The shafts can be separated more than $\boldsymbol{a}$, as in this thesis, the transmission ratio will continue to be constant, this difference will be such that the gear do not have contact in the pitch circle.

The parameters defined to design the gears are in table 3.1.

Table 3.1: Parameters plane gears

Parameter	Name	Value
$\boldsymbol{Z}$	Number of teeth	–
$\boldsymbol{a}$	Pressure angle	20 deg
$\boldsymbol{m}$	modulus	2
$\boldsymbol{r}_p$	Radius of the pitch circle	$\boldsymbol{r}_p = \boldsymbol{m} \cdot \boldsymbol{Z}/2$
$\boldsymbol{r}_a$	Radius of the outer circle	$\boldsymbol{r}_a = \boldsymbol{r}_p + \boldsymbol{h}_a$
$\boldsymbol{r}_b$	radius of the base circle	$\boldsymbol{r}_b = \boldsymbol{r}_p \cdot \cos(\boldsymbol{a})$
$\boldsymbol{r}_f$	Radius of the root circle	$\boldsymbol{r}_f = \boldsymbol{r}_p + \boldsymbol{h}_f$
$\boldsymbol{r}_c$	Radius of the root concave corner	$\boldsymbol{r}_c = \boldsymbol{m} \cdot 0.38$
$\boldsymbol{c}$	Angle of the point of the involute that intersects the pitch circle	$\boldsymbol{c} = (\sqrt{1/(cos^2(\boldsymbol{a}) - 1)})/(\pi)$
ϕ	Rotation angle used for making a gear symetric to the ZX plane	$\phi = -tan\boldsymbol{y}/\boldsymbol{z} + 90deg/\boldsymbol{Z}$

The first step is to sketch the involute curve, to do it the following equations have been introduced 3.5 3.6.

$$\boldsymbol{y} = \boldsymbol{r}_b \cdot \Big(\sin(\boldsymbol{t} \cdot \pi) - \cos(\boldsymbol{t} \cdot \pi) \cdot \boldsymbol{t}\pi \Big) \tag{3.5}$$

$$z = r_b \cdot \Big(\cos(t \cdot \pi) + \sin(t \cdot \pi) \cdot t\pi \Big) \tag{3.6}$$

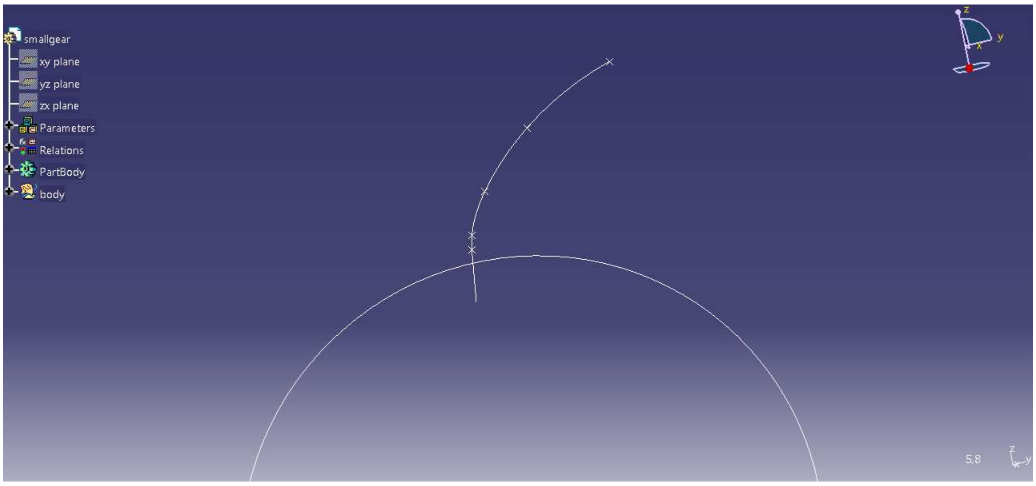


Figure 3.4: Involute curve.

The geometry has been rotated a ϕ angle, in order to do later the geometry of the teeth using symmetry, figure 3.4. The next task is introducing a corner between the involute and the root circle with r_c radius. When this was defined, the last step to do the teeth geometry is to use symmetry for completing the teeth, figure 3.5.

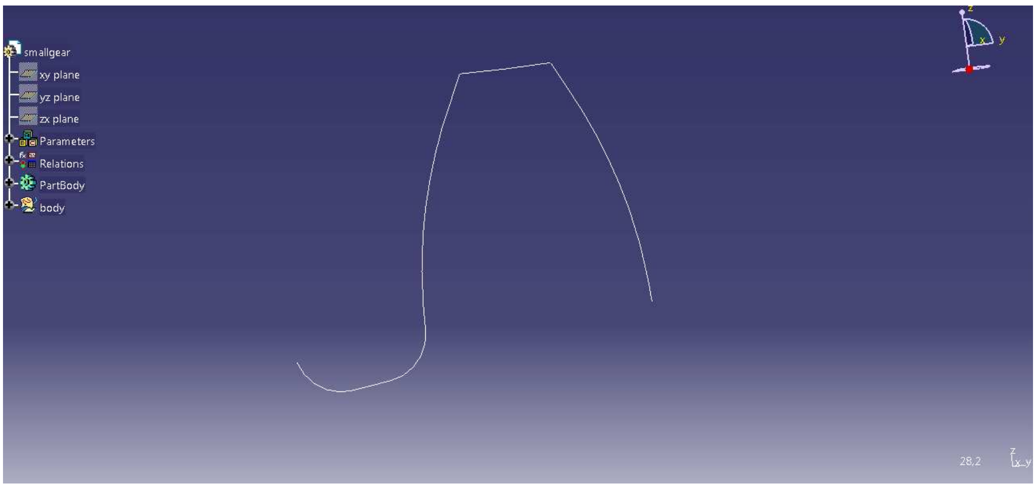


Figure 3.5: Teeth profile.

The gear profile is just a circular repetition of the tooth. To continue, in the *Part Design* modulus the gear geometry has been extrude to have the final gear.

As the gear is defined with parameters, the design could be modified changing the number of tooth, modulus, etc. The two gears are designed thanks to this capacity.

This pair of gears has a transmission ratio of one to four. As the equation 3.1 shows the number of tooth could be defined for the big gear 100 teeth, figure 3.7, and for the small gear 25, figure 3.6. With this parameters defined the distance between the shafts will be defined also using equation 3.4.

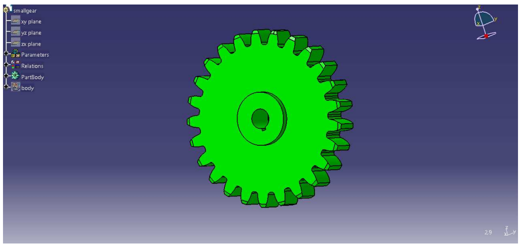


Figure 3.6: Spur gear with 25 tooth

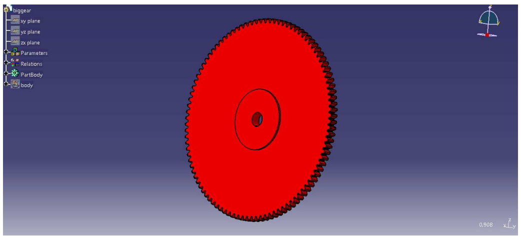


Figure 3.7: Spur gear with 100 tooth

3.2.3 Bevel gears

This pair of gears defines the second link in the gears train. The small one has been assembly in the same shaft than the small plane gear, and is also the driver gear. The big one is the diven gear, and it transmit the movement to the motor wheel.

This gear have been defined like the plane gears using the same modulus. The different was in defining the surface in the *Generative Shape Design* modulus, once this is done design will be completed in the *Part Design* modulus.

These gears keep all the relations that have been described before. In this case the gears have between them 90 degrees.

Some parameters have been defined in table 3.2.

The first step is to sketch one of the teeth, now the work space is not a plane, is aconics surface that has been defined previously, figure 3.8. The rest of the work is similar to the one done for the plane gears.

When the conics surface was defined the profile of the teeth has been sketched,

Table 3.2: Parameters conic gears

Parameter	Name	Value
$\boldsymbol{Z}_1$	Number of teeth	–
$\boldsymbol{Z}_2$	Number of teeth complementary	–
$\boldsymbol{a}$	Pressure angle	20 deg
$\boldsymbol{m}$	modulus	2
δ	Half angle of the front primitive cone	$\delta = atan(\boldsymbol{Z}_1/\boldsymbol{Z}_2)$
$\boldsymbol{l}_d$	Length of the teeth on the front primitive cone	10 mm
$\boldsymbol{P}$	Pitch of the teeth on a straight generative rack	$\boldsymbol{P} = \boldsymbol{m} \cdot \pi$
$\boldsymbol{r}_p$	Radius of the pitch circle	$\boldsymbol{r}_p = \boldsymbol{m} \cdot \boldsymbol{Z}/2$
$\boldsymbol{r}_a$	Radius of the outer circle	$\boldsymbol{r}_a = \boldsymbol{r}_p + \boldsymbol{h}_a$
$\boldsymbol{r}_b$	radius of the base circle	$\boldsymbol{r}_b = \boldsymbol{r}_p \cdot \cos(\boldsymbol{a})$
$\boldsymbol{r}_f$	Radius of the root circle	$\boldsymbol{r}_f = \boldsymbol{r}_p + \boldsymbol{h}_f$
$\boldsymbol{h}_a$	Addendum, height of a tooth above the pitch circle	$\boldsymbol{h}_a = \boldsymbol{m}$
$\boldsymbol{h}_f$	Dedendum, Depth of a tooth below the pitch circle	$\boldsymbol{h}_f = \boldsymbol{m} \cdot 1.25$

and then all the depth in the correct direction was also defined with the conics surface.

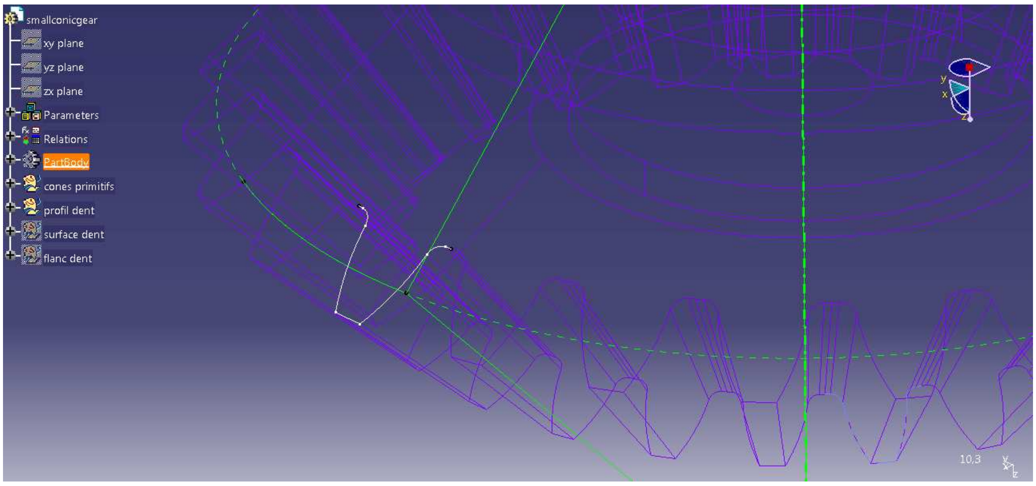


Figure 3.8: Involute bevel gear teeth.

After that the surface was sketched, the tooth surface has been defined, figure 3.9.

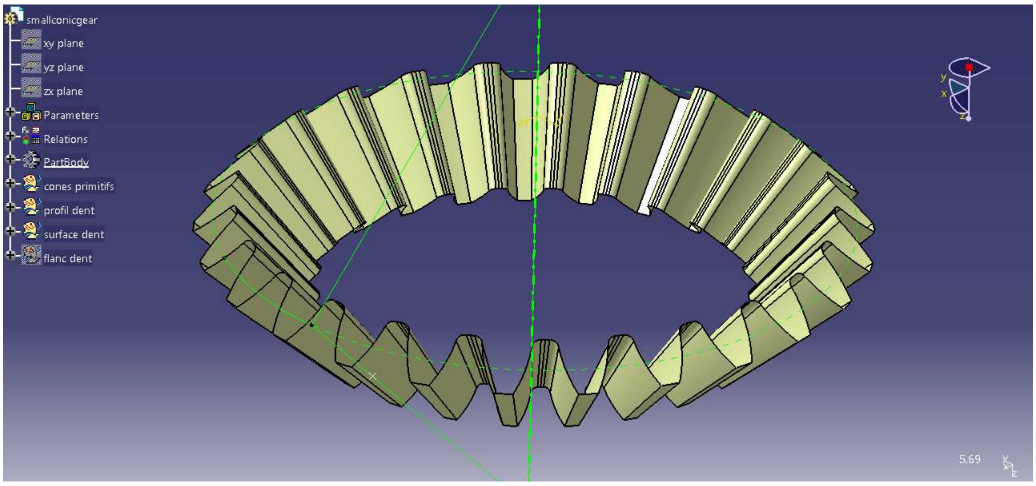


Figure 3.9: Tooth surface.

The conics internal and external surfaces have been drawn. Following the design has been finished in the *Part Design* modulus, figure 3.10.

Like the last case, the gears have been defined as a function of parameters, which could be change if there is any need of varying the of teeth in the pair of gears.

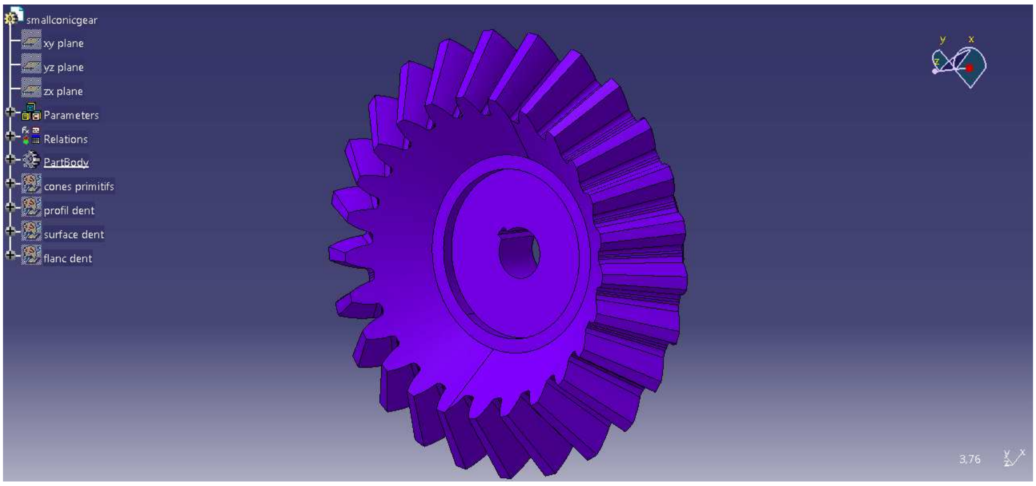


Figure 3.10: Bevel gear.

3.2.4 Cam disk

This disk transforms the rotation movement into a swinging movement in the direction wheel shaft with the follower part. This disk has been designed following the proposed model of Leonardo da Vinci. With this design the number of cams can be changed.

To design this part has been used the modulus *Part Design* and *Generative Shape Design*. Most of the part has been sketched in *Part Design* modulus, except the contact curve between the cam disk and the follower, figure 3.11. In such a way the parameter that defines the number of cams can be modified,figure 3.12.

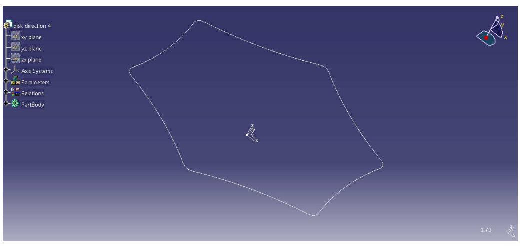


Figure 3.11: Contact curve

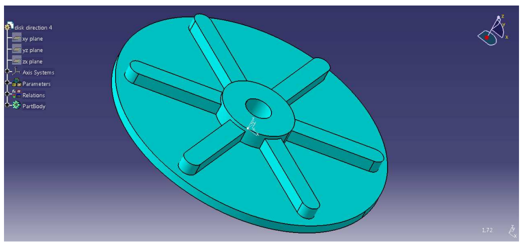


Figure 3.12: Cam disk with 6 cams

3.2.5 Follower

The follower has tho order to transmit the swinging movement to the direction wheel shaft. It was designed as a simple part without a particular geometry.

3.2.6 Ratchet

The ratchet is a mechanism that consists of two principal parts, a cogwheel and a plectrum. It is in charge transforming the swinging movement in a discontinuous rotation movement. In other words, the ratchet permits the turn in one direction and not in the opposite. This rotation causes the variation in the automobile trajectory.

The cogwheel has been designed trying that the minimum angle was as smaller as possible, figure 3.13. The plectrum has the same geometry that the teeth of the wheel, figure 3.14. Both have been sketched in the *Part Design* modulus.

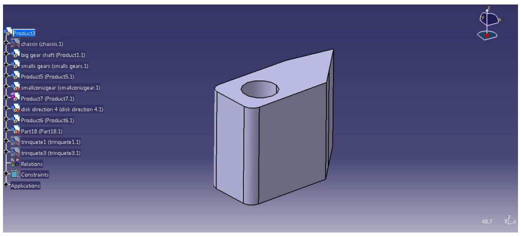


Figure 3.13: Plectrum.

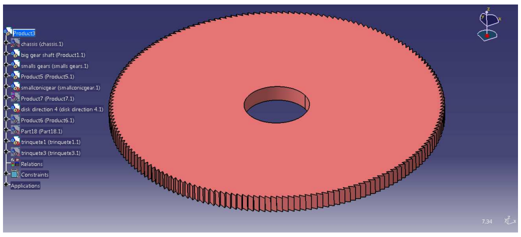


Figure 3.14: Cogwheel.

3.2.7 Shafts

The shaafts have been designed using only the *Part Design* modulus. The dimensions are the required for all the mechanism as a whole.

3.3 Assembly

This is the last step before the kinematics analysis. To do the assembly has been used the *Assembly Design* modulus. The process was to make the assemblies including part or subsets. Between this components has been estalished constrains, so that the movement follows certain rules about his performance.

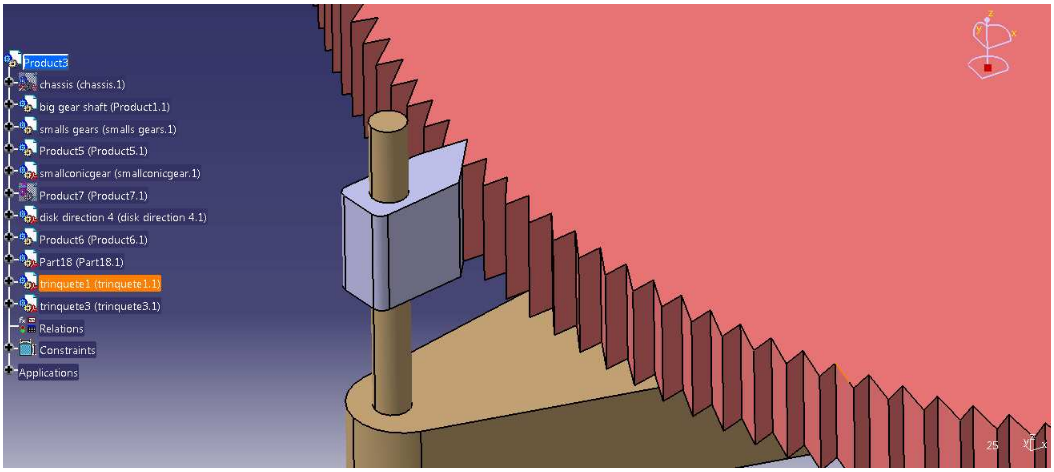


Figure 3.15: Assembly ratchet.

The first part that has been included in the assembly was the one considered as fix, it has been defined as chassis.

When the chassis is fixed, the spur geras are included as subsets, that is, the gears are with their shafts, figure 3.16. These have been positioned with the opportune constrains.

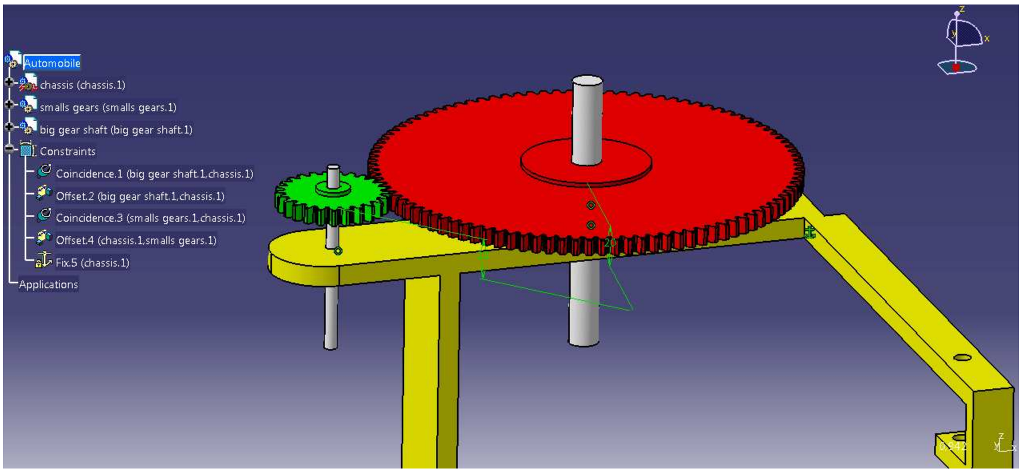


Figure 3.16: Assembly spur gears.

The next step was to include the bevel gears. As made in the spur gears, constricting the opportune movements, figure 3.17.

All the parts and subsets have been referenced in the same part, the chassis, here is important to do the simulations for the kinematics analysis.

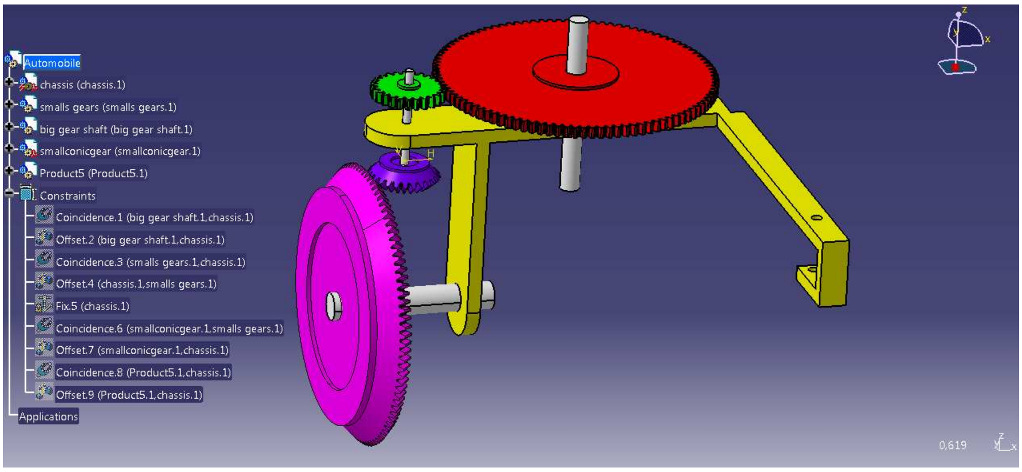


Figure 3.17: Assembly bevel gears.

The next parts are the cam disk and the follower, figure 3.18.

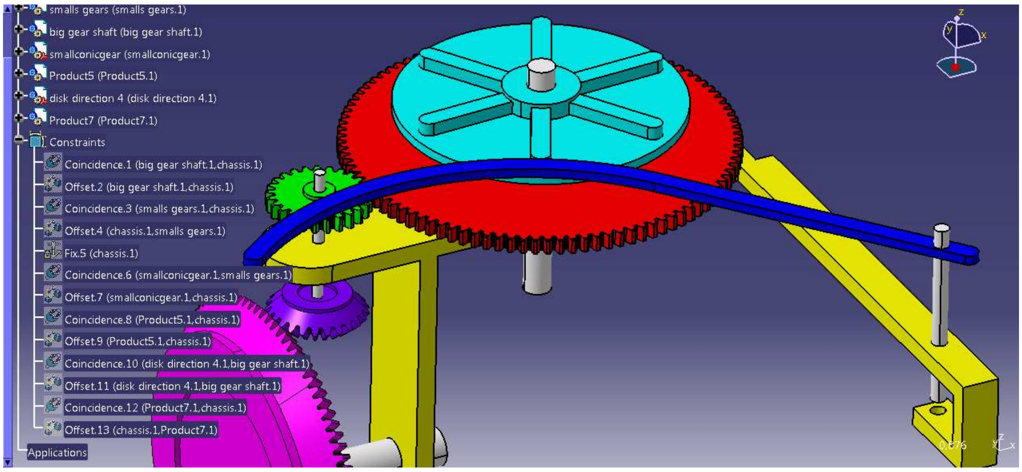


Figure 3.18: Assembly cam disk and follower.

To finish the assembly, the last step is including the ratchet couples in the direction wheel shaft, figures 3.19 3.20.

Once the mechanism is finished some of the movements could be simulated.

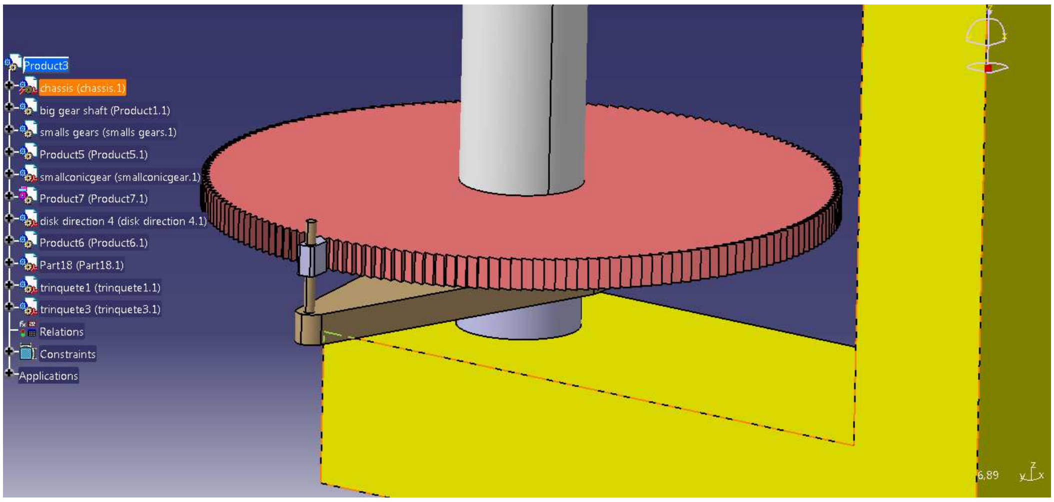


Figure 3.19: Assembly ratchet.

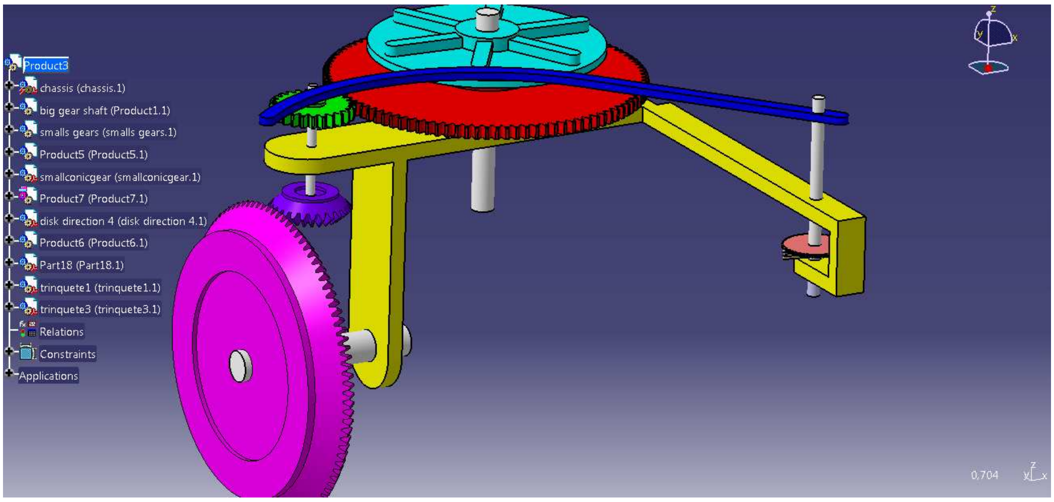


Figure 3.20: Assembly complet.

Chapter 4

Kinematics analysis

4.1 Introduction

This chapter shows the kinematics analysis. To do this analysis the mechanism must be correctly assemblied. For this work the *DMU kinematics* modulus was used, this is an independent CAD product, dedicated only to simulate mechanisms.

To do the analysis is necessary to define the articulations, and the relations between them and between the different parts. The automobile has been separated in three different mechanisms for faciliting the simulation.

4.2 Transmission

The first mechanism to analyse is the gear train, responsible for transmitting the motor power to the motor wheels, as has been mentioned in the other chapters the transmission ratio is one to one. The angle of entry, motor angle, is φ_1 and the angle of exit, the wheel angle, is φ_2, the resultant equation is equation 4.1. The linear velocity modulus, knowing that the wheel radius is $\boldsymbol{R}_w$, is equations 4.1 4.11. The direction of this velocity will be defined with the direction mechanism.

$$\varphi_1 = \varphi_2 \tag{4.1}$$

$$\boldsymbol{v} = \omega_2 \cdot \boldsymbol{r} \tag{4.2}$$

4.3 Direction

The direction has been separated in two mechanisms; the first one is composed of the cam disk and the follower, and the second one of the ratchet.

If the angle in the follower named φ_3, his variation will be a continuous sequence of parabolas,figure 4.1, which will have less amplitude and less periods if the cam disk has more cams, figure 4.2.

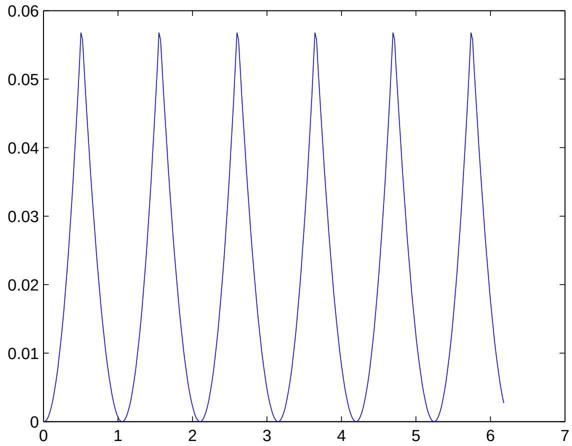


Figure 4.1: Sequence with six cams.

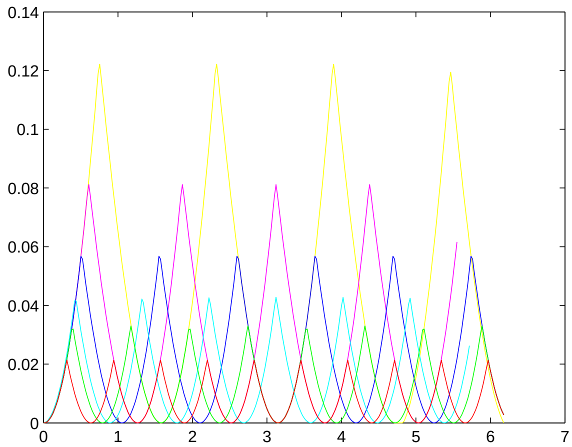


Figure 4.2: Sequence with 4 to 10 cams.

From this evolution the ratchet only use the curve part that has positive slope. When this curve has negative slope the out angle in the ratchet will be constant, figure 4.3.

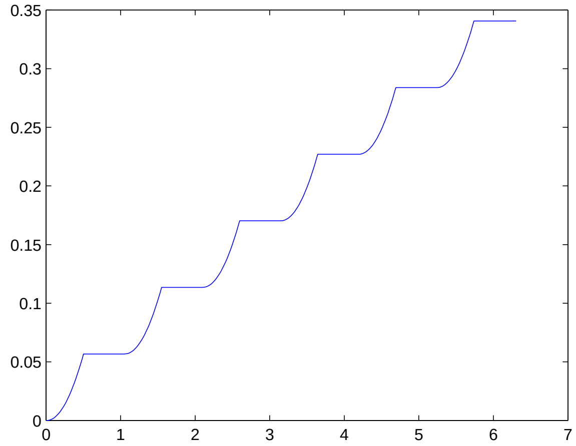


Figure 4.3: Ratchet evolution with six cams.

4.4 Trajectory

The automobile has three wheels, two fronts and one back. This configuration does not allow doing turns that are more than 90 degrees. The instantaneous rotation point is in the line that crosses the two front wheels shafts.

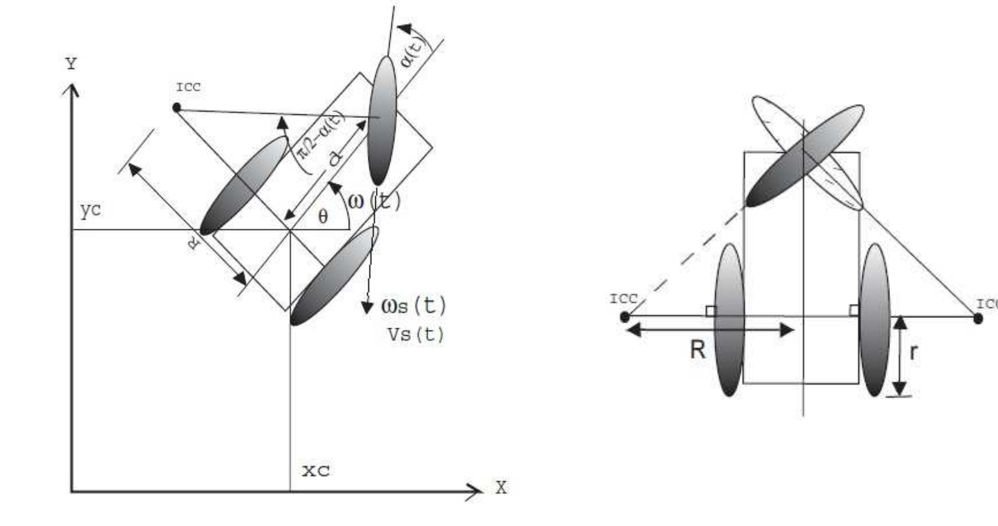
The position is calculated relative to a start point using kinematics [3]. The final equations will be the equations 4.3 4.4

$$\boldsymbol{x} = \boldsymbol{f}_1(\boldsymbol{t}) \tag{4.3}$$

$$\boldsymbol{y} = \boldsymbol{f}_2(\boldsymbol{t}) \tag{4.4}$$

To solve these equations is important to establish the kinematic model, shown in the figure 4.4. The direction wheel has an α angle (out angle in the ratchet). The automobile rotates with a speed omega around the instantaneous rotation point.

The parameters $\boldsymbol{r}$, $\boldsymbol{V}_s$, ω_s, are the radius, linear speed and angular speed from the direction wheel, and d will be the distance between the direction wheel and the front center wheels.



(a) (b)

Figure 4.4: Kimenatic model (a) and (b)

The linear speed from the back wheel is obtained from the equation 4.5

$$\boldsymbol{V}_s(\boldsymbol{t}) \quad = \quad \omega_s(\boldsymbol{t}) \cdot \boldsymbol{r} \tag{4.5}$$

The instant bend radius is calculated with the equation 4.6

$$\boldsymbol{R} \quad = \quad d \cdot \tan(\pi/2 - \alpha) \tag{4.6}$$

ω_s is obtained with equation 4.6, equation 4.7

$$\boldsymbol{S} \quad = \quad d \cdot \omega(\boldsymbol{t}) \tag{4.7}$$

Where $\boldsymbol{S}$, is the length rounded θ. From the derivate of equation 4.7 is obtained, equation 4.8

$$\boldsymbol{V} \quad = \quad d \cdot \omega(\boldsymbol{t}) \tag{4.8}$$

$\boldsymbol{V}$ is the speed component in the back wheel, in the perpendicular axis of the automobile. Obtained the equation 4.9.

$$\boldsymbol{V} \quad = \quad \boldsymbol{r} \cdot \omega_s(\boldsymbol{t}) \cdot \sin(\alpha) \tag{4.9}$$

Combining the equation 4.5 with equation 4.9 and finding the value ω, equation 4.10

$$\omega(\boldsymbol{t}) \quad = \quad \frac{\boldsymbol{V}_s(\boldsymbol{t}) \cdot \sin(\alpha(\boldsymbol{t}))}{d} \tag{4.10}$$

The automobile speed equation is 4.11

$$\boldsymbol{v}(\boldsymbol{t}) = \boldsymbol{V}_s \cdot \cos(\alpha) \tag{4.11}$$

The position equations are 4.12 and 4.13

$$\dot{x} = \boldsymbol{V}_s \cdot \cos(\alpha) \cdot \cos(\theta) \tag{4.12}$$

$$\dot{y} = \boldsymbol{V}_s \cdot \cos(\alpha) \cdot \sin(\theta) \tag{4.13}$$

To know the automobile position, the previous equation has been integrated, equations 4.14, 4.15 and 4.16.

$$\boldsymbol{x}_c = \int \boldsymbol{v}(\boldsymbol{t}) \cdot \cos(\theta(\boldsymbol{t})) \cdot \mathrm{d}\boldsymbol{t} \tag{4.14}$$

$$\boldsymbol{y}_c = \int \boldsymbol{v}(\boldsymbol{t}) \cdot \sin(\theta(\boldsymbol{t})) \cdot \mathrm{d}\boldsymbol{t} \tag{4.15}$$

$$\theta_c = \int \omega(\boldsymbol{t}) \cdot \mathrm{d}\boldsymbol{t} \tag{4.16}$$

For the automobile case. The angle equation has two different parts. One that has a dependency on ϕ_1 and the other one is a constant function.

The information obtained in the simulation is a group of points, to have a dependence between ϕ_1 and the time, has been supposedthat one cycle is two seconds, so the phi1 dependence with the time is 4.17, only in the case that the cam disk has 6 cams.

$$\phi_1 = 0.5236 \cdot \boldsymbol{t} \tag{4.17}$$

To simplify the problem, the direction wheel diameter is the same that the motor wheels. The angular speed will be the same, equation 4.18

$$\omega_s = 0.5236 \cdot rad/s \tag{4.18}$$

The alpha angle dependence with the time is, equation 4.19 and 4.20

$$\alpha = 0.0637 \cdot \boldsymbol{t}^2 \qquad \forall \boldsymbol{t} \in (\boldsymbol{n}, \boldsymbol{n}+1) \tag{4.19}$$

$$\alpha = 0.0637 \cdot (\boldsymbol{n}+1)^2 \qquad \forall \boldsymbol{t} \in (\boldsymbol{n}+1, \boldsymbol{n}+2) \tag{4.20}$$

$$\forall \boldsymbol{n} \in N$$

Once all the dependences with the time have been known, the position problem can be solved.

The problem was that the curvature angle could not be calculated, because alpha has a quadratic dependency with time. So the solution was to approximate the points with a linear curve. Obtaining equations 4.19 and 4.20, that are less precise, equations 4.21 and 4.22.

$$\alpha = 0.592 \cdot \boldsymbol{t} \qquad \forall \boldsymbol{t} \in (\boldsymbol{n}, \boldsymbol{n}+1) \tag{4.21}$$

$$\alpha = 0.592 \cdot (\boldsymbol{n}+1) \qquad \forall \boldsymbol{t} \in (\boldsymbol{n}+1, \boldsymbol{n}+2) \tag{4.22}$$

$$\forall \boldsymbol{n} \in N$$

This approximation has other problem; the position equation could not be integrated if the α angle is dependent on time. The trajectory equation can be only calculated if the angle α is not dependent on time.

Solving with the data from the table 4.1, equations 4.23 and 4.24 are obtained.

Table 4.1: Parameters

Parameter	Name	Value
ω_s	angular speed	0.5236 rad/s
$\boldsymbol{r}$	Radius	110 mm
∂b	Distance between front wheel and back wheel	210 mm

$$\boldsymbol{x}_c = 38.654 \cdot \sin(0.745 \cdot \boldsymbol{t}) + 65.56 \cdot \sin(0.439 \cdot \boldsymbol{t}) \tag{4.23}$$

$$\boldsymbol{y}_c = 65.6 \cdot \cos(0.439 \cdot \boldsymbol{t}) - 38.655 \cdot \cos(0.745 \cdot \boldsymbol{t}) \tag{4.24}$$

This is the first constant part, to the next parts will be the same calculus. The other part can be approximated represented. So the approximated trajectory is, figure 4.5

With different number of cams in the cam disk, figure 4.6

The trajectory is more linear if the cam disk has more number of cams.

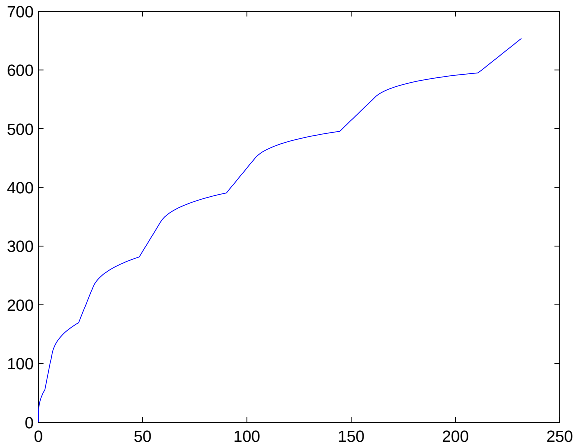


Figure 4.5: Trajectory defined with 6 cams.

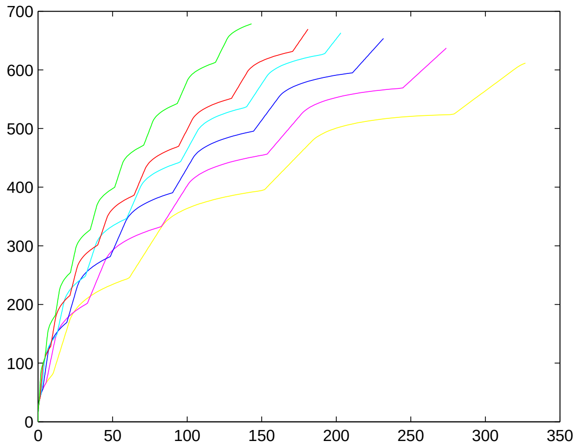


Figure 4.6: Trajectory defined with 4 to 10 cams.

Chapter 5

Conclusions

Once having modeled and built the mechanism in the way reported along the text, the next conclusions can be extracted:

- The movement of the mechanism fits to the expected working process of the automobile described in the wording, despite the result is not real.
- Before the design of the automobile had been important to have dimension to facility the mechanism design. The documents consulted has not it. The Leonardo's sketchs were not detailed.
- The software used allowed to study the movement of the model, obteined experimental data, to do the final analysis.
- This design might not be suitable for obtaining Leonardo's expectations, an automatic machine, but make it possible to design other simple mechanism with the same end.

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Erklärung

Hiermit versichere ich, dass ich

- die vorliegende Arbeit selbständig verfasst habe,
- keine anderen als die angegebenen Quellen benutzt und alle wörtlich oder sinngemäß aus anderen Werken übernommenen Aussagen als solche gekennzeichnet habe,
- die eingereichte Arbeit weder vollständig noch in Teilen bereits veröffentlicht habe und
- dass das elektronische Exemplar mit den anderen Exemplaren übereinstimmt.

Datum	Unterschrift