Design and analysis of the chassis of an electric kart

Bachelor's degree final project

Bachelor's Degree in Mechanical Engineering

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3. Product conditions

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1.Report

1.1. Project objective

The objective of this project is to design a chassis for an electric kart for amateur drivers. The idea is to make a structure for a kart that can be used by the amateur drivers to build its kart or for companies that rent these karts in a circuit.

This project is going to be focused only in the design and analysis of the structure of the kart; that means that the selection of the different components that form a kart is excluded of this project.

So, at the end of the project the final solution must be a chassis that can be adopted for de customers to build its kart, to enjoy the sport of the karting or to compete in amateur races. It could also be used by kart renting companies if they want to make its own karts, to rent them in their circuits.

1.2. Precedents

Since that sport is appeared in the second half of the last century, the internal combustion engines have been used; but, in the last years, the develop of new batteries technology, and its application in electric cars has done that the use of electric motors as an interesting alternative.

Also, the construction of indoor circuits in the cities has increased the electric karts market; for this reason, in 2016 the FIA, with Mach 1 and Bosch, presented its electrical Kart, and the technical regulations for electric karts.

In comparison with the other components of the electric kart (tyres, brakes, direction system…), which are similar or identical to those of a combustion kart, the batteries, the engine, and the chassis are different. To find batteries and engines that can give us the performance that we are looking for, it is more or less easy, because the batteries, for example, could be the same that are used in the combustion kart, for start the engine, but in a higher quantity; or use batteries that are being used in the little electrical cars. The case of the engine is like to that of batteries, in the market we can find engines that cover our needs.

But, if we try to find a chassis, we could find a chassis for a combustion kart; but this is different to that it is used in al electric kart because the distribution of masses is different. For this reason, I have decided to design one for this kind of karts.

1.3. Factors to consider

1.3.1. Technical regulations

The technical regulations used to develop this project are the "Technical Regulations for Electric Karts (E-Karting)" of 2019. Them are going to be used to determine the material, dimensions, mass of the kart and batteries, sections and unions between the different parts of the chassis.

1.3.2. Actual market

In order to define some of the characteristics of the kart, like the common power and torque of the engine; and to have an idea about the general dimensions, the weight, and the performance of the kart, an analysis of the marked has been made:

A) Characteristics:

Company	Model	Group	Engine			
			Power [kW]		Torque [Nm]	
			Nominal	Peak	Nominal	Peak
SPEED 2 MAX	SLC	1	6	17		30
BIZ KARTS	ECOVOLT NG+	2	10,5		45	
SPEED 2 MAX	SXP		6	17		30
Caroli Kart	Stinger Electric		5,5	12,5	17	40
Rimo Germany	SINUS ion		5,6	23		
Tom Kart	Go-Kart			16		
OTL Electro Kart	Race Kart			23		
ADVEO	EK16			23		
Birel ART	E-Kart			20		57

Table 1: Karts in the market. Engines [3-12]

Company	Model	Batteries			
		Type	Voltage [V]	Charge capacity [Ah]	Energy [kWh]
SPEED 2 MAX	SLC	Lead	12,00	48,00	0,576
BIZ KARTS	ECOVOLT NG+	Li-Ion	51,20	58,59	3,000
SPEED 2 MAX	SXP	Li	12,00	60,00	0,720
			12,00	100,00	1,200
Caroli Kart	Stinger Electric	LiFePO4	48,00	60,00	2,880
Rimo Germany	SINUS ion	LiFeMnPO4	48,00	40,00	1,920
				60,00	2,880
				120,00	5,760
Tom Kart	Go-Kart	LiFePO	50	40	2,000
OTL Electro Kart	Race Kart	LiPO	80,00		
ADVEO	EK16	Li			2,000
					3,000
Birel ART	E-Kart	Li-Ion	102,00	35,29	3,600

Table 2: Karts in the market. Batteries [3-12]

Company	Model	Kart			
		Weight [kg]	Long [mm]	Width [mm]	High [mm]
SPEED 2 MAX	SLC	200	2100	1350	800
BIZ KARTS	ECOVOLT NG+	200	1900	1310	600
SPEED 2 MAX	SXP	140	2100	1350	800
Caroli Kart	Stinger Electric	195	1920	1300	600
Rimo Germany	SINUS ion	186	2200	1390	600
Tom Kart	Go-Kart	182	1920	1400	1150
OTL Electro Kart	Race Kart	120			
Birel ART	E-Kart	165			

Table 3: Karts in the market. Dimensions [3-12]

B) Performance:

Company	Model	Max speed [km/h]
SPEED 2 MAX	SLC	70 (Speed limitation in France)
	SXP	
Tom Kart	Go-Kart	65
OTL Electro Kart	Storm EFD	72
	Race Kart	102

Table 4: Karts in the market. Performance [3-12]

1.4. Alternative solutions

As it is said before; in the point 1; in this project, only the chassis is being studied; therefore, alternative solutions are not to be given for the engine, batteries, direction system; and the other components.

1.4.1. Material

As it is said in the FIA regulations, only magnetic structural steel or structural steel alloy meeting the ISO 4948 classifications and the ISO 4949 designations is allowed; for this reason, aluminium or carbon fibre are disdained. Some useful steels are:

A) AISI 4130:

One of the most common steels that is used to make the chassis of the kart is the AISI 4130 which properties are this:

Tensile strength [MPa]	530 to 1040
Yield strength [MPa]	440 to 980
Elongation [%]	13 to 26
Carbon content [%]	0,28 to 0,33
Phosphorus content [%]	0 to 0,035
Sulphur content [%]	0 to 0,04
Nitrogen content [%]	0

Table 5: AISI 4130 properties [13]

B) S-450 J0

Other structural steel that have some interesting properties is the S450J0 which properties are:

Tensile strength [MPa]	550 to 720
Yield strength [MPa]	450
Elongation (min) [%]	17
Carbon content (max) [%]	0,2
Phosphorus content (max) [%]	0,035
Sulphur content (max) [%]	0,035
Nitrogen content (max) [%]	0,025

Table 6: S-450 J0 properties [14]

As AISI 4130 steel is one of the most common for this application, and steel S450J0 can be difficult to find, AISI 4130 steel is selected. Its facility to be welded also makes it an interesting solution.

1.4.2. Unions

As it is said in the technical regulations, the unions between the tubes of the chassis must be made by welding. There are different types of welding:

A) Arc welding:

Arc welding use electrodes and energy to create an electrical arc between the electrode and the materials; these are melted, and when them are cooled they are fuse together. Arc welding is divided in consumable and non-consumable methods. This are the kinds of arc welding:

A1) Stick Welding (SMAW or Shielded Metal Arc Welding):
Although it could be considerate archaic nowadays, is an economical process because it requires a minimum of equipment. In this process uses welding sticks made up of filler material and flux. They flux protect the molten metal; and the filler is used to join the pieces of material.

In spite of it is an economical process, sometimes its results are not the adequate, because it can suffer from porosity, shallow penetration, cracking and it is highly vulnerable to severe weather and is generally less durable.

A2) MIG Welding (GMAW or Gas Metal Arc Welding):
In this process a cable is used, which is protected by an inert gas, connected to an electrode to join the metal parts. This process is one of the most used because it does not require a high level of precision to obtain acceptable results.

This process is sensible to external factors such as wind and dust; therefore, the operator has to control voltage and wire speed in order to obtain the expect result; if not, some quality problems may appear. The most common are dross and porosity. If these are not taken care of properly, the results could be worst tan the result of a TIG welding.

MIG is easy to master than TIG to the operator because in the first one the operator only has to use one had, to guide the welding gun. In the other one, the operator has to use the two hand to make the process.

A3) Flux-Cored Arc Welding (FCAW):
Flux-Cored Arc Welding is quite similar to MIG. The differential fact is that in FCAW special tubular wire with flux are used, and in some situations the shielding gas is not needed. Its best advantages are that is extremely inexpensive, easy and fast to make, and it can be realised outdoor; its main disadvantage is that the results are not aesthetically pleasing.

A4) Submerged Arc Welding (SAW):
The main characteristics of this process are its minimal emissions of welding fumes and arc lights making it safer than other processes. This fact it is due that all the process is make beneath a blanket of granular fusible flux consisting of silica, calcium fluoride, lime, manganese oxide, and other compounds. Due that the process it is make beneath of that blanket, the welding is protected from ultraviolet and infrared radiation; habitual in that process. This process involves a minimal preparation, making it an efficient and quick process.

A5) TIG Welding (GTAW or Gas Tungsten Arc Welding):

TIG Welding use a non-consumable tungsten electrode and inert gas (usually argon). Tungsten in a rare and hard element that offers a purity and high-quality weld. Although the result could be affected by the ability of the operator; it is the most used process because it offers a high precision level and a greater control over the weld area. It is generally used in the manufacturing of vehicles, tubes and bikes. It could be used to weld of non-ferrous metals too.

A6) Electrostatic Welding (ESW):

The electric arc is struck by wire and then fed into the welding puddle along with flux until the slag reaches the electrode and extinguishes the arc. It is used on thick and non-ferrous metals; it offers a very efficient, single-pass welding. Generally, is used in maritime and aerospace industry.

A7) Atomic Hydrogen Welding (AHW):

The technique involves placing two tungsten electrodes in a hydrogen atmosphere; hydrogen molecules are broken and recombined, increasing the temperature up to 3000ºC. This temperature permits weld tungsten, providing strong and cohesive welds.

This process is slowly being replaced by GMAW process.

A8) Carbon Arc Welding (CAW):

The process welds metals together by heating them with a non-consumable carbon electrode, heating them to temperatures in excess of 3000ºC.

This process, nowadays, is practically replaced by more efficient and save methods.

A9) Electrogas Welding (EGW):

Electrogas Welding is similar to Electroslag welding in that the metal being welded is struck with a consumable electrode, without the use of pressure. In that case the arc is not extinguished. It is used in the shipbuilding and storage tank industries.

B) Gas Welding:

Also known as Oxyfuel welding or oxyacetylene welding, is the process of combining fuel gasses and pure oxygen in order to increase the welding torch's flame temperature to around 350ºC. It is used in industries like pipe and tube welding.

C) Resistance Welding:

This process involves applying force to adjacent surfaces and then applying an electric current near and upon the surfaces, generating intense heat. There are many variations of this technique, namely spot welding, seam welding, butt welding, flash welding, projection welding, and upset welding.

D) Energy Beam Welding (EBW):

It is done in a total vacuum, firing a beam of high-velocity electrons. This kinetic energy is transformed in heat that melts the welding materials and fuse them together when they are cooled. The most habitual processes are laser beam welding and electron beam welding.

E) Solid-State Welding:

This process relies on join the materials without significantly melting them. There are many variations of solid-state welding, including ultrasonic welding, explosion welding, friction welding (including friction stir welding), magnetic pulse welding, co-extrusion welding, cold welding, diffusion bonding, exothermic welding, high-frequency welding, hot pressure welding, induction welding, and roll welding.

On my view, TIG welding is the process that offers good results, and like is one of the most common methods, it is going to be more economic than others that offer similar results; for this reason, this process is chosen.

1.4.3. Size and tubes disposition

The FIA technical regulations for electric carts said that the chassis has to be made by circular tubes, so only the size and the tube distribution could be chosen.

Another interesting point is to know which type of effort the tubes have to support. The main effort is a bending moment; so, to determinate if a tube is going to support the effort, its main characteristic parameter is the elastic section modulus (S or W, in that project I am going to uses W)

For a circular tube this parameter it is calculated like:

$$W_y = W_z = \frac{\frac{1}{4} \cdot \pi \cdot (R^4 - r^4)}{R} \quad (4.1)$$

$$r = R - t \quad (4.2)$$

R = external ratio

r = internal ratio

t = thickness

The normal diameter in the karts chassis is about 30 mm; but to analyse other options, as they may be necessary, tubes up to 50 mm could be used. About the thickness, the less thickness that is going to be uses is of 2 mm, in order to obtain good results in the welding processes; the maximum that could be considered as a real option is 5 mm. The module of the most common tubes that can be on the market are these:

D (mm)	30	30	30	40	40	40	50	50	50
R (mm)	15	15	15	20	20	20	25	25	25
t (mm)	2	3	5	2	3	5	2	3	5
r (mm)	13	12	10	18	17	15	23	22	20
A (mm^2)	176	254	393	239	349	550	302	443	707
W (mm^3)	1155	1565	2127	2161	3003	4295	3480	4912	7245

Table 7: Common tubes. Dimensions

After knowing these parameters two possible distributions could be proposed; one with more tubes, but smallest; other one, with less tubes, but biggest. The first option gives a most rigid structure in the middle and more deformable in the front of the kart. But heavier and with more welds, them could be conflictive points. The second one, give a lighter structure with less welds; but it could be that the chasses deforms too much in the middle, and less than it is required to increase the comfort during the race. This is said because this vehicle cannot wear socks-absorbing; due this, if the structure is deformable; in the front, the comfort is going to increase. This is analysed in the point 1.8.7. of the calculus annex.

A) Fist option: tubes of 30x2 mm

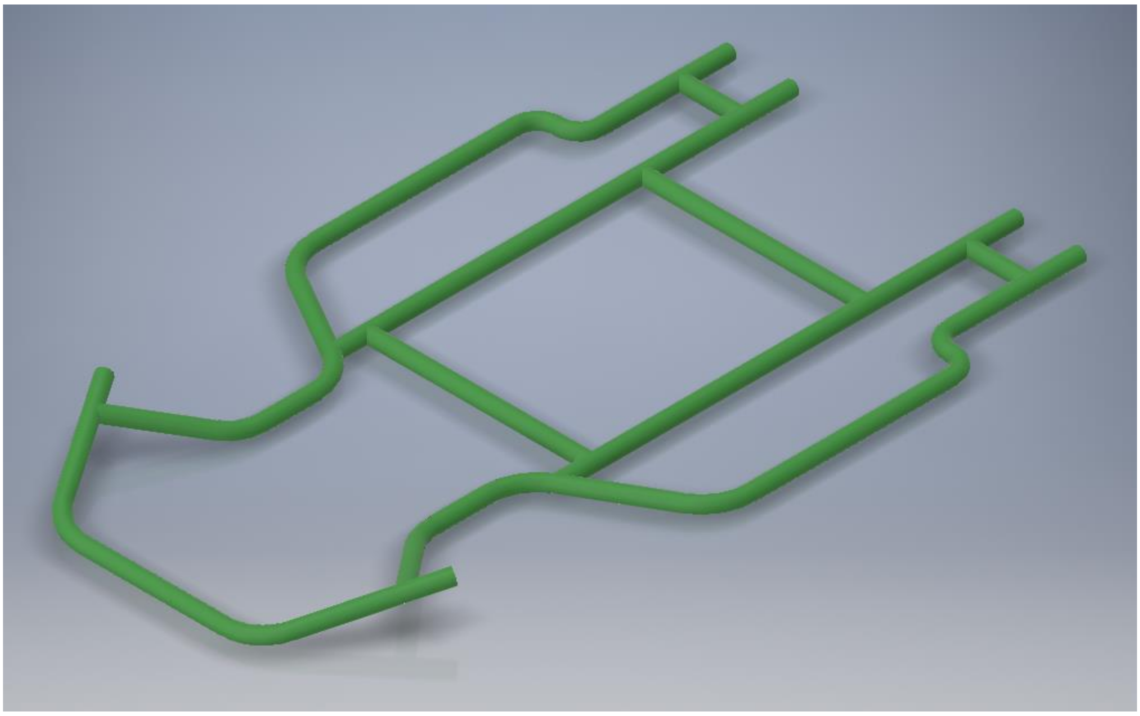


Figure 1: 30x2 model

B) Second option: tubes of 50x2 mm

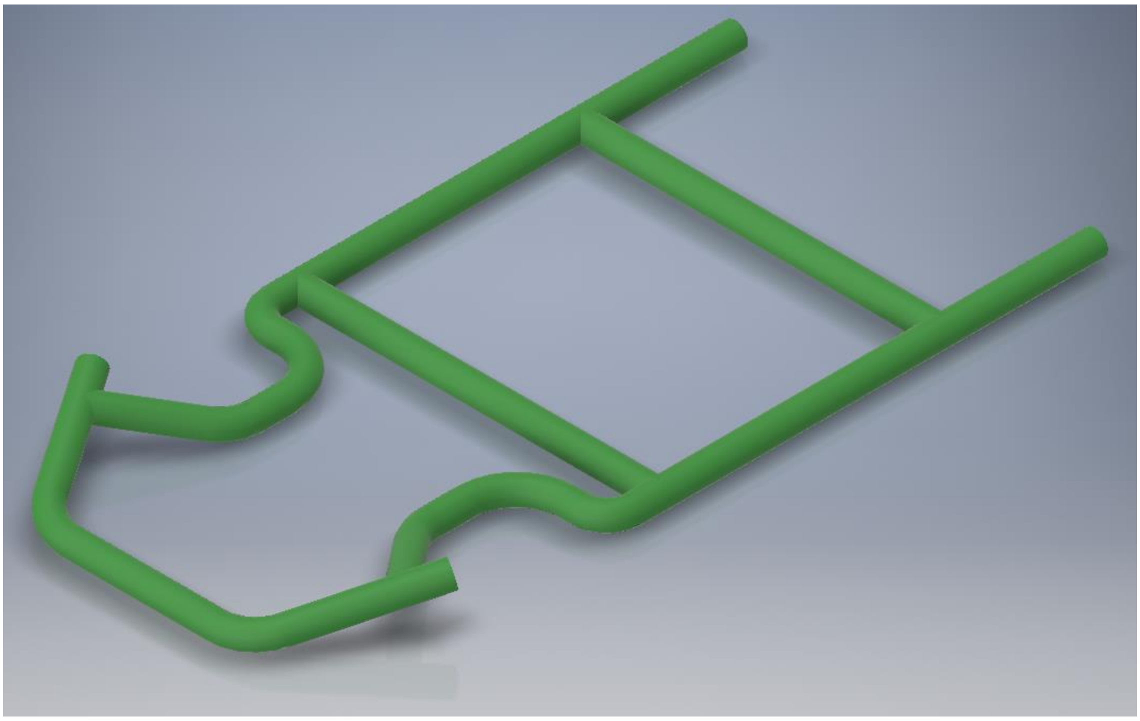


Figure 2: 50x2 model

Like the welds are going to be make in a factory, there are not to be a lot of mistakes in this process, and in order to obtain a les deformable structure, overall in the central part, where the driver is, the first option is chosen.

1.5. Final solution adopted

The final solution is a tubular structure made with tubes of 30 mm of diameter and 2 mm of thickness. Them are joined using a process of TIG welding, with weld of 1,5 mm.

The steel used is normalized AISI 4130; with the following characteristics:

Mechanical properties [MPa]		Alloy composition [%]	
Elastic Modulus	190000	Chromium (Cr)	0,8-1,1
Fatigue Strength	320	Manganese (Mn)	0,4-0,6
Ultimate tensile Strength	690	Silicon (Si)	0,15-0,35
Yield tensile Strength	440	Molybdenum (Mo)	0,15-0,25
		Sulphur (S)	0-0,04
		Phosphorus (P)	0-0,035

Table 8: Normalized AISI 4130 properties

The final structure has a mass of 17,7 kg including the bumpers. These bumpers are made with the same steel and joined to the chassis using bolts of M8 (the technical regulations admit a minimum diameter of 6 mm). The bumpers and the chassis are designed following the dimensions give in the draws that are in the regulations in order to them can obtain the CIK-FIA homologation. They could be seen in the E.2.00.A drawing.

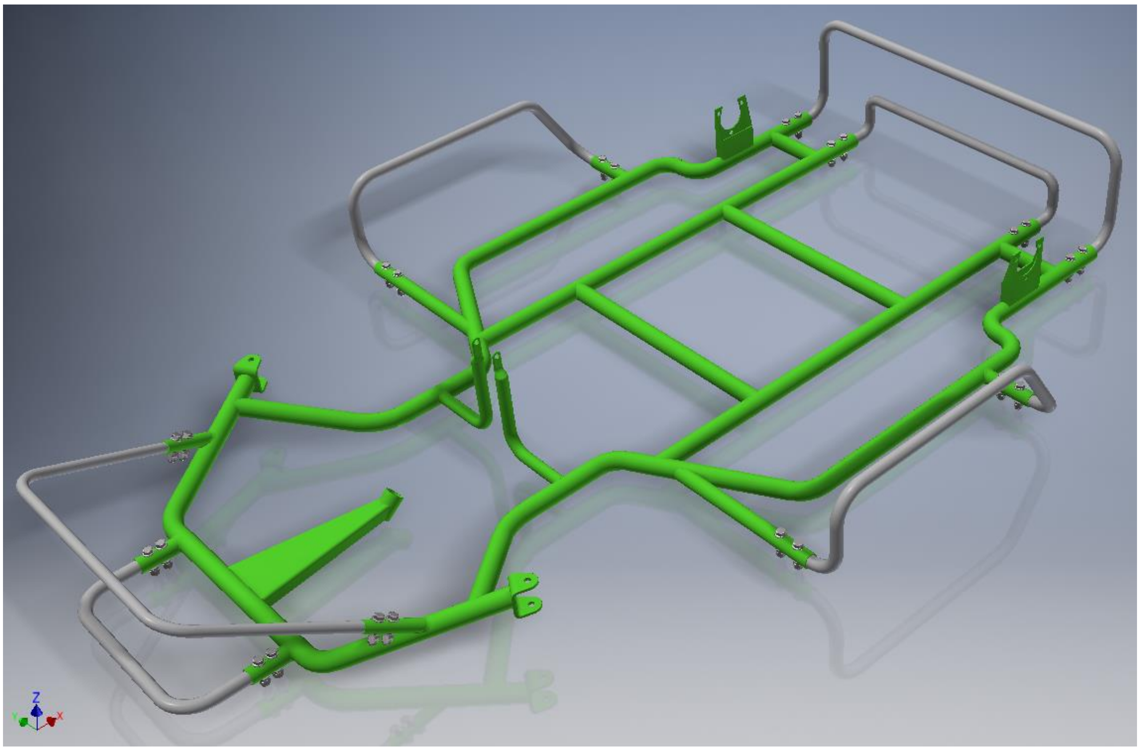


Figure 3: Chassis final model

For more details of its dimensions see the plans.

1.6. Final solution justification

To be sure that this structure is going to support the efforts due to the acceleration, deceleration and turn; the weight, and the vibrations derived of the irregularities of the circuit a model 3D of a kart is made.

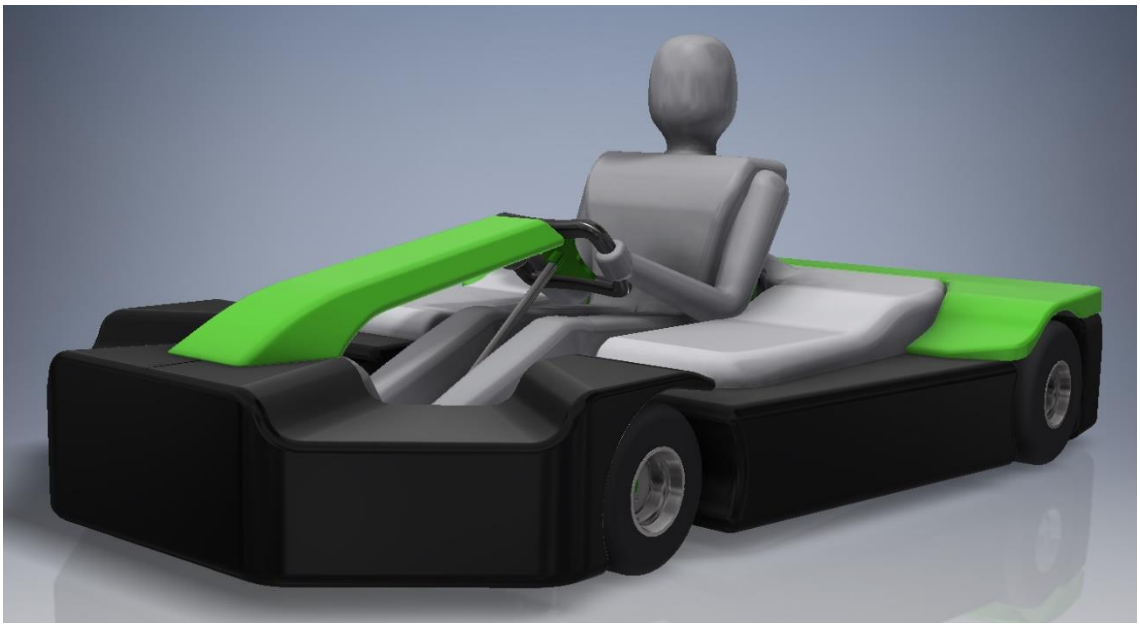


Figure 4: Kart model

With this model the centre of gravity is obtained. This data is used in the calculus of the forces. Other important date is the mass; so, between the range of masses that the technical regulations permit; in the annex of calculus it is analysed which is the most critical for the structure. With this datum, and the characteristics of the engine; them are a mix between the datum in the table 1; and a little analysis made in the point 1.8.2.

Once that these values are known the maximum accelerations are calculated; the maximum acceleration (3,5 m/s²); the maximum deceleration (13 m/s²); and the maximum lateral acceleration (15 m/s²). Other cases are analysed, but there not give important results; for more details check the annex of calculus (points 1.8.3 to 1.8.7).

1.6.1. Forces and deformations

The forces produced by these accelerations are calculated and introduced into the model; obtaining the following results:

Case		Max σ [MPa]	Xf [-]	Xy [-]	Max. deformations [mm]		
					X	Y	Z
Acceleration	3,5 m/s²	113,3	2,82	3,88	0,41	0,15	2,58
	10,5 m/s²	94,0	3,40	4,68	0,54	0,32	2,02
Deceleration	13 m/s²	232,5	1,38	1,89	0,65	0,29	4,61
	7,5 m/s²	269,4	---	1,63	1,08	0,22	4,81
	6 m/s²	180,0	---	2,44	0,32	0,17	3,74
Turn	15 m/s²	239,9	1,33	1,83	0,88	5,78	3,02
Critical		269,4	1,33	1,63	1,08	5,78	4,81

Table 9: Results of the tensional and deformations analysis

As it can see in the previous table (table 9) the critical situation is the turn; and in that case the coefficient of fatigue is of 1,33. That means that the tension that the chassis could suffer before to fail to fatigue is a 33% highest that the real tension. At elasticity this coefficient increases to 1,63. In general a design is considered like good when this coefficient is between 1,5 and 2 for the critical situation. So, it could be considered that these results are acceptable.

Looking now to the deformations, it can be seen that in the vertical axis (Z), the maximum deformation is in the deceleration case; with a value of 4,81 mm, under to the 5 mm that was the limit introduced in the calculus; due that some elasticity and deformation is searched in order to absorb the irregularities of the asphalt. The rest of the displacements are not some important because there is not a physical limit; like the floor in the vertical axis.

The last important value in the tensions and deformations is the torsional one; giving a result of 140 Nm/º. This value could be a little small, but if the structure is changed in order to increase it, the tensions are going to be increased to; overall in the turn case, where the coefficient to fatigue is near the limit; so it is better to have a structure a little more rigid; than to have a more elastic chassis, but that it could be broken.

1.6.2. Vibrations

To end the design a little analysis to obtain the natural frequencies of the chassis has been made obtaining the following results:

Mode		f [Hz]
Static	1	8,00E-06
	2	2,00E-05
	3	2,50E-05
	4	2,60E-05
	5	2,70E-05
	6	2,90E-05
Dynamic	7	28,4
	8	35,3
	9	52,1

Table 10: Chassis Natural Frequencies

The natural frequencies of the static modes are nearly 0 Hz; something that was expected; and the natural frequencies of the dynamic modes are high enough to not suffer resonance.

Some clarification; these values are obtained with a medium value of the mases. It is practically impossible that the final kart; so; if during the test of a kart that use this chassis some problem derivate of the resonance problem is observed there are two easy solutions; the first one is changing the tyres and use a rigid ones; in order to increase the natural frequencies and go out of the range of frequencies due to the irregularities of the circuit. The second one is or reduce the mass of the batteries; changing its technology; or reducing its capacity. This is going to reduce the autonomy; but it may solve the problem; or if the mas is lower than the maximum one; some loads could be attached to the chassis, in order to change the mass and the natural frequencies.

1.7. Others

In this final part of the report there is a planification of the time that every part of the project has to last; and an initial budget; that is going to be developed in depth in the point 4 of this project.

1.7.1. Planification

In the following table every rectangle represents one week; and the colour in these rectangles means that this part of the project was made during this week.

	March					April				May					June				July	
Project Objectives																				
Precedents																				
Factors to Consider																				
Alternative Solutions																				
Final solution Adopted																				
Final Solution Justification																				
Other Factors																				
Determination of the parameters																				
3D Model																				
Forces Calculation																				
Tensions and Deformations																				
Vibrations																				
Technical Drawings																				
Product Conditions																				
Budged																				

Table 11: Planification

1.7.2. Initial Budget

In this case for any component of the final product an approximate price is given:

Component	Price
Chassis	450 €
Frontal Bumpers (2)	100 €
Side Bumpers (2)	100 €
Back Bumpers (2)	100 €
Total	750 €

Table 12: Initial Budged

1.8. Calculus annex

In this annex are detailed the calculus done to obtain the tensions and the deformations of the chassis, and its natural frequencies.

1.8.1. Mass

In that case the minimum and the maximum mass of the kart it is indicated in the technical regulations (point 2.4.2); were the karts are divided in two groups in function of them batteries; obtaining these weights:

Group	Batteries technology	Min. Weight [kg]		Max. Weight [kg]
		Only Kart	Kart+battereies+driver	Kart+battereies+driver
1	Pb-Acid Ni-Fe	80	230	270
2	Ni-Zn Ni-MH Li-Ion Li-Me-Po	80	175	232

Table 13: Kart masses [1]

At these point the mass of the batteries and the driver have to be defined; because they are the principal loads of the kart. In that case I am going to restrict the weight of the driver at 100 kg. So, if the mass of the pilot is 100 kg the mass of the batteries has to be:

Group	Batteries	
	Min. Weight [kg]	Max. Weight [kg]
1	50	90
2	-5	52

Table 14: Batteries weight

It is obvious that a negative mass it is not possible; so if someone would use this chassis to make its kart; and obtain the minimum weight possible in the group 2; its weight have to be, more or less, of 80 kg; in order to let 15 kg to the batteries. But in that case, I want to project a chassis able to be used in all the range of masses; so, I am going to analyse the chassis for the maximum weight of the first group.

1.8.2. Engine characteristics

As it can be seen in the table 1; part 1.3.2 the peak torque of the motors is lower than 50 Nm; there is an engine with more torque, but as it could be seen in the part E of these annex, 50 Nm of torque is enough for a kart to an amateur driver.

For example, an engine that could be used is the SAIETTA 119R-55; with these characteristics:

SAIETTA 119R-55	
Peak Power [kW]	12
Peak Torque [Nm]	48
Efficiency [%]	94
Voltage [V]	48
Peak Current [A]	290
Weight [kg]	12

Table 15: Saietta 119R-55 characteristics

1.8.3. Maximum speed

In order to guarantee that the previous engine it could be a good option the maximum speed is going to be calculated. To do this, the principal resistances that a vehicle suffer have to be calculated; the aerodynamical, the inclination of the road; and the friction one.

A) Aerodynamical resistance:

It could be calculated like:

$$R_a = \frac{1}{2} \cdot \rho_a \cdot C_x \cdot A \cdot v^2 \quad (\mathbf{8.3.1})$$

$\rho_a = \text{air density} = 1{,}2\ \frac{\text{kg}}{\text{m}^3}$

$C_x = \text{aerodynamical coeficient} = 0{,}5$

The habitual coefficient for a car is between 0,3 and 0,4; but in that case, where we could find some parts uncovered; engine, batteries; chassis; the real coefficient could be around 0,5.

$A = \text{base} \cdot \text{height} = 1{,}34 \cdot 0{,}425 = 0{,}570\ \text{m}^2$

$v = \text{velocity in } \frac{\text{m}}{\text{s}}$

B) Resistance due to the inclination of the road:

It could be calculated like:

$$R_i = m \cdot g \cdot \sin\alpha \quad (\mathbf{8.3.2})$$

$m = \text{mass of the kart}$

$g = \text{gravity} = 9{,}81 \frac{\text{m}}{\text{s}^2}$

$\alpha = \text{inclination}$

In the circuits the inclination usually is small; so, I am going to consider that this resistance is of 0 N.

C) Friction resistance:

It could be calculated like:

$$R_f = m \cdot g \cdot \cos\alpha \cdot (f_0 + f_2 \cdot v^2) \quad (\mathbf{8.3.3})$$

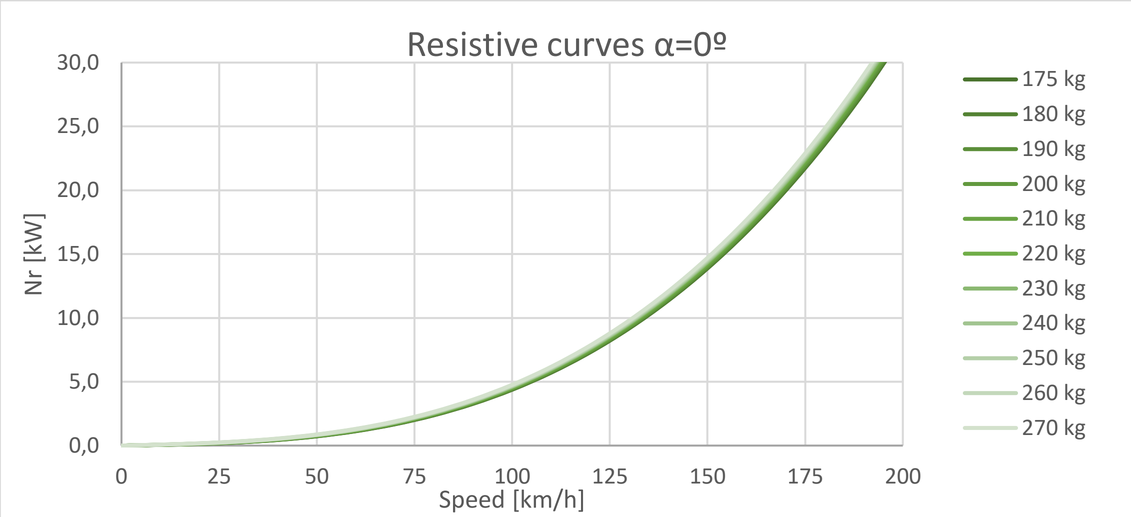
$f_0 = 0{,}01$

$f_2 = 6{,}48 \cdot 10^{-6}$

D) Lost power:

In order to know if the mass is important to determinate the maximum speed is calculated the power lost to beat these resistances in function of the mass. This power could be calculated like:

$$N_r = (R_a + R_i + R_f) \cdot v \quad (\mathbf{8.3.4})$$



Graphic 1: Effect of the mass in the resistances

In order to facility the lecture of the results of this equation, them are represented in the following graphic:

As it can be seen in the graphic the mass does not have a big influence in the lost power; this is because the inclination is of 0º; if not it would have a big influence; due to the increment of the Ri.

Once that the lost power is calculated it can be calculated the maximum speed; considering the:

$\eta_e = \text{engine efficiency} = 94\%$

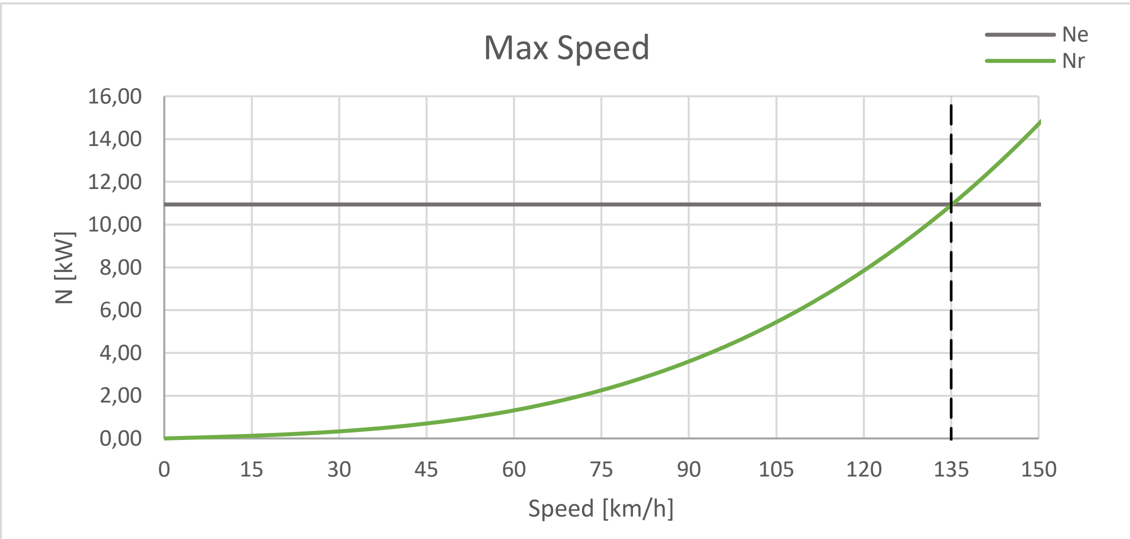
$\eta_t = \text{transmission efficiency} = 97\%$

So, the maximum power that could be expected is:

$$N_e = P_p \cdot \eta_e \cdot \eta_t \quad \mathbf{(8.3.4)}$$

$P_p = \text{Peack power} = 12\ \text{kW}$

With these data the max power obtained is of 10,94 kW; and the maximum speed with this power is of 135 km/h; as it can be seen in the following graphic:



Graphic 2: Max speed

With this datum it could be affirmed that this engine is optimal for the Kart; because the maximum speed is higher than the maximum speeds of the karts that could be found in the market; table 4; part 1.3.2.

1.8.4. Centre of gravity

Once that the engine and the mass are been defined the last parameter to define, to obtain the forces that is going to suffer the chassis, is the centre of gravity. To calculate it a model in Inventor is done; a model including all the important components with them masses and dimensions; and a model of the pilot.

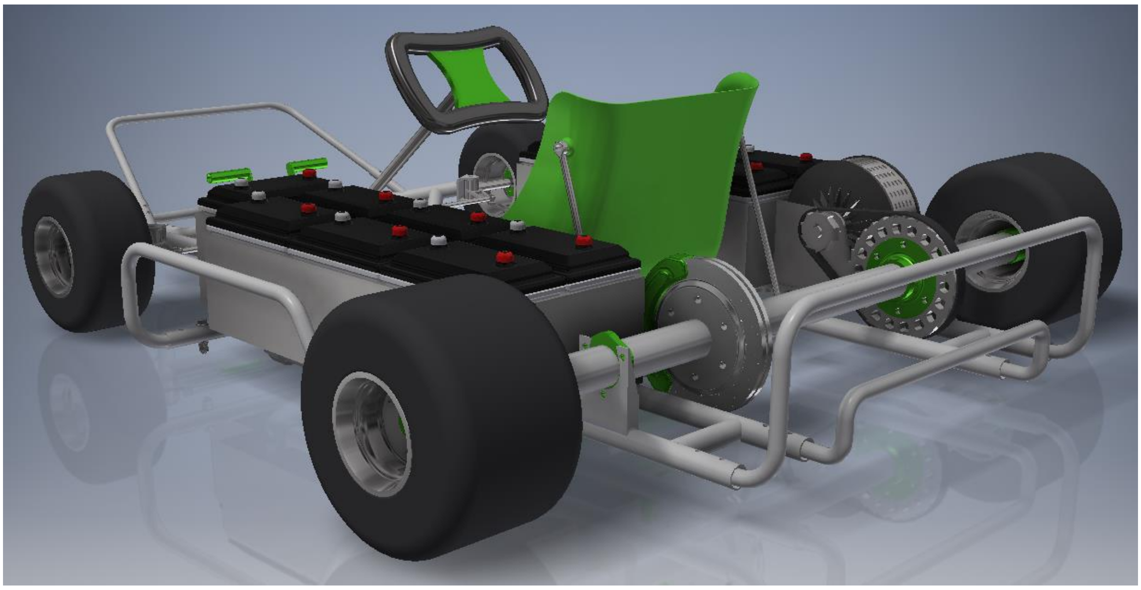


Figure 5: Kart without bodywork

Figure 6: 3D Pilot model

To dimension the pilot are been used the typical dimensions used in the design of chairs.

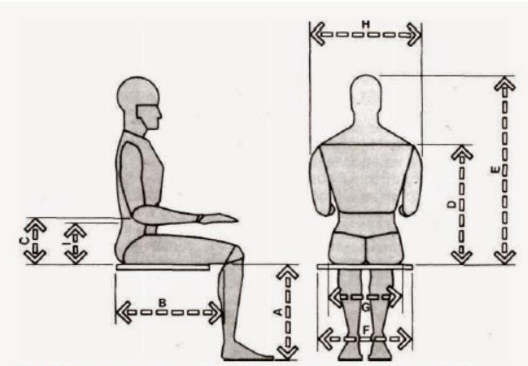


Fig. 4-4. Dimensiones antropométricas fundamentales que se necesitan para el diseño de sillas.

	HOMBRES				MUJERES			
	Percentil				Percentil			
	5		95		5		95	
MEDIDA	pulg.	cm	pulg.	cm	pulg.	cm	pulg.	cm
A Altura poplitea	15.5	39.4	19.3	49.0	14.0	35.6	17.5	44.5
B Largura nalga-popliteo	17.3	43.9	21.6	54.9	17.0	43.2	21.0	53.3
C Altura codo reposo	7.4	18.8	11.6	29.5	7.1	18.0	11.0	27.9
D Altura hombro	21.0	53.3	25.0	63.5	18.0	45.7	25.0	63.5
E Altura sentado, normal	31.6	80.3	36.6	93.0	29.6	75.2	34.7	88.1
F Anchura codo-codo	13.7	34.8	19.9	50.5	12.3	31.2	19.3	49.0
G Anchura caderas	12.2	31.0	15.9	40.4	12.3	31.2	17.1	43.4
H Anchura hombros	17.0	43.2	19.0	48.3	13.0	33.0	19.0	48.3
I Altura lumbar	Véase nota							

Figure 7: Human dimensions

The result obtained of this model is:

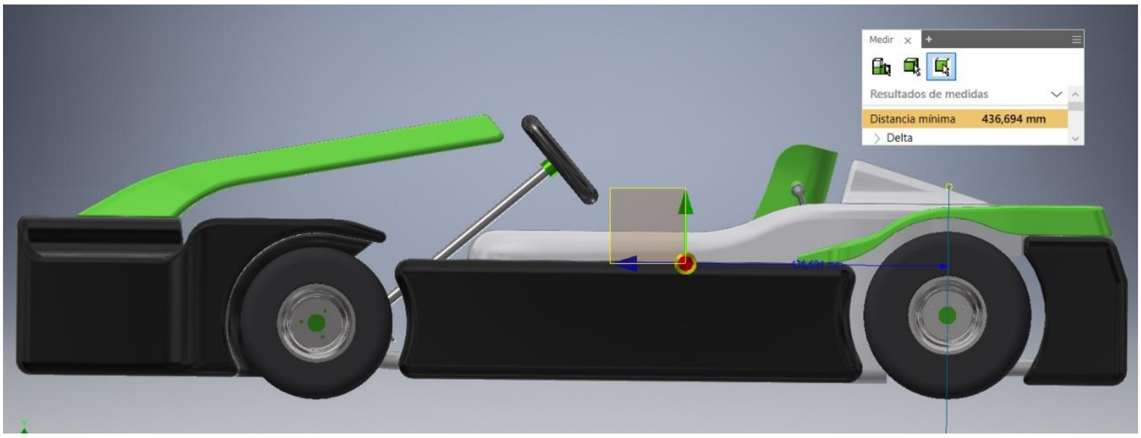


Figure 8: b dimension of the kart

In this image the pilot had to be invisible in order to be able to select the centre of gravity; but its mass is considered to calculate the position of the centre.

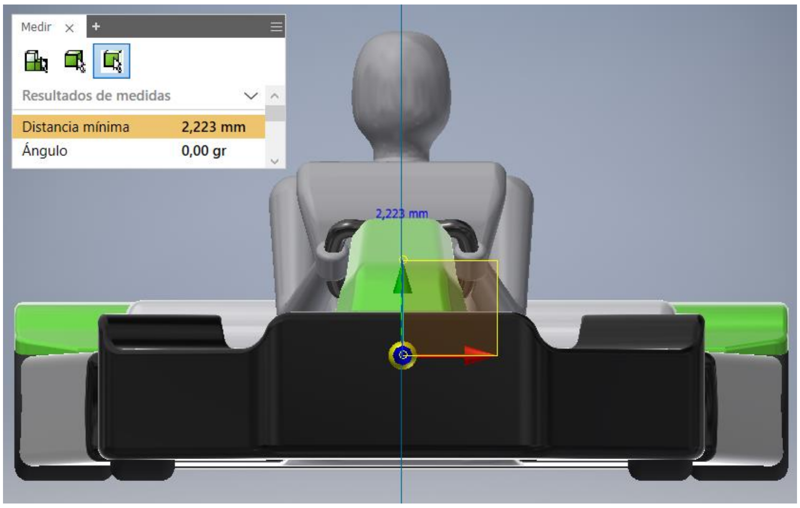


Figure 10: Y CdG Kart

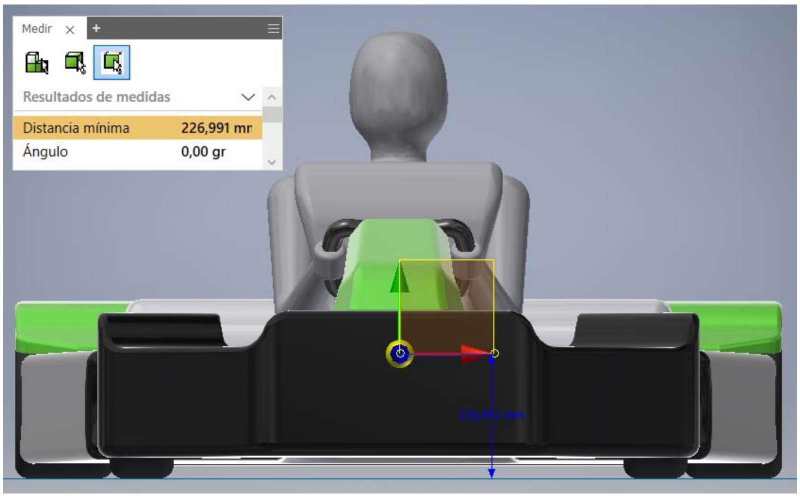


Figure 9: Z CdG Kart

As this model is only a possible configuration in order to obtain realistic values; the values used to do the calculus are:

P (distance between shafts) **[mm]**	1050
a (distance between the frontal shaft and the centre of gravity) **[mm]**	615
b (distance between the motive shaft and the centre of gravity) **[mm]**	435
Y (horizontal position with respect to the middle of the kart) **[mm]**	0
Z (vertical position with respect to the floor) **[mm]**	225

Table 16: Kart dimensions

1.8.5. Maximum accelerations

Once that the dimensions and masses are determinate; the accelerations that is going to suffer the chassis can be calculated.

A) Maximum acceleration in straight:

To calculate the maximum acceleration this equation can be used:

$$a_{max} = \frac{\frac{T_m \cdot \eta_m}{R_w \cdot \tau_g} - (R_a + R_i + R_f)}{m + \frac{4 \cdot I_w}{{R_w}^2} + \frac{I_m \cdot \eta_m}{{R_w}^2 \cdot {\tau_g}^2}} \quad \mathbf{(8.5.1)}$$

$T_m = \text{Motor torque} = 50\ \text{Nm}$

$\eta_m = \text{Engine efficiency} = 95\ \%$

$R_w = \text{ Wheel radius} = 15\ \text{cm}$ (it is the most used but up to 17,5 it is accepted)

$\tau_g = \text{ Transmission ratio} = \frac{1}{3}$ (it is the most common value in Karts)

$R_a = R_i = R_f = 0\ \text{N}$ (because $v = 0\ m/s$; to obtain the maximum acceleration)

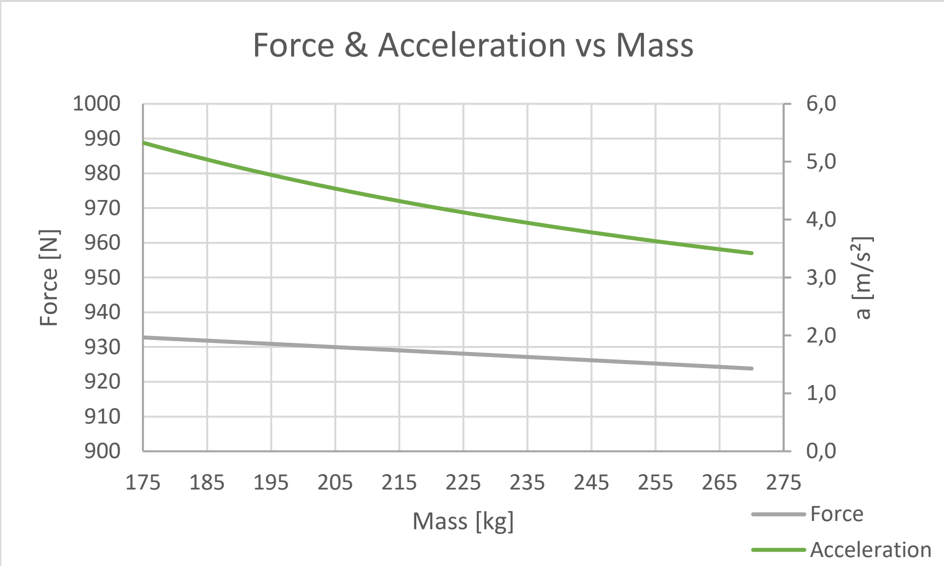
$m = \text{Mass of the kart}$

$\frac{4 \cdot I_w}{{R_w}^2} = \text{Resistence produced by the wheels}$

(It is depreciable respect to the Mass of the kart)

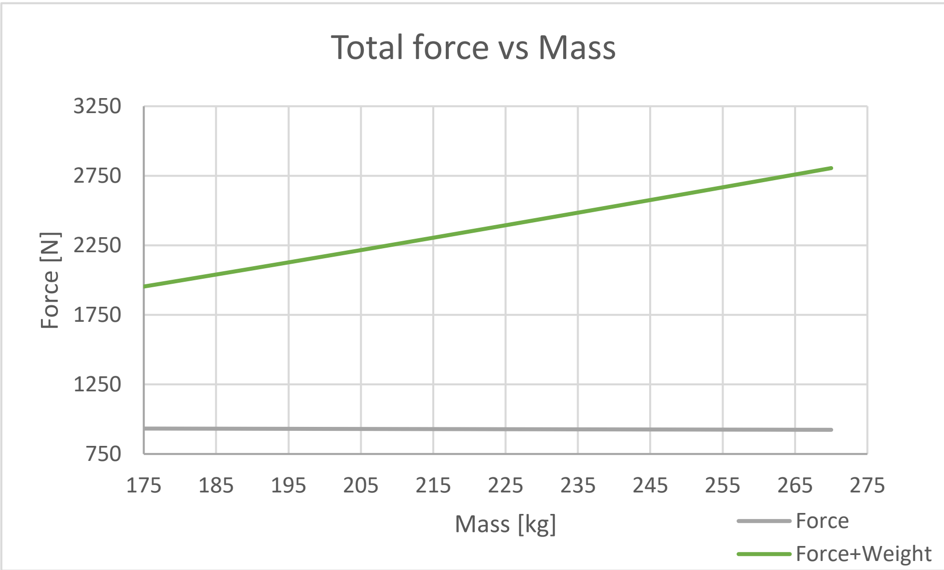
$I_m = \text{Engine inertia} = 45\ \frac{kg}{cm^2}$

In order to know the effect of the mass in the force and in the acceleration, it is done this graphic; that compare these magnitudes with the mass:



Graphic 3: Force and acceleration vs mass

As it can be seen in the graphic, the effect of the mass in the force is small; but if the combination of this force with the mass is analysed:



Graphic 4: Acceleration + Weight vs mass

In that case it is obvious that the worst situation is with a mass of 270 kg; so, the chassis is going to be designed to this mass.

The las test that has to be done into this force is to check if the wheels can transmit this force to the asphalt. To know this the reaction in the motive shaft is needed; and it can be calculated like:

$$N_b = \frac{a \cdot W + z \cdot F}{P} \quad \mathbf{(8.5.3)}$$

N_b = Reaction in the motive shaft

W = Weight = 2.648,7 N

F = Inertia force = 923,8 N

$$N_b = \frac{0{,}615 \cdot 2.648{,}7 + 0{,}225 \cdot 923{,}8}{1{,}05} = 1.749{,}3 \text{ N}$$

A normal value for the coefficient of friction between the wheel of a kart and the asphalt is between 1,2 and 1,3; in that case $\mu_w = 1{,}3$; so, the maximum force that the wheels of the motive shaft could transmit is:

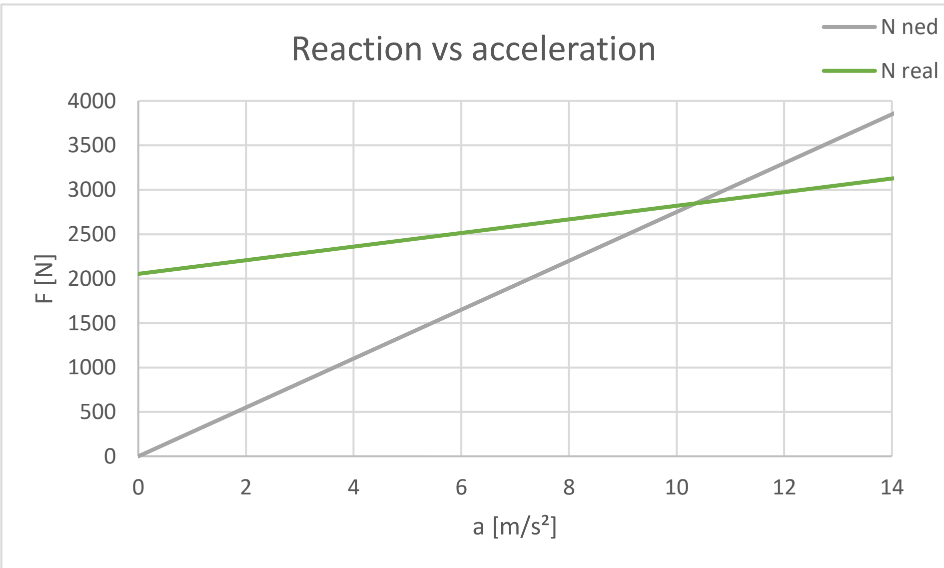
$$F_{max} = N_b \cdot \mu_w = 1.749{,}3 \cdot 1{,}3 = 2274{,}1 \text{ N} \quad (\mathbf{8.5.4})$$

So, this force could be transmitted because is smaller than the maximum force.

For a kart of 270 kg the maximum acceleration that it can be expected is of 3,42 m/s²; with an engine of 50 Nm and a gear with a $\tau = 1/3$

But these are the common values; it is interesting to know which is the maximum acceleration that it could be expected in a kart with these dimensions and with only one motive shaft; the back one. This combination it is the most common. Combining the equations 8.5.3 and 8.5.4 it can be obtained when the force required (N ned) is going to be higher than the force that could provide the motive wheels ($\text{N real} = F_{max}$).

$$\text{N ned} = m \cdot a \quad (\mathbf{8.5.5})$$



Graphic 5: Maximum acceleration

In this hypothesis the maximum acceleration is of 10,35 m/s². Whit the equation 8.5.1 it could be obtained the relation of the relation between the gears; them have to be a $\tau = 0{,}112$ to obtain this acceleration with an engine of 50 Nm. This new gear could reduce the maximum speed; but like I am looking which are the critical forces for the chassis; this calculus is out of the analysis.

So, two acceleration analysis are going to be done, one with 3,5 m/s², due to this could be a real maximum acceleration; and other one with 10,5 m/s²; in order to know if the acceleration have to be imitated in the technical conditions of use (Point 3).

B) Maximum deceleration in straight:

Like it is said in the article 2.11 the brakes have to work in the four wheels, and they must have an independent front and rear operating system. In order to guarantee braking in almost two wheels. So, in order to know which is going to be the maximum deceleration in any of the three possible cases; braking in the four wheels; in the frontal wheels; and in the back wheels.

Wheel in the first case; braking in the four wheels; the kart could be considered like an dimensionless point; in which the maximum deceleration could be calculated using the equation 8.5.4.

$$\mathrm{m} \cdot \delta = \mathrm{m} \cdot \mathrm{g} \cdot \mu_{\mathrm{w}} \Rightarrow \delta = 9{,}81 \cdot 1{,}3 = 12{,}75 \ \mathrm{m/s^2}$$

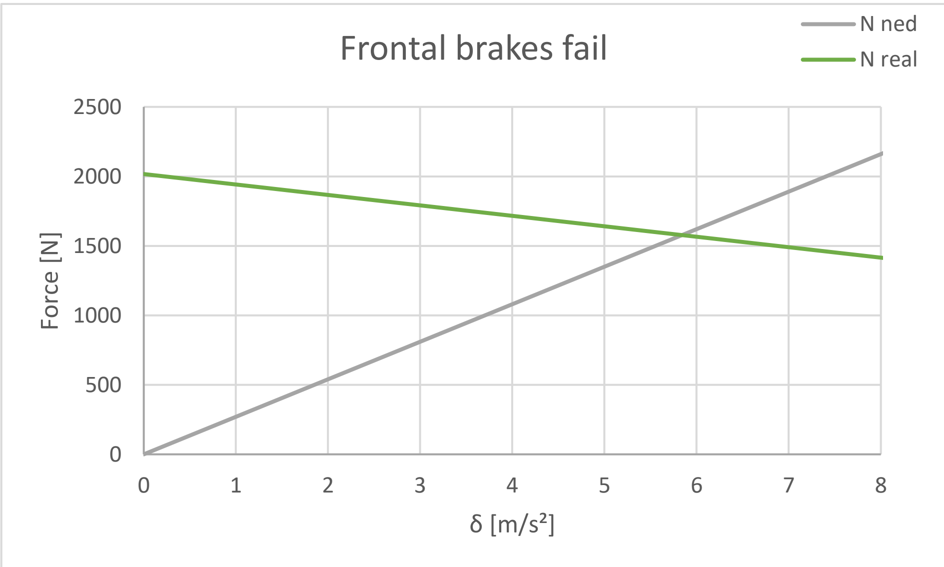
$\delta = \text{deceleration}$

In the other cases it is necessary to know when the required force (N ned) is going to be higher than the force that the wheels in any shaft could provide (N real).

So; if the frontal brakes fail, the maximum deceleration is going to be:

$$\mathrm{N\ ned} = \mathrm{m} \cdot \delta \quad \mathbf{(8.5.6)}$$

$$\mathrm{N\ real} = \frac{\mathrm{a} \cdot \mathrm{m} \cdot \mathrm{g} - \mathrm{z} \cdot \mathrm{m} \cdot \delta}{\mathrm{P}} \cdot \mu_{\mathrm{w}} \quad \mathbf{(8.5.7)}$$



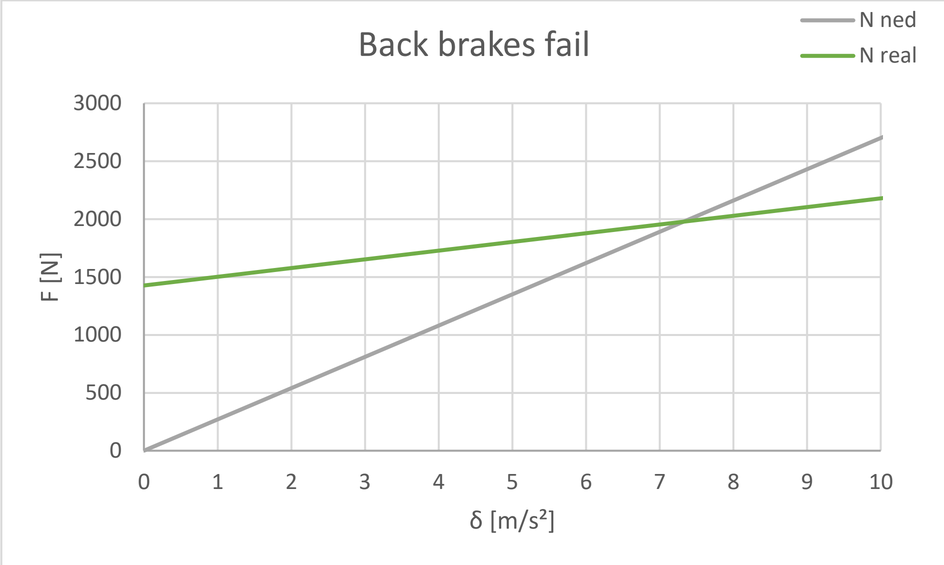
Graphic 6: Maximum deceleration if the frontal brakes fail

Combining the equations 8.5.6 and 8.5.7 the intersection between the two lines:

$$\frac{\delta}{1{,}3} = \frac{(0{,}615 \cdot 9{,}81) - (0{,}225 \cdot \delta)}{1{,}05} \Rightarrow \delta = 5{,}84 \ \mathrm{m/s^2} \quad \mathbf{(8.5.8)}$$

If the back brakes fail the maximum deceleration is going to be:

$$\mathrm{N\ real} = \frac{\mathrm{b} \cdot \mathrm{m} \cdot \mathrm{g} + \mathrm{z} \cdot \mathrm{m} \cdot \delta}{\mathrm{P}} \cdot \mu_{\mathrm{w}} \quad \mathbf{(8.5.9)}$$



Graphic 7: Maximum deceleration if the back brakes fail

Combining the equations 8.5.6 and 8.5.9 the intersection between the two lines:

$$\frac{\delta}{1{,}3} = \frac{(0{,}435 \cdot 9{,}81) + (0{,}225 \cdot \delta)}{1{,}05} \Rightarrow \delta = 7{,}32\ \mathrm{m/s^2} \quad \mathbf{(8.5.10)}$$

In that case 3 analysis are going to be done one with 13 m/s²; other with 6 m/s² and the last one with 7,5 m/s². In any of this analysis the restrictions are going to be changed; but this is going to be explained in the point 1.8.7.

C) Maximum lateral acceleration at constant speed:

If the vehicle is considered like a dimensionless point the maximum lateral speed could be considered like:

$$a_c = \mu_w \cdot g = 1{,}3 \cdot 9{,}81 = 12{,}75\ \mathrm{m/s^2}$$

A consideration that has to be done at this point is if with this acceleration the reaction in the internal wheels is going to be lower than 0. In order to know this, it is going to be calculated the lateral acceleration that reduce the reaction in the internal wheels to 0.

Frontal shaft:

$$0 = \frac{m \cdot g \cdot b}{2 \cdot P} - \frac{m \cdot \frac{b}{P} \cdot a_c \cdot z}{C_f} = \frac{g}{2} - \frac{a_c \cdot z}{C_f} \Rightarrow a_c = \frac{g \cdot C_f}{2 \cdot z} \quad \mathbf{(8.5.11)}$$

C_f = Distance between frontal wheels = 1,055 m

Back shaft:

$$0 = \frac{m \cdot g \cdot a}{2 \cdot P} - \frac{m \cdot \frac{a}{P} \cdot a_c \cdot z}{C_b} = \frac{g}{2} - \frac{a_c \cdot z}{C_b} \Rightarrow a_c = \frac{g \cdot C_b}{2 \cdot z} \quad \mathbf{(8.5.12)}$$

C_b = Distance between back wheels = 1,4 m

The solution to these expressions is 23 and 30,5 m/s² respectively. So, the maximum lateral acceleration is 12,75 m/s²

Other consideration that is going to be done in that case is the oversteer and the understeer; to know which kind of vehicle that is, the following equation can be used:

$$K = \frac{m}{P^2} \cdot \left(\frac{b}{C_{\alpha_f}} - \frac{a}{C_{\alpha_b}}\right) \quad \mathbf{(8.5.13)}$$

$K =$ Coefficient of understeer

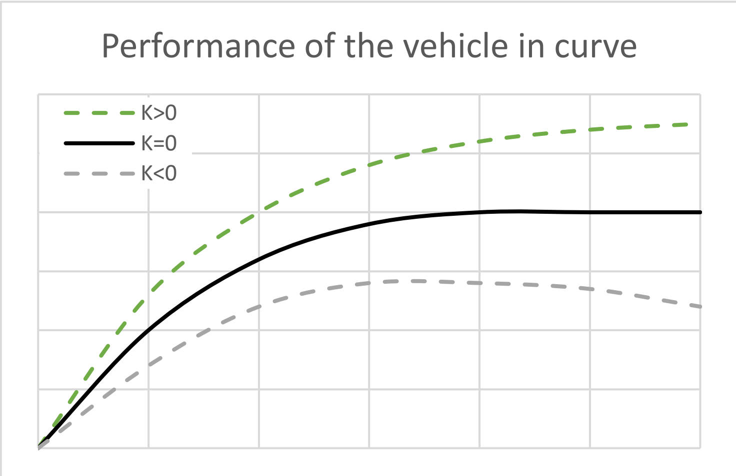
$C_{\alpha_f} \simeq C_{\alpha_b} =$ It is going to be supposed that this coefficient is going to be the same in the front and back wheels

If $C_{\alpha_f} \simeq C_{\alpha_b}$ and $b < a \Rightarrow K < 0 \Rightarrow$ That means that the Kart could suffer oversteer.

If we consider the basic equation of the centripetal acceleration; if the radium decreases the acceleration increases:

$$a_c = \frac{v^2}{R} \quad \mathbf{(8.5.14)}$$

As it can be seen in this graphic a vehicle with a K<0 reduce the radius:



Graphic 8: Kart trajectory according to K

So, in order to consider this effect, the a_c used is going to be 15 m/s^2.

At this point it has to be said that like the kart does not have a differential in the motive shaft; a moment that counteract the effect of the oversteer is going to be generated. But like know if this effect counteracts totally the oversteer it is not so easy; the acceleration of 15 m/s² is going to be used in order to work in the safety side.

D) Maximum acceleration in curve:

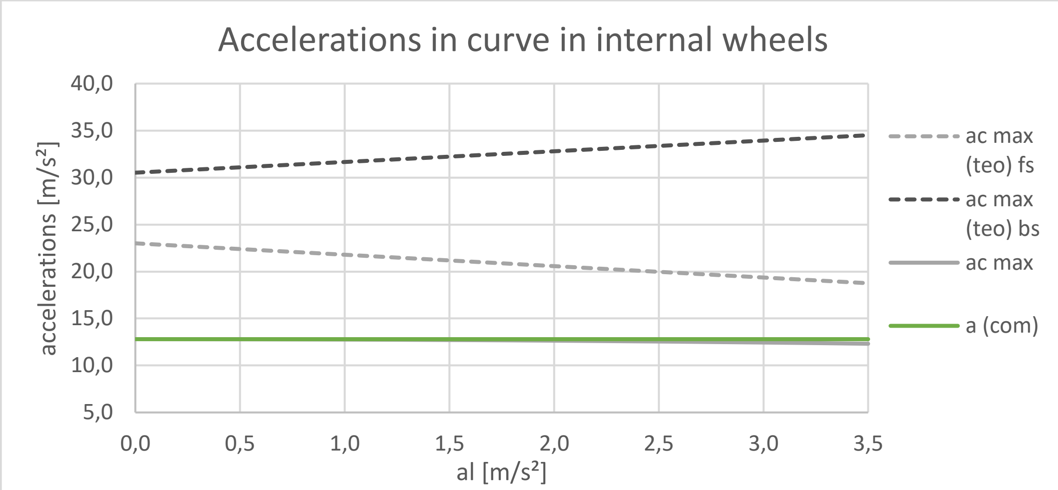
That case is a combination of the A and the C case; in this situation it is necessary to know when the combination of the lateral and longitudinal forces is higher. To know that is going to calculate a graphic in which is going to do a comparison between the lateral and the combination of the two accelerations in front of the longitudinal acceleration. In that case it has to be known which is the maximum combination of accelerations that produce a reaction of 0 Newtons in the frontal internal wheels.

Frontal internal wheel:

$$0 = \frac{m \cdot g \cdot b}{2 \cdot P} - \frac{m \cdot a_l \cdot z}{2 \cdot P} - \frac{\left(\frac{m \cdot b}{P}\right) \cdot z \cdot a_c}{C_f} \Rightarrow a_c = \left(\frac{g \cdot b - a_l \cdot z}{2 \cdot b \cdot z}\right) \cdot C_f \quad \mathbf{(8.5.14)}$$

Back internal wheel:

$$0 = \frac{m \cdot g \cdot a}{2 \cdot P} + \frac{m \cdot a_l \cdot z}{2 \cdot P} - \frac{\left(\frac{m \cdot a}{P}\right) \cdot z \cdot a_c}{C_b} \Rightarrow a_c = \left(\frac{g \cdot a + a_l \cdot z}{2 \cdot a \cdot z}\right) \cdot C_b \quad \mathbf{(8.5.15)}$$



Graphic 9: Acceleration in curve in internal wheels

In this solution some considerations have to be done; the lateral acceleration has to be lower than 13 m/s²; the longitudinal acceleration has to be lower than 3,5 m/s²; and the combination has to be lower than 13 m/s².

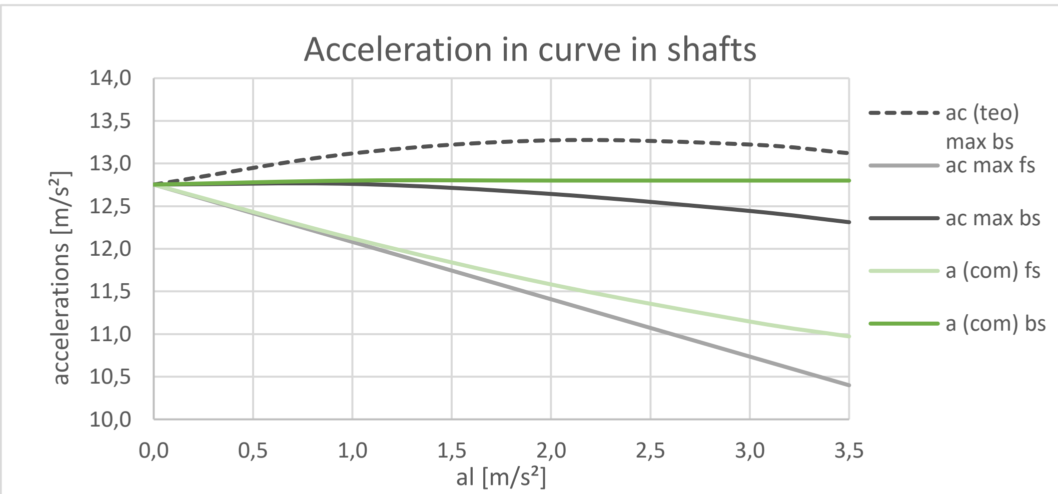
Frontal shaft:
In the frontal shaft the vertical reaction has to be, at less, equal to the lateral force:

$$\left(\frac{m\cdot g\cdot b}{P}-\frac{m\cdot a_l\cdot z}{P}\right)\cdot \mu_w=\frac{m\cdot a_c\cdot b}{P}\Rightarrow a_c=\left(\frac{g\cdot b-a_l\cdot z}{b}\right)\quad (\mathbf{8.5.16})$$

Back shaft:
In the back shaft the vertical reaction kas to be, at less, equal to the combination of the lateral and the longitudinal force:

$$\left(\frac{m\cdot g\cdot a}{P}+\frac{m\cdot a_l\cdot z}{P}\right)\cdot \mu_w=\sqrt{\left(\frac{m\cdot a_c\cdot b}{P}\right)^2+(m\cdot a_l)^2}\Rightarrow$$

$$a_c=\left(\frac{P}{a}\right)\cdot\sqrt{\left[\left(\frac{g\cdot a+a_l\cdot z}{b}\right)\cdot \mu_w\right]^2-a_l{}^2}\quad (\mathbf{8.5.17})$$



Graphic 10: Acceleration in curve in shafts

As it can be seen in the graphic the maximum combination that the two shafts could provide at the same time is when the longitudinal acceleration is 0. That case is the same that the case calculated in the previous point.

E) Maximum deceleration in curve:

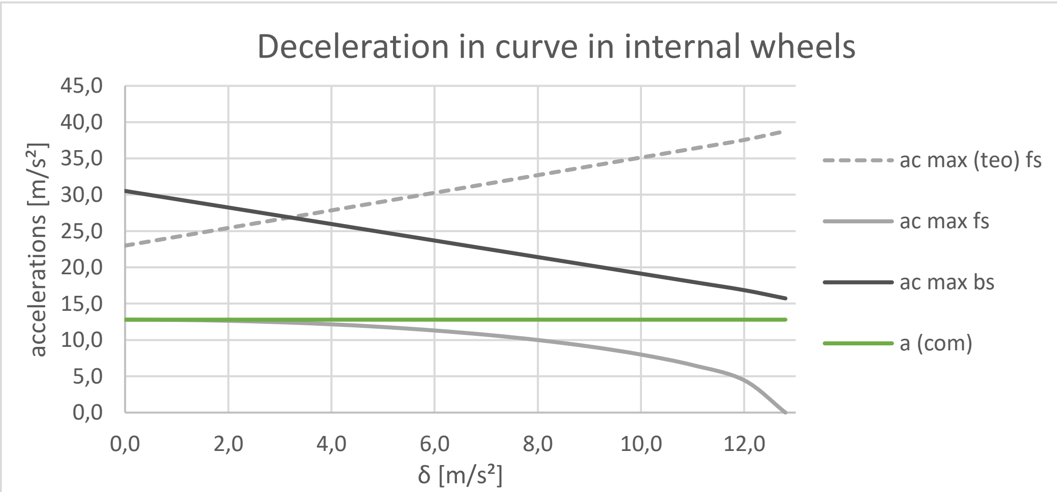
This case is a combination between B and C cases. In this case the same equations that are used in the previous case can be used; but with some modifications:

Frontal internal wheel:

$$0=\frac{m\cdot g\cdot b}{2\cdot P}+\frac{m\cdot \delta\cdot z}{2\cdot P}-\frac{\left(\frac{m\cdot b}{P}\right)\cdot z\cdot a_c}{C_f}\Rightarrow a_c=\left(\frac{g\cdot b+\delta\cdot z}{2\cdot b\cdot z}\right)\cdot C_f\quad \mathbf{(8.5.18)}$$

Back internal wheel:

$$0=\frac{m\cdot g\cdot a}{2\cdot P}-\frac{m\cdot \delta\cdot z}{2\cdot P}-\frac{\left(\frac{m\cdot b}{P}\right)\cdot z\cdot a_c}{C_f}\Rightarrow a_c=\left(\frac{g\cdot a-\delta\cdot z}{2\cdot a\cdot z}\right)\cdot C_b\quad \mathbf{(8.5.19)}$$



Graphic 11: Deceleration in curve in internal wheels

In this solution some considerations have to be done; the lateral acceleration has to be lower than 13 m/s²; the deceleration has to be lower than 13 m/s²; and the combination has to be lower than 13 m/s².

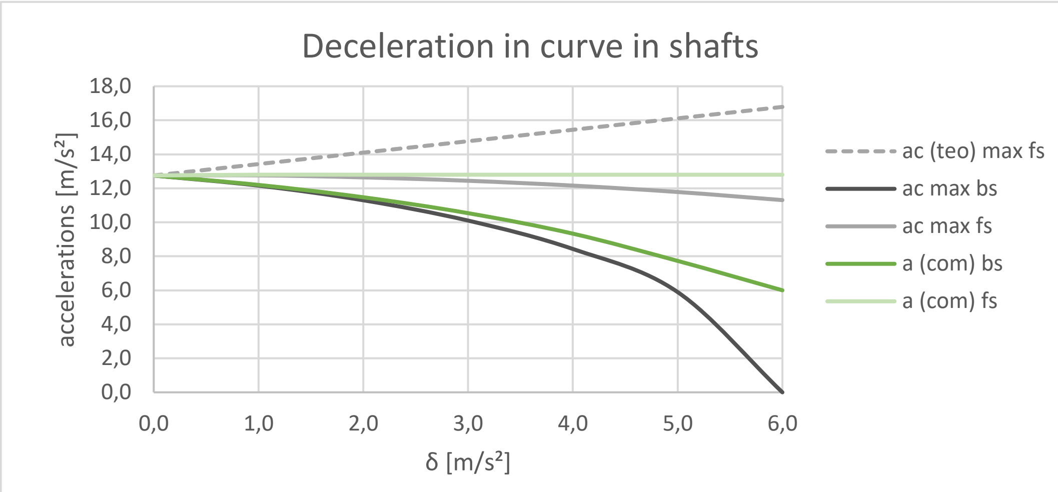
Frontal shaft:
In the frontal shaft the vertical reaction has to be, at less, equal to the lateral force:

$$\left(\frac{m\cdot g\cdot b}{P}+\frac{m\cdot \delta\cdot z}{P}\right)\cdot \mu=\frac{m\cdot a_c\cdot b}{P}\Rightarrow a_c=\left(\frac{g\cdot b+\delta\cdot z}{b}\right)\quad \mathbf{(8.5.20)}$$

Back shaft:
In the back shaft the vertical reaction kas to be, at less, equal to the combination of the lateral and the longitudinal force:

$$\left(\frac{m\cdot g\cdot a}{P}-\frac{m\cdot \delta\cdot z}{P}\right)\cdot \mu=\sqrt{\left(\frac{m\cdot a_c\cdot b}{P}\right)^2+(m\cdot \delta)^2}\Rightarrow$$

$$a_c = \left(\frac{P}{a}\right) \cdot \sqrt{\left[\left(\frac{g \cdot a - \delta \cdot z}{b}\right) \cdot \mu\right]^2 - \delta^2} \quad \mathbf{(8.5.21)}$$



Graphic 12: Deceleration in curve in shafts

In that case happens the same than in the previous case; the critical situation is when the deceleration is 0 m/s². This case is the same that the case C.

F) Final accelerations:

After studying these cases the conclusion is that to know if the chassis resist at these accelerations five situations have to be analysed:

Situation	Longitudinal acceleration [m/s²]	Lateral acceleration [m/s²]
Acceleration	3,5	0,0
	10,5	0,0
Deceleration	-13,0	0,0
	-7,5	0,0
	-6,0	0,0
Turn	0,0	15,0

Table 17: Final accelerations

1.8.6. Forces

To introduce these forces in the software it can be chose between two options; the first one is to introduce only a force in the centre of gravity. The problem of this option is that in some cases the structure is going to be overloaded in some parts, and underloaded in other ones. For this reason these forces are going to be introduced in three different points; the first one in the centre of gravity the pilot; which represents the 37% of the total mass; the second one in the centre of gravity of the batteries that are on the left side; which represents the 19% of the mass; and the last one in the centre of gravity of the batteries that are on the right and the engine; which represents the 19% of the total mass.

These forces are going to be introduced in the model like vertical, horizontal, and axial forces, and bendings moments.

A) Centre of gravity of the components:

A1) Pilot:

The centre of gravity of the pilot respect to the chassis tubes where is connected the seat is:

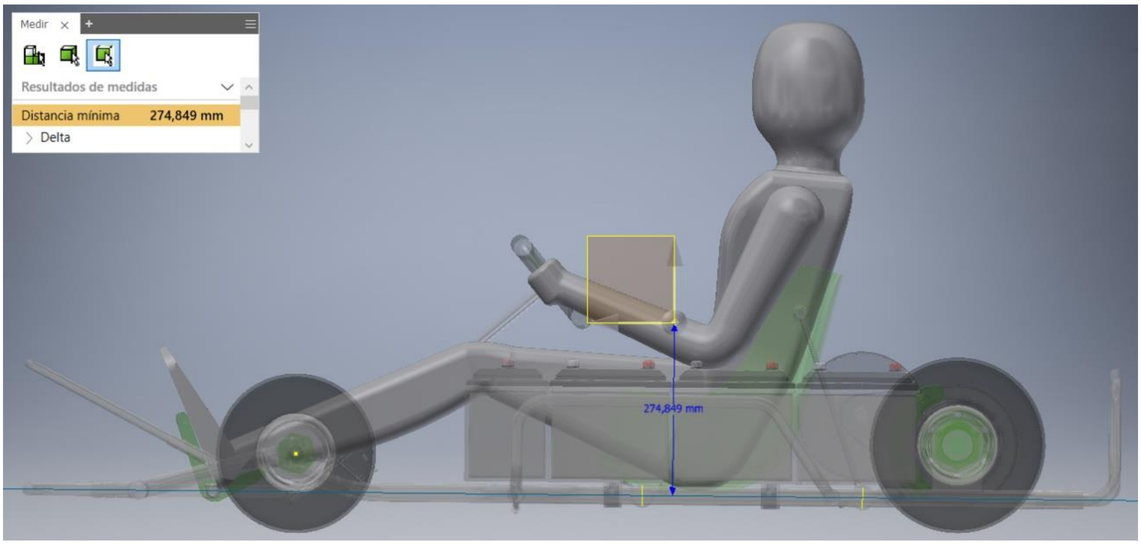


Figure 11: Z CdG Pilot

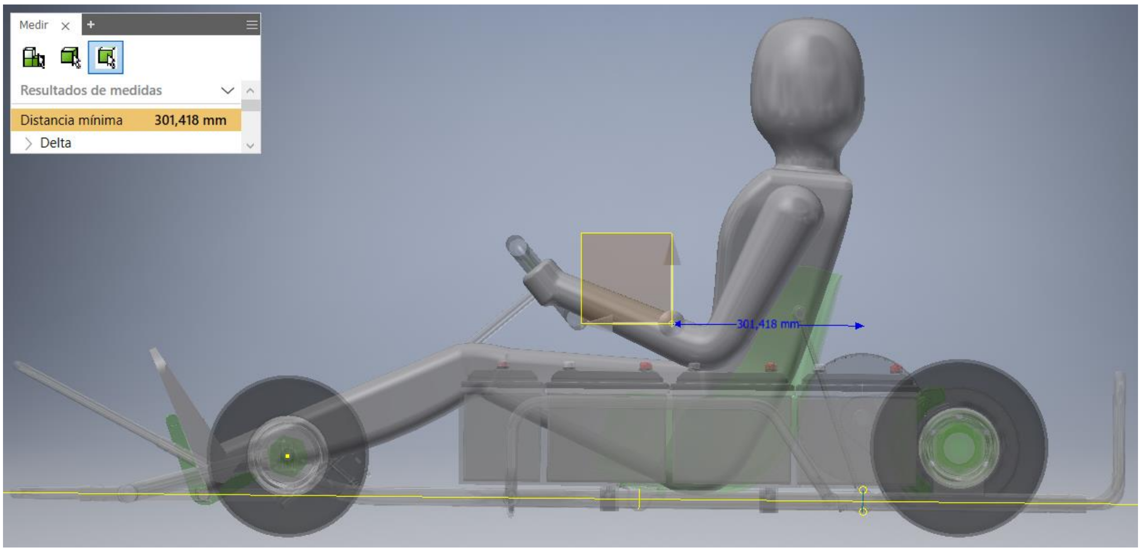


Figure 12: b CdG Pilot

The position of the centre of gravity of the pilot is z = 275 mm; ap = 50 mm; and bp = 300 mm.

A2) Batteries and engine:

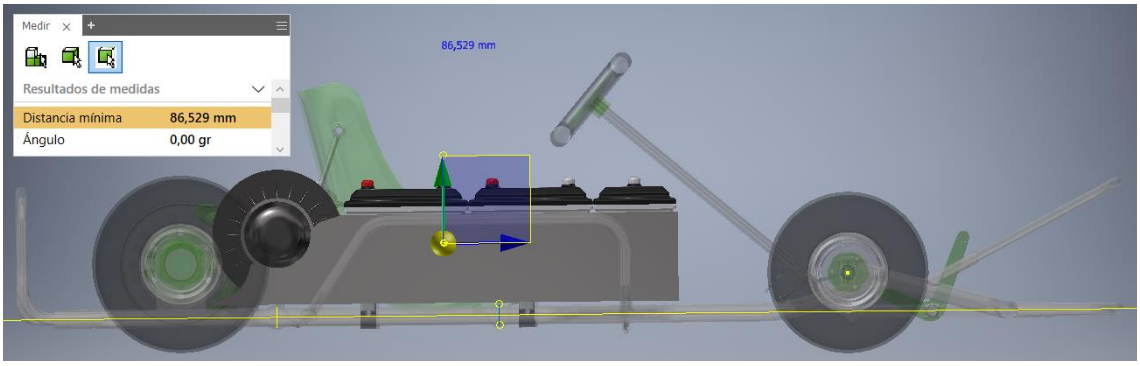


Figure 13: a CdG batteries and engine

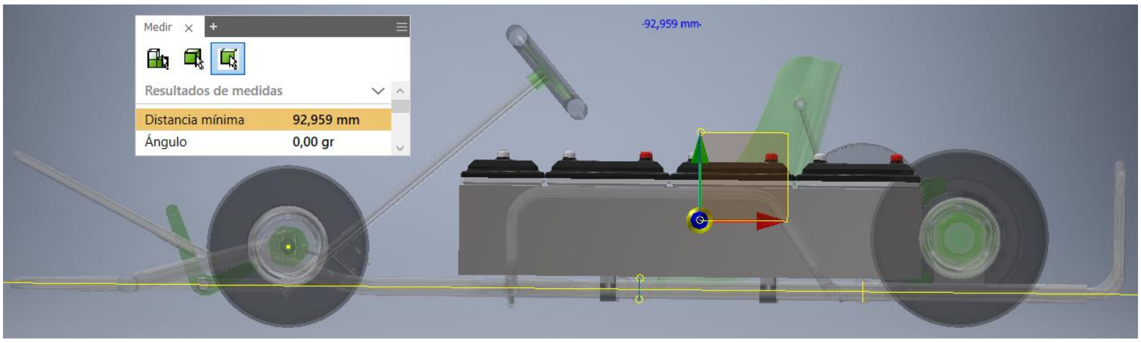


Figure 14: a CdG batteries

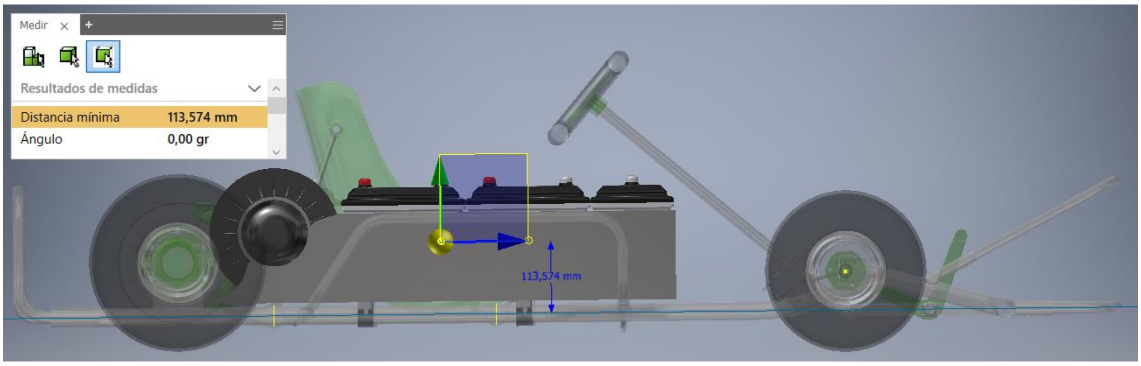


Figure 15: z CdG batteries and engine

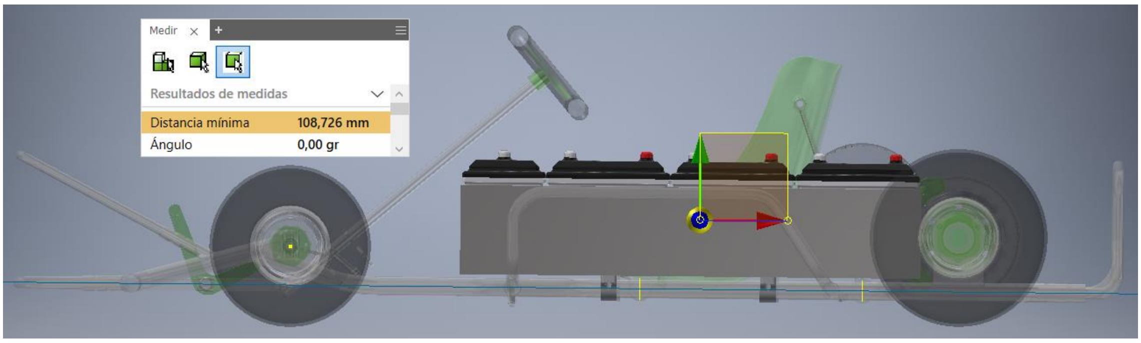


Figure 16: z CdG batteries

To simplify the model, it is going to be used the same distances in any sides; 90 mm for a; and 110 mm for z

B) Points of application of these forces:

To introduce the forces of the batteries them supports are combined in one, this one is in the same position of the gravity centre of the batteries.

The forces of the pilot are going to be applicated in the two central tubes of the structure.

An additional force is going to be introduced to consider the mass of the small components. This mass is going to be considered like 40 kg.

Component	Mass [kg]
Kart (+)	80
Chassis (-)	20
Non-suspended masses (-)	20
Total	40

Table 18: Additional mass

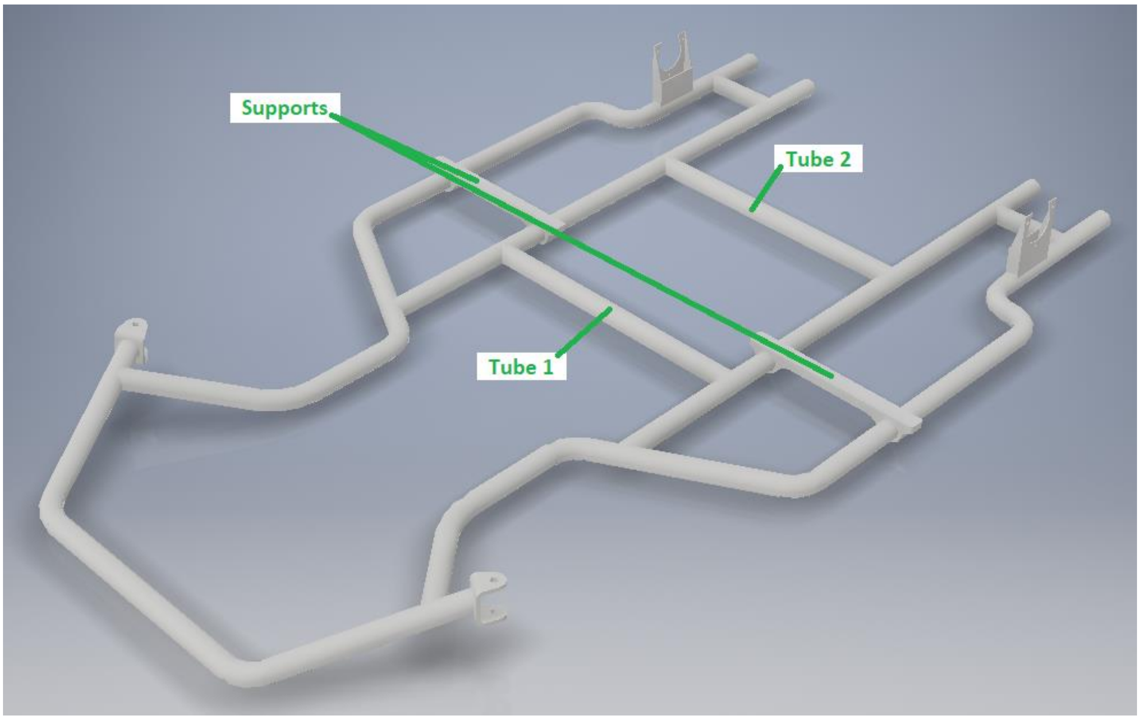


Figure 17: Points of application of the forces

C) Magnitudes of the forces:

Once that the position of the gravity centres and the accelerations are known, the forces that are going to be introduced could be calculated.

Batteries:

Forces due to the weight:

$$W = m_b \cdot g \quad (-z) \quad \mathbf{(8.6.1)}$$

Forces due to the acceleration:

$$A = m_b \cdot a_l \quad (+x) \quad (8.6.2)$$

$$M_A = m_b \cdot a_l \cdot z_b \quad (+y) \quad (8.6.3)$$

Forces due to the deceleration:

$$D = m_b \cdot \delta \quad (-x) \quad (8.6.4)$$

$$M_D = m_b \cdot \delta \cdot z_b \quad (-y) \quad (8.6.5)$$

Forces due to the lateral acceleration:

$$G = m_b \cdot a_c \quad (+y) \quad (8.6.6)$$

$$M_G = m_b \cdot a_c \cdot z_b \quad (-x) \quad (8.6.7)$$

Where z_b is the z coordinate of the gravity centre of the batteries; and m_b the mass of the batteries (50 kg).

Pilot, and additional mass:

Forces due to the weight:

$$W_1 = \left(m_p \cdot \frac{b_p}{P_p} + m_a \cdot \frac{b_a}{P_p}\right) \cdot g \quad (-z) \quad (8.6.8)$$

$$W_2 = \left(m_p \cdot \frac{a_p}{P_p} + m_a \cdot \frac{a_a}{P_p}\right) \cdot g \quad (-z) \quad (8.6.8)$$

Forces due to the acceleration:

$$A_1 = \left(m_p \cdot \frac{b_p}{P_p} + m_a \cdot \frac{b_a}{P_p}\right) \cdot a_l \quad (+x) \quad (8.6.9)$$

$$A_{1v} = -\left(\frac{m_p \cdot z_p + m_a \cdot z}{P_p}\right) \cdot a_l \quad (-z) \quad (8.6.10)$$

$$A_2 = \left(m_p \cdot \frac{a_p}{P_p} + m_a \cdot \frac{a_a}{P_p}\right) \cdot a_l \quad (+x) \quad (8.6.11)$$

$$A_{2v} = \left(\frac{m_p \cdot z_p + m_a \cdot z}{P_p}\right) \cdot a_l \quad (-z) \quad (8.6.12)$$

Forces due to the deceleration:

$$D_1 = \left(m_p \cdot \frac{b_p}{P_p} + m_a \cdot \frac{b_a}{P_p}\right) \cdot \delta \quad (-x) \quad (8.6.13)$$

$$D_{1v} = \left(\frac{m_p \cdot z_p + m_a \cdot z}{P_p}\right) \cdot \delta \quad (-z) \quad (8.6.14)$$

$$D_2 = \left(m_p \cdot \frac{a_p}{P_p} + m_a \cdot \frac{a_a}{P_p}\right) \cdot \delta \quad (-x) \quad (8.6.15)$$

$$D_{2v} = -\left(\frac{m_p \cdot z_p + m_a \cdot z}{P_p}\right) \cdot \delta \quad (-z) \quad (8.6.16)$$

Forces due to the lateral acceleration:

$$G_1 = \left(m_p \cdot \frac{b_p}{P_p} + m_a \cdot \frac{b_a}{P_p}\right) \cdot a_C \quad (+y) \quad \mathbf{(8.6.17)}$$

$$M_{g1} = \left(m_p \cdot \frac{b_p}{P_p} \cdot z_p + m_a \cdot \frac{b_a}{P_p} \cdot z\right) \cdot a_C \quad (-x) \quad \mathbf{(8.6.18)}$$

$$G_2 = \left(m_p \cdot \frac{a_p}{P_p} + m_a \cdot \frac{a_a}{P_p}\right) \cdot a_C \quad (+y) \quad \mathbf{(8.6.19)}$$

$$M_{g2} = \left(m_p \cdot \frac{a_p}{P_p} \cdot z_p + m_a \cdot \frac{a_a}{P_p} \cdot z\right) \cdot a_C \quad (-x) \quad \mathbf{(8.6.20)}$$

$a_p =$ distance between the gravity centre of the pilot and the tube 1

$a_a =$ distance between the gravity centre of the kart and the tube 1

$b_p =$ distance between the gravity centre of the pilot and the tube 2

$b_a =$ distance between the gravity centre of the kart and the tube 2

$P_p =$ distance between tubes

$z_p =$ position z of the centre of gravity of the pilot

$z_b =$ position z of the centre of gravity of the batteries

The direction of these forces is respect the axis system in the figure 3.

Cases:

With these equations and all the parameters calculated before the forces that have to be introduce in the software are:

Case	Point	Force [N]			Moment [Nm]		
		X	Y	Z	X	Y	Z
Acceleration 3,5 m/s²	Batteries support	175	0	-491	0	19	0
	Tube 1	368	0	-712	0	0	0
	Tube 2	122	0	-661	0	0	0
Acceleration 10,5 m/s²	Batteries support	525	0	-491	0	58	0
	Tube 1	1104	0	-74	0	0	0
	Tube 2	366	0	-1299	0	0	0
Turn 15 m/s²	Batteries support	0	750	-491	-83	0	0
	Tube 1	0	1577	-1031	-386	0	0
	Tube 2	0	523	-342	-93	0	0

Table 19: Forces 1 Model 1 support

Case	Point	Force [N]			Moment [Nm]		
		X	Y	Z	X	Y	Z
Deceleration 13 m/s²	Batteries support	-650	0	-491	0	-72	0
	Tube 1	-1367	0	-2216	0	0	0
	Tube 2	-453	0	843	0	0	0
Deceleration 7,5 m/s²	Batteries support	-375	0	-491	0	-41	0
	Tube 1	-789	0	-1715	0	0	0
	Tube 2	-261	0	342	0	0	0
Deceleration 6 m/s²	Batteries support	-300	0	-491	0	-33	0
	Tube 1	-631	0	-1578	0	0	0
	Tube 2	-209	0	205	0	0	0

Table 20: Forces 2 Model 1 support

1.8.7. Tensions and deformations

A) Models:

Once that the forces are calculated they are introduced in the software. To obtain the results the restrictions have to be introduced; in that case a fixed restrictions have been introduces in the unions between the chassis and the motive shaft; and a sliding restrictions (in the z direction) are introduced in the supports of the direction system.

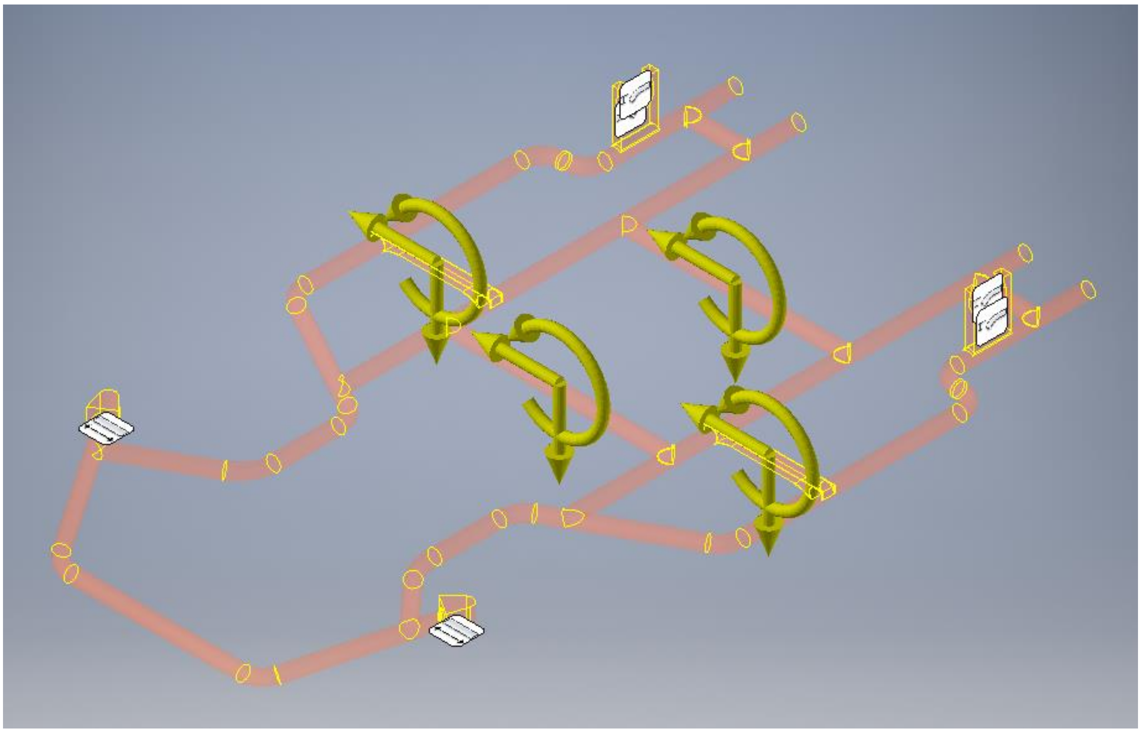


Figure 18: Model 1 support

In this model a mesh of 4841 points and 9505 elements have been used to calculate the tensions for any case.

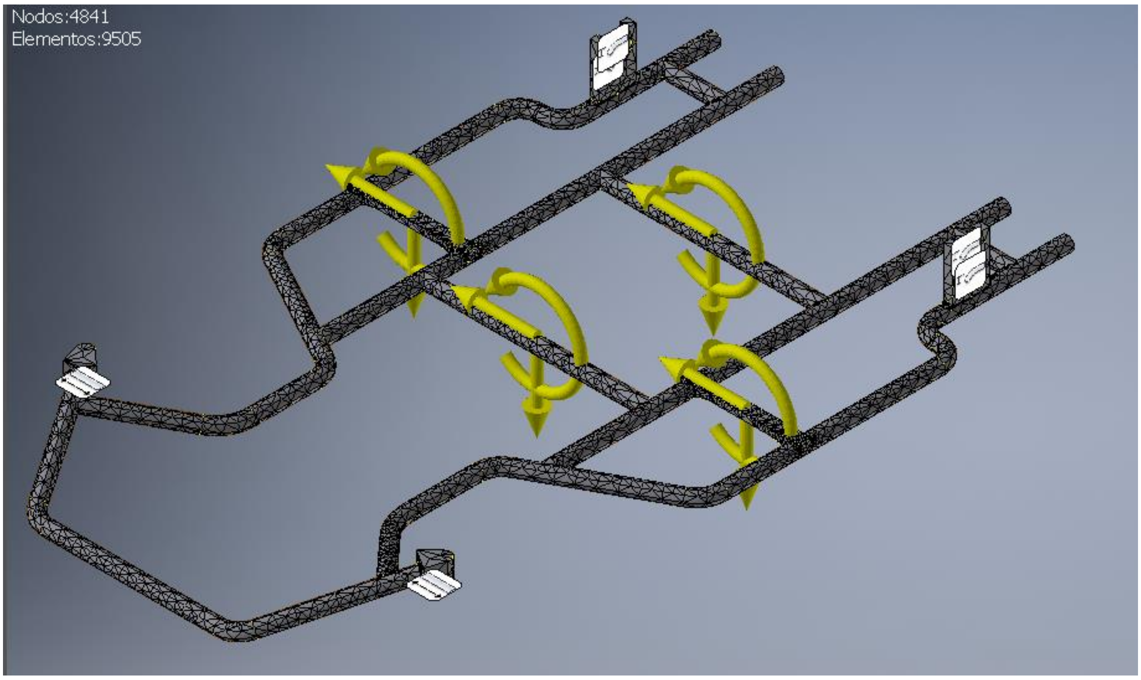


Figure 19: Mesh of the 1 support Model

This model gives a maximum tension in an inexistent point; the union between the batteries support and the chassis. This happens in the turn analysis; giving a maximum tension of 294,3 MPa; near to the 320 MPa that is the fatigue tension of the steel used.

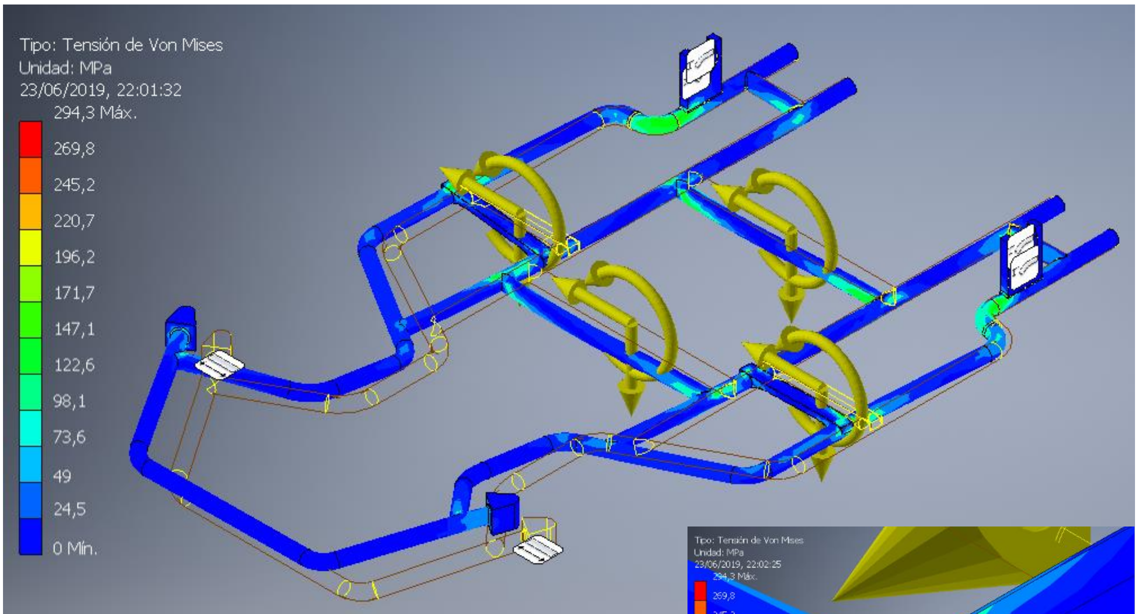


Figure 20: Critical tension in 1 support model

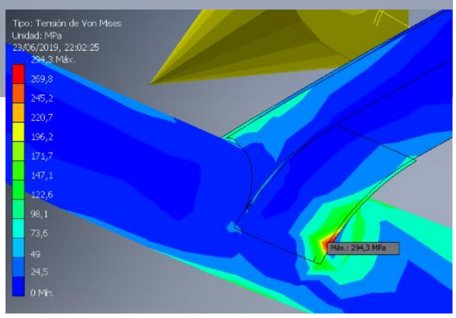


Figure 21: 1 support model, critical point

In order to obtain results closer to the reality a new model with the two supports in any side has done. In that case some modifications to the equations used to the first model have to be done for the batteries supports.

Forces due to the weight:

$$W_{b1} = \left(m_b \cdot \frac{b_b}{P_b}\right) \cdot g \quad (-z) \quad \mathbf{(8.6.21)}$$

$$W_{b2} = \left(m_b \cdot \frac{a_b}{P_b}\right) \cdot g \quad (-z) \quad \mathbf{(8.6.22)}$$

Forces due to the acceleration:

$$A_{b1} = \left(m_b \cdot \frac{b_b}{P_b}\right) \cdot a_l \quad (+x) \quad \mathbf{(8.6.23)}$$

$$A_{b1v} = -\left(\frac{m_b \cdot z_b}{P_b}\right) \cdot a_l \quad (-z) \quad \mathbf{(8.6.24)}$$

$$A_{b2} = \left(m_b \cdot \frac{a_b}{P_b}\right) \cdot a_l \quad (+x) \quad \mathbf{(8.6.25)}$$

$$A_{b2v} = \left(\frac{m_b \cdot z_b}{P_b}\right) \cdot a_l \quad (-z) \quad \mathbf{(8.6.26)}$$

Forces due to the deceleration:

$$D_{b1} = \left(m_b \cdot \frac{b_b}{P_b}\right) \cdot \delta \quad (-x) \quad \mathbf{(8.6.27)}$$

$$D_{b1v} = \left(\frac{m_b \cdot z_b}{P_b}\right) \cdot \delta \quad (-z) \quad \mathbf{(8.6.28)}$$

$$D_{b2} = \left(m_b \cdot \frac{a_b}{P_b}\right) \cdot \delta \quad (-x) \quad \mathbf{(8.6.29)}$$

$$D_{b2v} = -\left(\frac{m_b \cdot z_b}{P_b}\right) \cdot \delta \quad (-z) \quad \mathbf{(8.6.30)}$$

Forces due to the lateral acceleration:

$$G_{b1} = \left(m_b \cdot \frac{b_b}{P_b}\right) \cdot a_C \quad (+y) \quad \mathbf{(8.6.31)}$$

$$M_{bg1} = \left(m_b \cdot \frac{b_b}{P_b} \cdot z_b\right) \cdot a_C \quad (-x) \quad \mathbf{(8.6.32)}$$

$$G_{b2} = \left(m_b \cdot \frac{a_b}{P_b}\right) \cdot a_C \quad (+y) \quad \mathbf{(8.6.33)}$$

$$M_{bg2} = \left(m_b \cdot \frac{a_b}{P_b} \cdot z_b\right) \cdot a_C \quad (-x) \quad \mathbf{(8.6.34)}$$

a_b = distance between the gravity centre of the batteries and the support 1

b_b = distance between the gravity centre of the batteries and the support 2

P_b = distance between supports

So the forces that have to be introduced in the software are the following for any case:

Case	Point	Force [N]			Moment [Nm]		
		X	Y	Z	X	Y	Z
Acceleration 3,5 m/s²	Batt. Supp. 1	88	0	-168	0	0	0
	Batt. Supp. 2	88	0	-322	0	0	0
	Tube 1	368	0	-712	0	0	0
	Tube 2	122	0	-661	0	0	0
Acceleration 10,5 m/s²	Batt. Supp. 1	263	0	-14	0	0	0
	Batt. Supp. 2	263	0	-476	0	0	0
	Tube 1	1104	0	-74	0	0	0
	Tube 2	366	0	-1299	0	0	0
Turn 15 m/s²	Batt. Supp. 1	0	263	-172	-29	0	0
	Batt. Supp. 2	0	263	-172	-29	0	0
	Tube 1	0	1320	-863	-315	0	0
	Tube 2	0	480	-314	-81	0	0
Deceleration 13 m/s²	Batt. Supp. 1	-325	0	-531	0	0	0
	Batt. Supp. 2	-325	0	41	0	0	0
	Tube 1	-1367	0	-2216	0	0	0
	Tube 2	-453	0	843	0	0	0
Deceleration 7,5 m/s²	Batt. Supp. 1	-188	0	-410	0	0	0
	Batt. Supp. 2	-188	0	-80	0	0	0
	Tube 1	-789	0	-1715	0	0	0
	Tube 2	-261	0	342	0	0	0
Deceleration 6 m/s²	Batt. Supp. 1	-150	0	-377	0	0	0
	Batt. Supp. 2	-150	0	-113	0	0	0
	Tube 1	-631	0	-1578	0	0	0
	Tube 2	-209	0	205	0	0	0

Table 21: Forces Model 2 supports

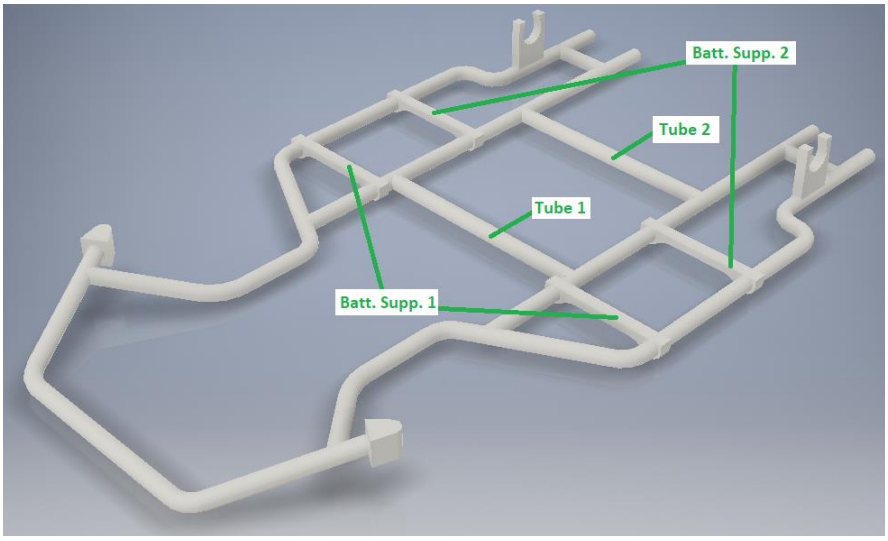


Figure 22: Poins of application of the forces (final model)

In the new model the same restrictions than before are used (in general); and the same application of the software; that simplify the model if its thickness is small.

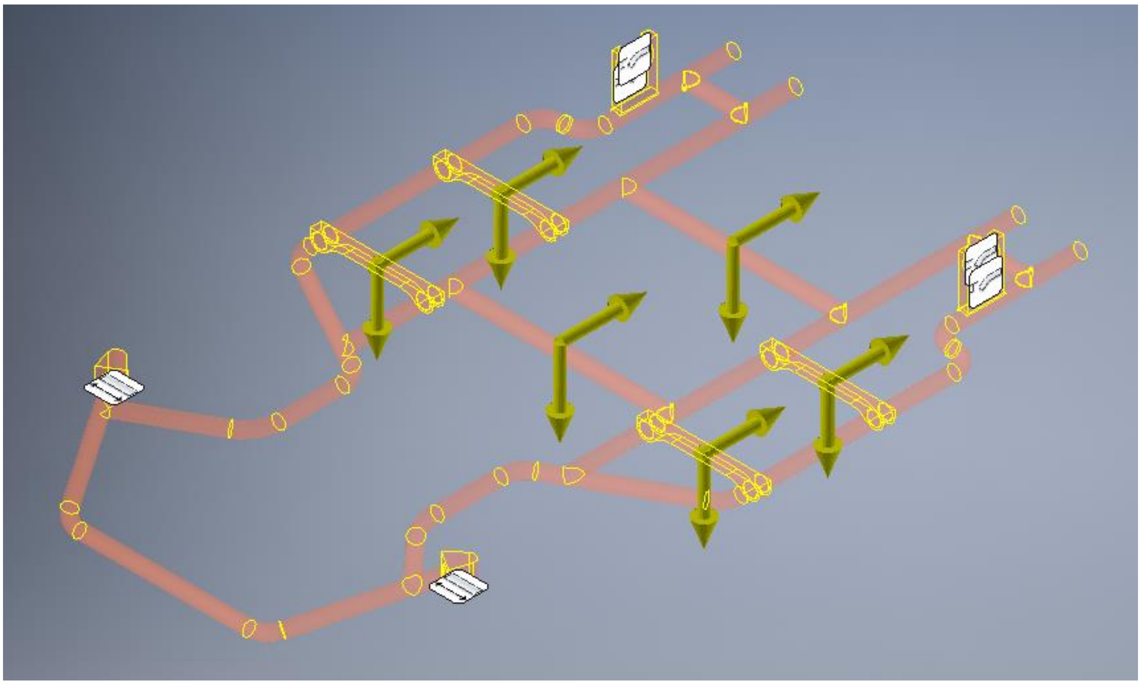


Figure 23: 2 supports model, edited

The new model is used a mesh of 5365 points and 10310 elements. The restrictions used are the same that in the previous model

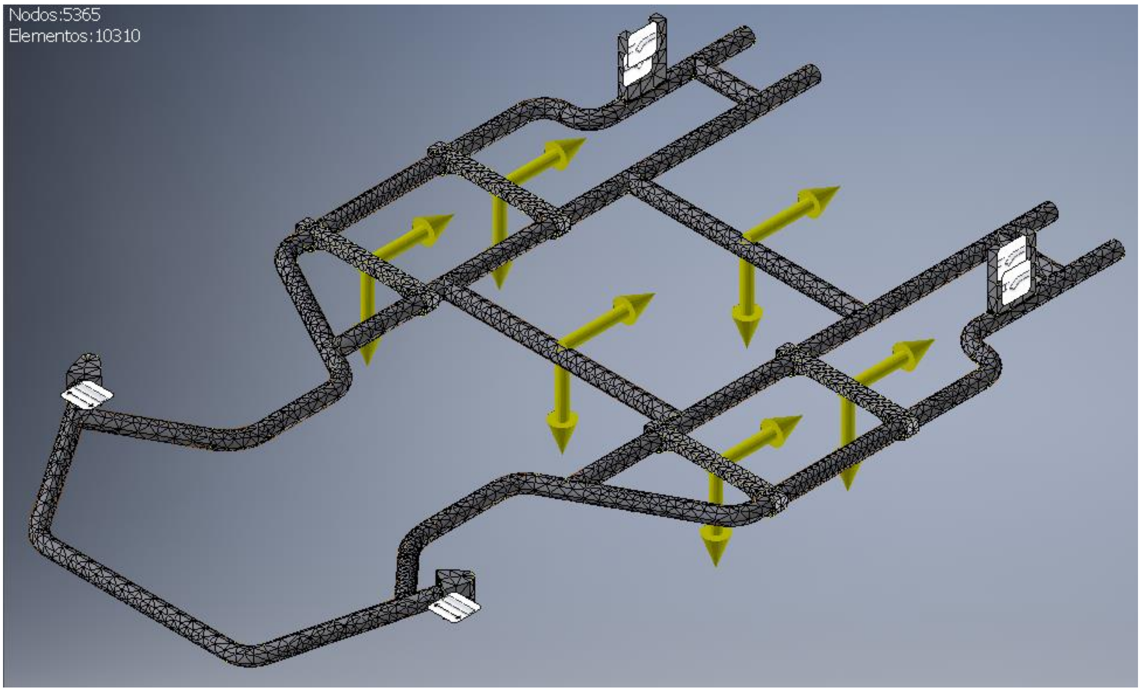


Figure 24: 2 supports model, mesh

B) Results:

Like the three cases are going to be commons in the use of the kart; the maximum tension accepted is going to be 320 MPa; that is the fatigue tension for this steel. The deformation that have to be limited is the vertical one; just a big deformation in tis directions could make the pilot or some components of the kart crash or brush against the asphalt; por this reason this maximum deformation accepted is going to be of 5 mm.

B1) Acceleration 3,5 m/s²

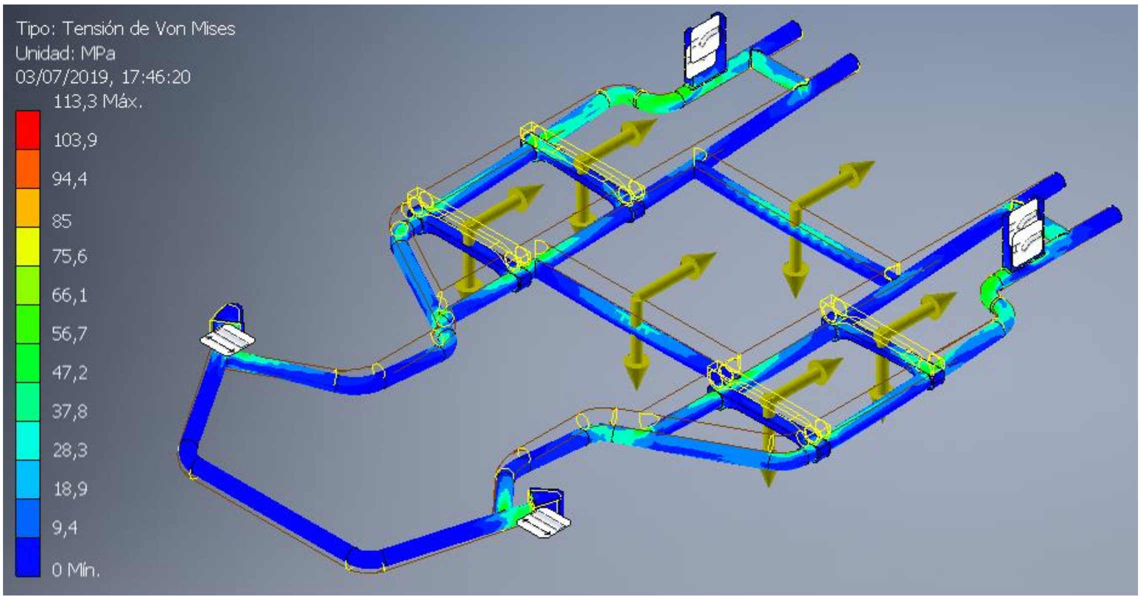


Figure 25: Tension in Acceleration 3,5 m/s²

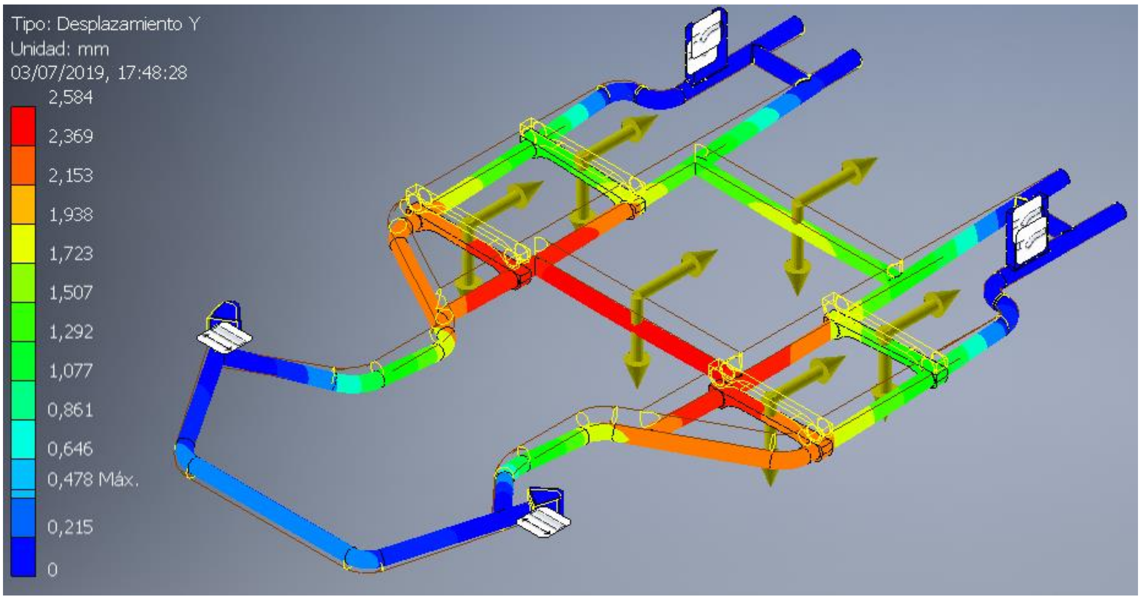


Figure 26: Deformation in Acceleration 3,5 m/s²

As it can be seen in the graphics the maximum tension in this case is of 113,3 MPa; and the deformation in the vertical direction is of 2,58 mm. This tension gives a safety coefficients of:

$$X_f = \frac{\sigma_f}{\sigma_{real}} = \frac{320}{113{,}3} = 2{,}82 \quad (\mathbf{8.7.1})$$

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{113{,}3} = 3{,}88 \quad (\mathbf{8.7.2})$$

B2) Acceleration 10,5 m/s²

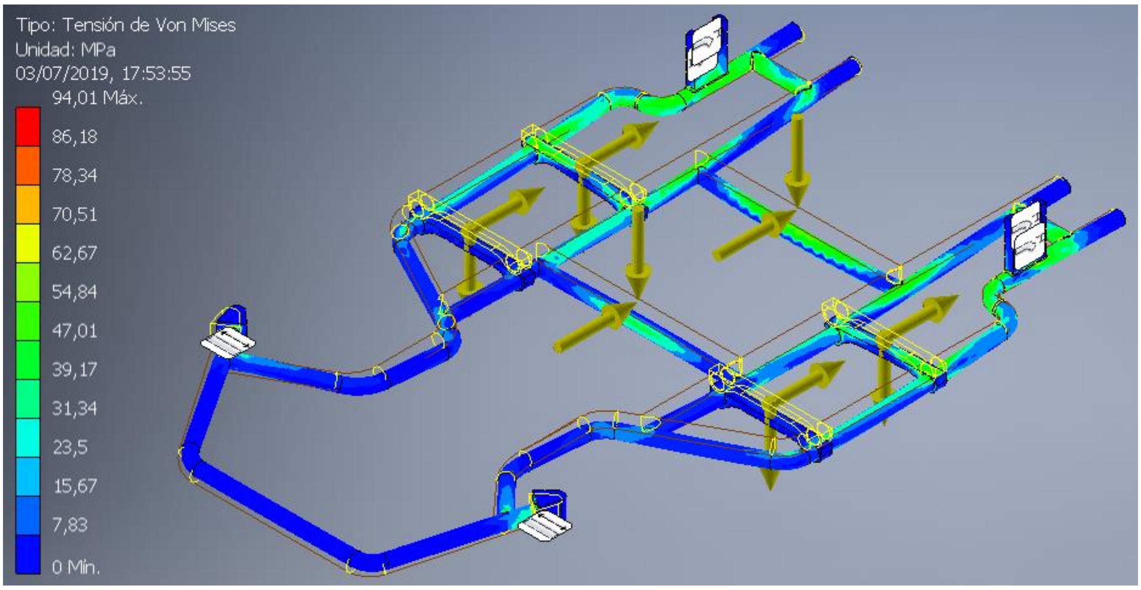


Figure 27: Tension in Acceleration 10,5 m/s²

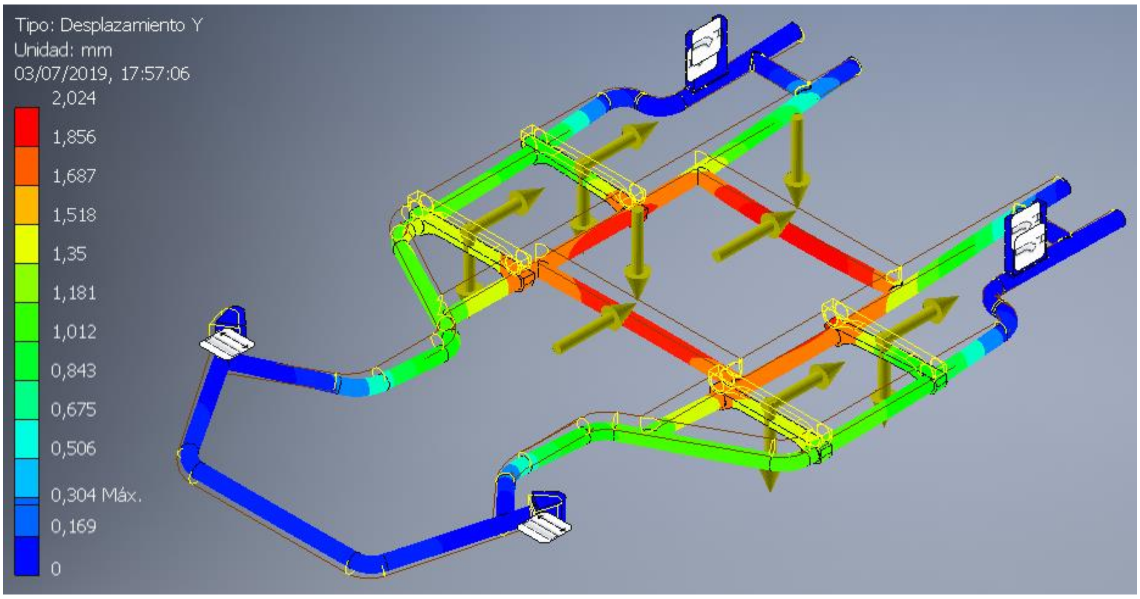


Figure 28: Deformation in Acceleration 10,5 m/s²

As it can be seen in the graphics the maximum tension in this case is of 94 MPa; and the deformation in the vertical direction is of 2,02 mm. This tension gives a safety coefficients of:

$$X_f = \frac{\sigma_f}{\sigma_{real}} = \frac{320}{94} = 3{,}40$$

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{94} = 4{,}68$$

As it can be observed in the figures this situation is less complicated for the structure; this is because due to the dynamics of the acceleration there is a load transfer, reducing the loads in the points where in the previous case there was a higher load.

B3) Turn 15 m/s²

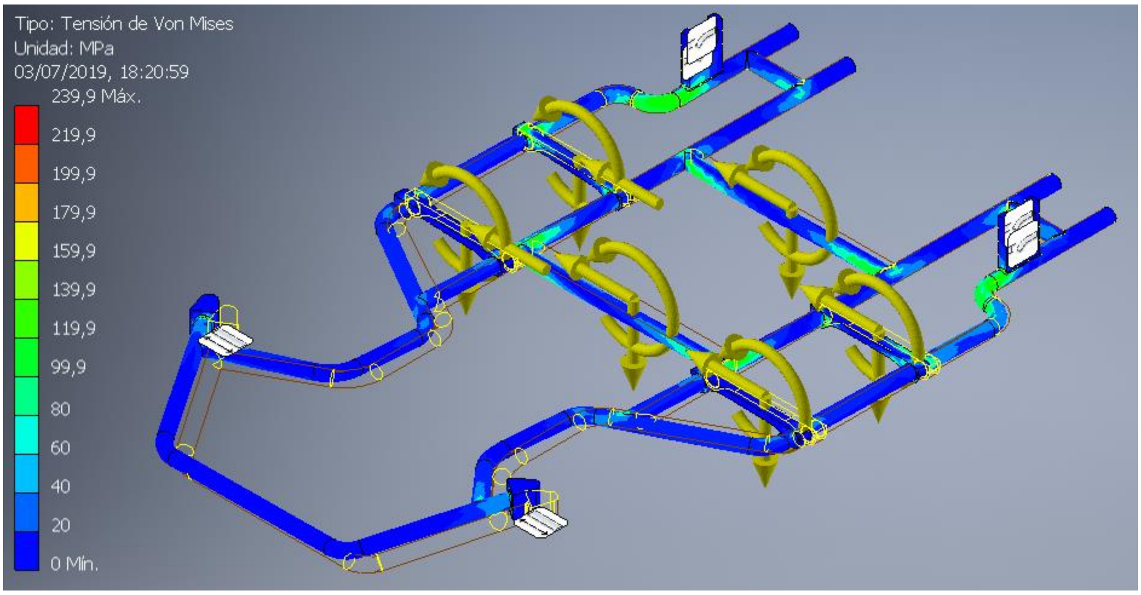


Figure 29: Tension in turn 15 m/s²

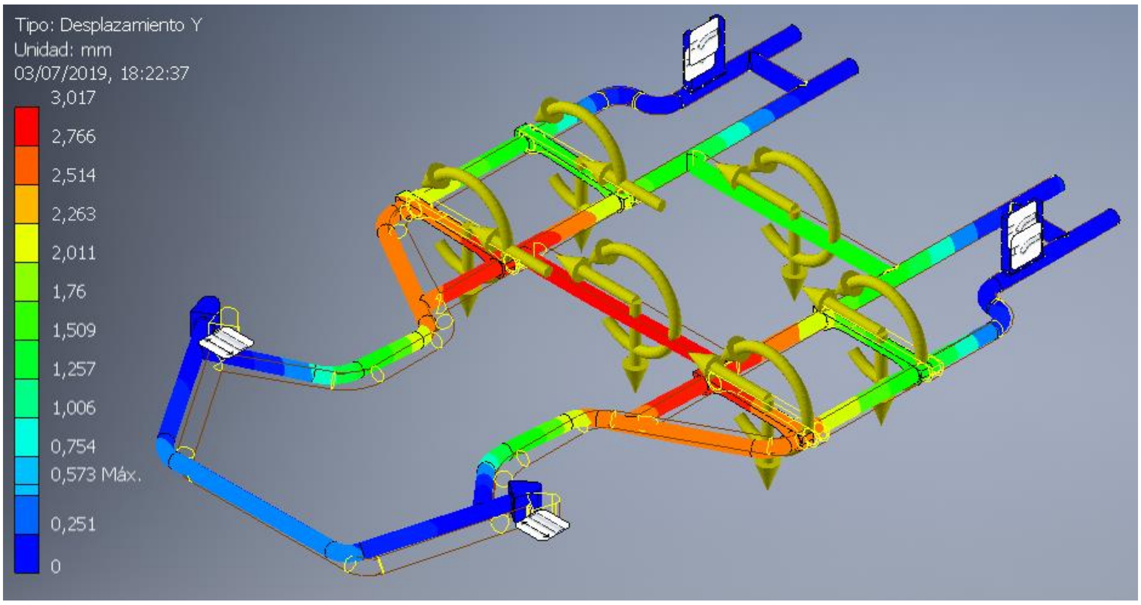


Figure 30: Vertical deformation in turn 15 m/s²

As it can be seen in the graphics the maximum tension in this case is of 239,9 MPa; and the deformation in the vertical direction is of 3,02 mm. This tension gives a safety coefficients of:

$$X_f = \frac{\sigma_f}{\sigma_{real}} = \frac{320}{239{,}9} = 1{,}33$$

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{239{,}9} = 1{,}83$$

In this case is interesting to know the horizontal deformation.

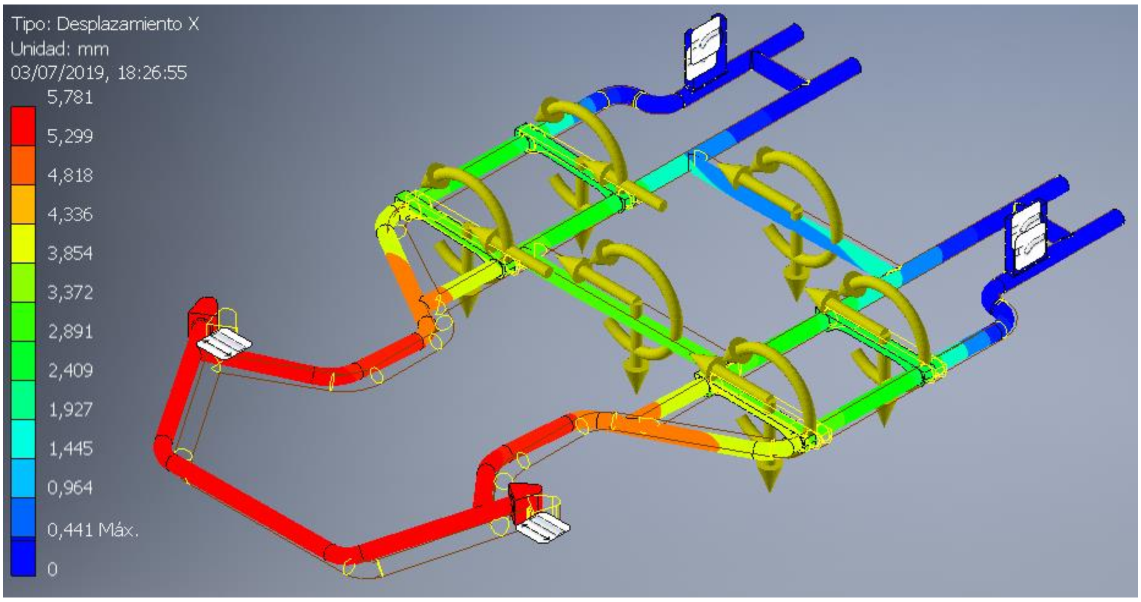


Figure 31: Lateral deformation in turn 15 m/s²

B4) Deceleration 13 m/s²

In that case the restrictions have to be changed; introducing fixed restriction in the four points.

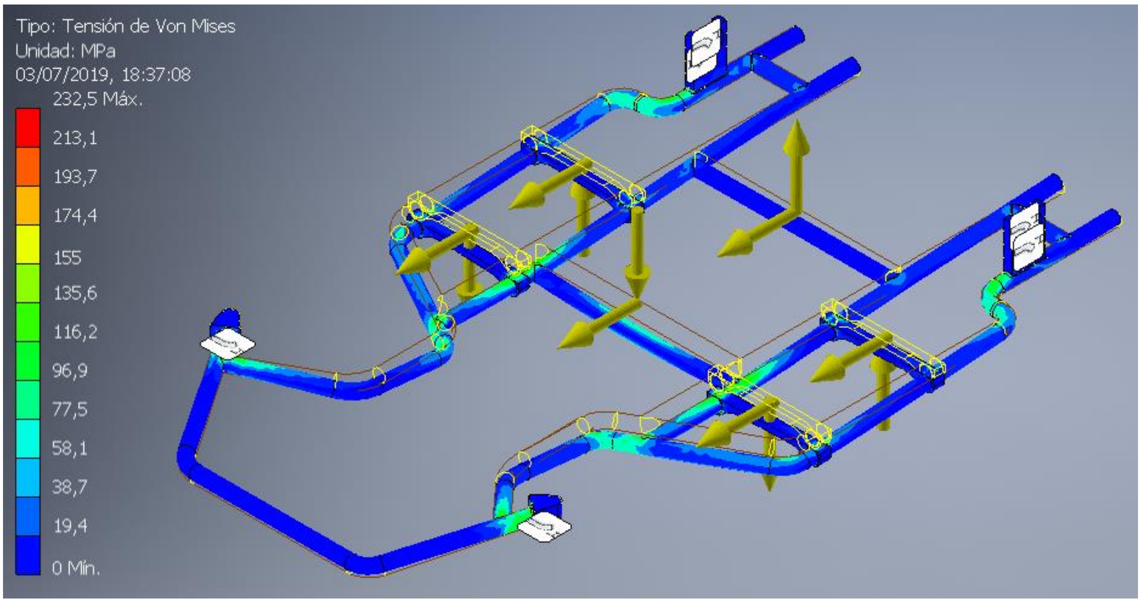


Figure 32: Tension in deceleration 13 m/s²

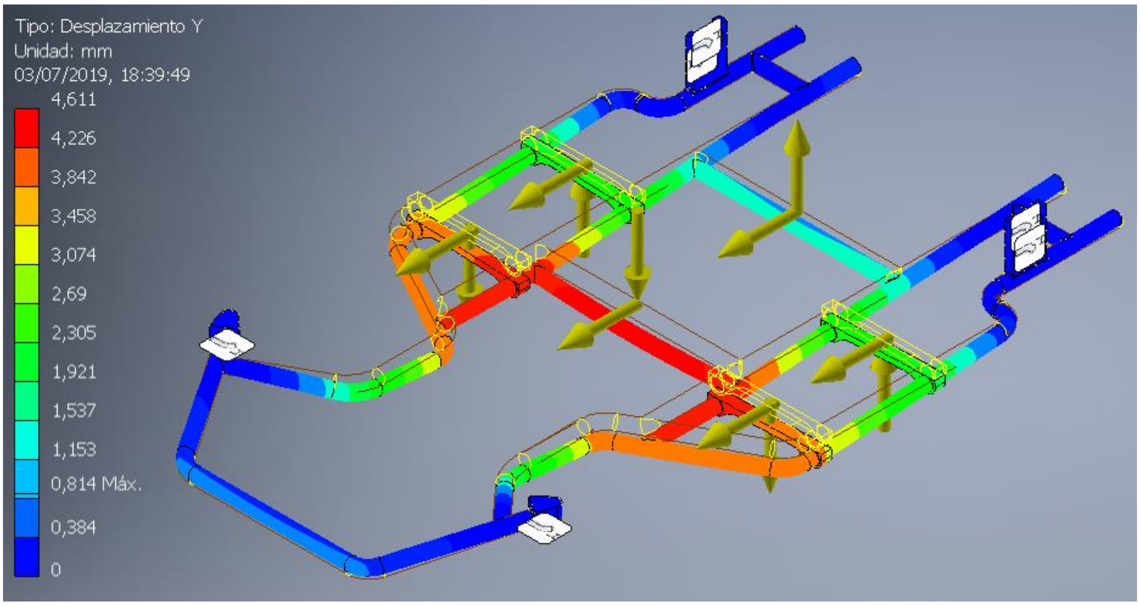


Figure 33: Deformation in deceleration 13 m/s²

As it can be seen in the graphics the maximum tension in this case is of 232,5 MPa; and the deformation in the vertical direction is of 4,61 mm. This tension gives a safety coefficients of:

$$X_f = \frac{\sigma_f}{\sigma_{real}} = \frac{320}{232{,}5} = 1{,}37$$

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{232{,}5} = 1{,}89$$

B5) Deceleration when the back brakes fail, 7,5 m/s²

In that case the restrictions have to be fixed in the front and a restriction that only block de vertical displacement in the vertical axis.

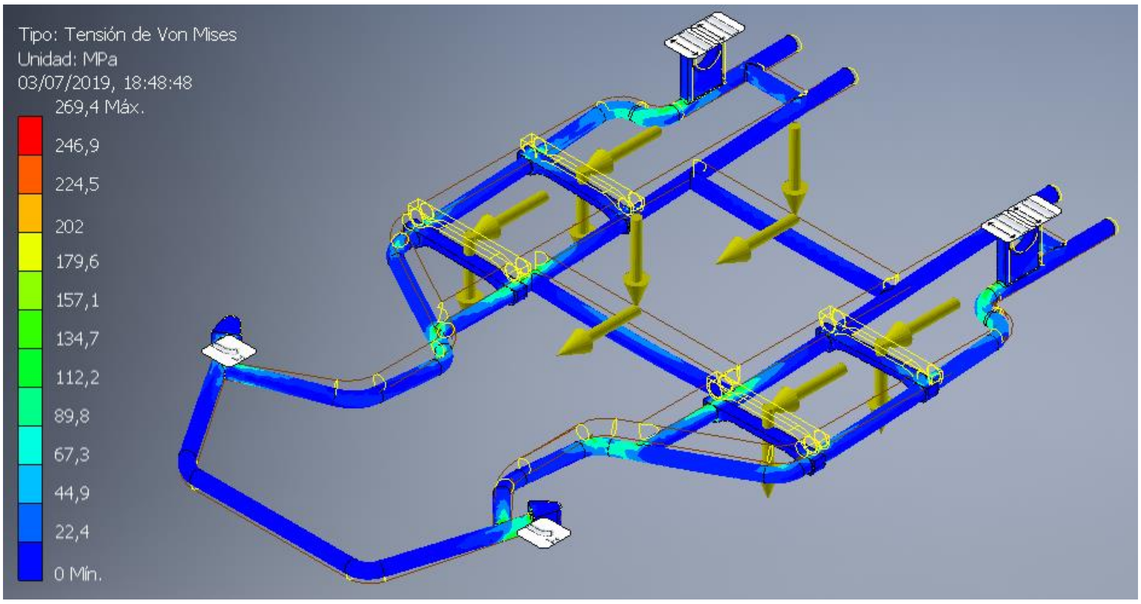


Figure 34: Tension in deceleration 7,5 m/s²

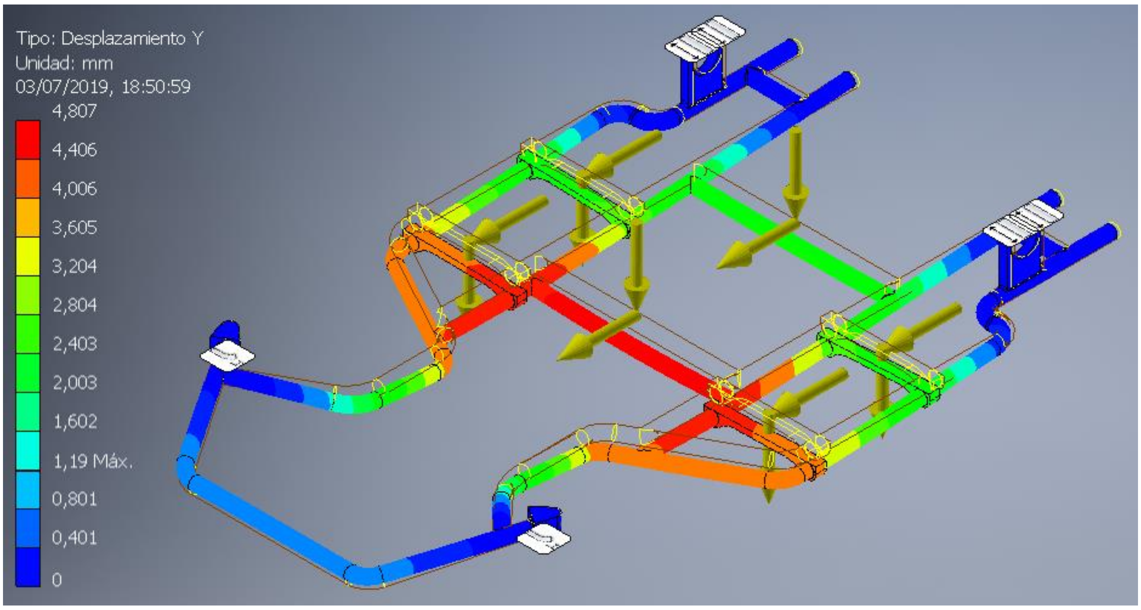


Figure 35: Deformation in deceleration 7,5 m/s²

As it can be seen in the graphics the maximum tension in this case is of 269,4 MPa; and the deformation in the vertical direction is of 4,81 mm. But this should not be a usual situation; due this, the important coefficient in this case is the elastic one.

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{269{,}4} = 1{,}63$$

B6) Deceleration when the frontal brakes fail, 6 m/s²

In that case the restrictions are the used as common.

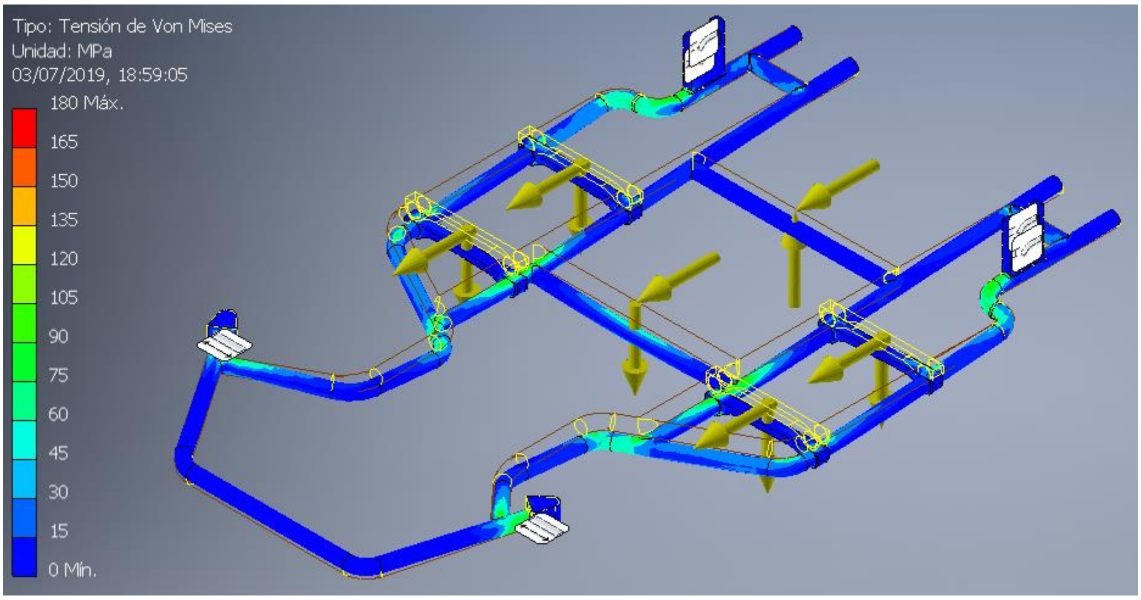


Figure 36: Tension in deceleration 6 m/s²

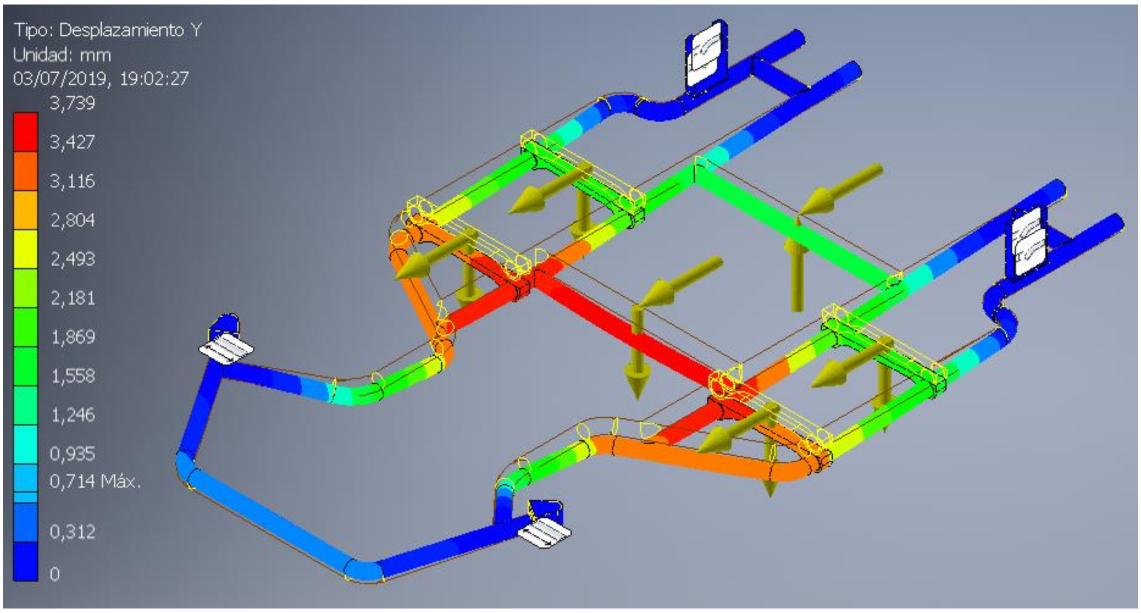


Figure 37: Deformation in deceleration 6 m/s²

As it can be seen in the graphics the maximum tension in this case is of 180 MPa; and the deformation in the vertical direction is of 3,74mm. But, as in the previous case, this should not be a usual situation; due this, the important coefficient in this case is the elastic one.

$$X_y = \frac{\sigma_y}{\sigma_{real}} = \frac{440}{180} = 2,44$$

B6) Results:

Case		Max σ [MPa]	Xf [-]	Xy [-]	Max. deformations [mm]		
					X	Y	Z
Acceleration	3,5 m/s²	113,3	2,82	3,88	0,41	0,15	2,58
	10,5 m/s²	94,0	3,40	4,68	0,54	0,32	2,02
Deceleration	13 m/s²	232,5	1,38	1,89	0,65	0,29	4,61
	7,5 m/s²	269,4	---	1,63	1,08	0,22	4,81
	6 m/s²	180,0	---	2,44	0,32	0,17	3,74
Turn	15 m/s²	239,9	1,33	1,83	0,88	5,78	3,02
Critical		269,4	1,33	1,63	1,08	5,78	4,81

Table 22: Results of the tensional and deformations analysis

1.8.8. Torsional deformation

Like this vehicle cannot have shocks-absorbing the structure have to be deformable in order to increases the comfort; for this reason, is important to know its torsional deformation. In order to calculate this value, the previous model it is used, blocking the unions between the structure and the motive shaft, and introducing two forces (of 100 N) in the lateral unions between the chassis and the direction system.

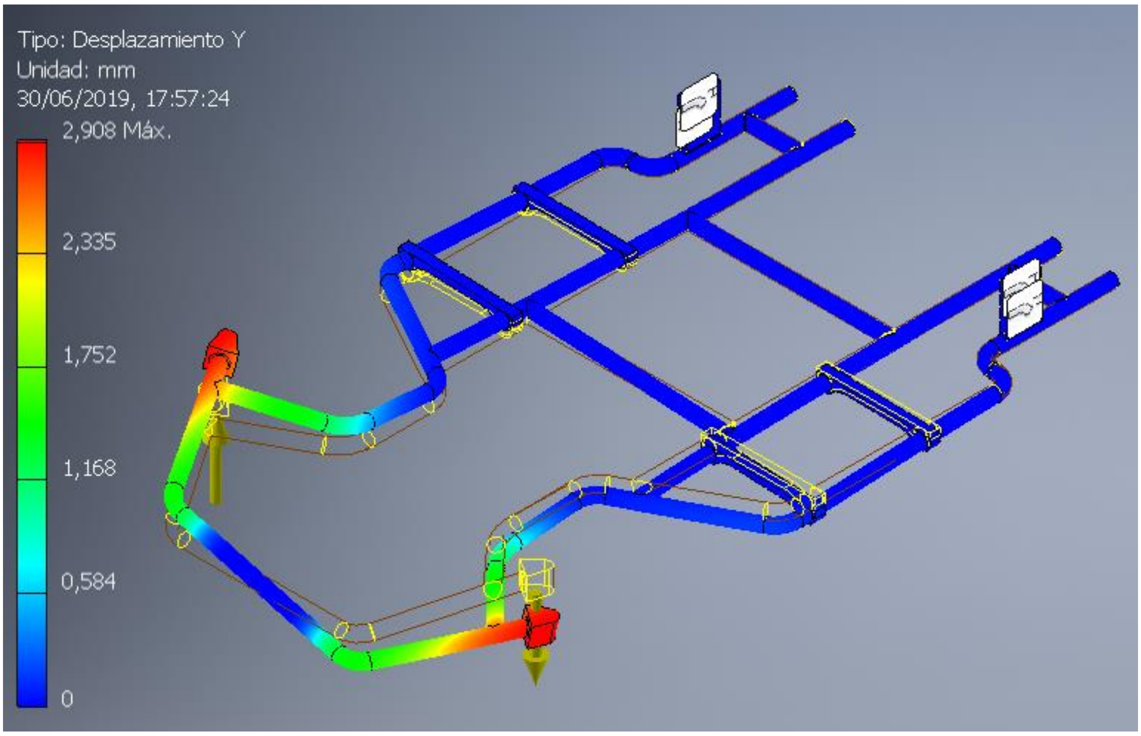


Figure 38: Torsional deformation

Knowing this displacement and the distance between the forces the angle could be calculate:

$$\text{tang}\ (\alpha) = \frac{2 \cdot 2{,}9}{700} \Rightarrow \alpha = 0{,}5^{\text{o}} \quad (\mathbf{8.8.1})$$

So, the deformation respect to de moment is going to be:

$$\frac{0{,}5}{100 \cdot 0{,}7} = 140\ \text{Nm}/^{\text{o}} \quad (\mathbf{8.8.2})$$

This value is a little small; but if the structure is modified to increase it, the tensions and deformations in the turn case are going to increase; in all the cases, but in this case this effect is going to be higher. In the previous point the coefficients of security are calculated; and the fatigue coefficient in the case of the turn was in the limit; so, if the structure is modified, this coefficient is going to be lower. In order to avoid this, the structure is not going to be changed.

1.8.9. Vibrations

The las point of this annex is going to be dedicated to do a little analysis of the vibration problem. As it is said before this vehicle cannot have shocks-absorbing; due this, the natural frequencies have to be different to the frequencies due to the irregularities of the track.

Modal analysis

To obtain the natural frequencies of the structure a model in the Software SAP2000 is done. This software it is overall used in civil engineering; and its form of work is transforming any frame in a mass and a spring; so in order to obtain an elastic model similar to the reality any frame is divided in more frames; obtaining a model formed by 131 frames, and 128 points. In this model it is not included all the mass; it is included a medium mass; 70 kg in batteries; 80 kg for the pilot; and 40 kg like an additional mass.

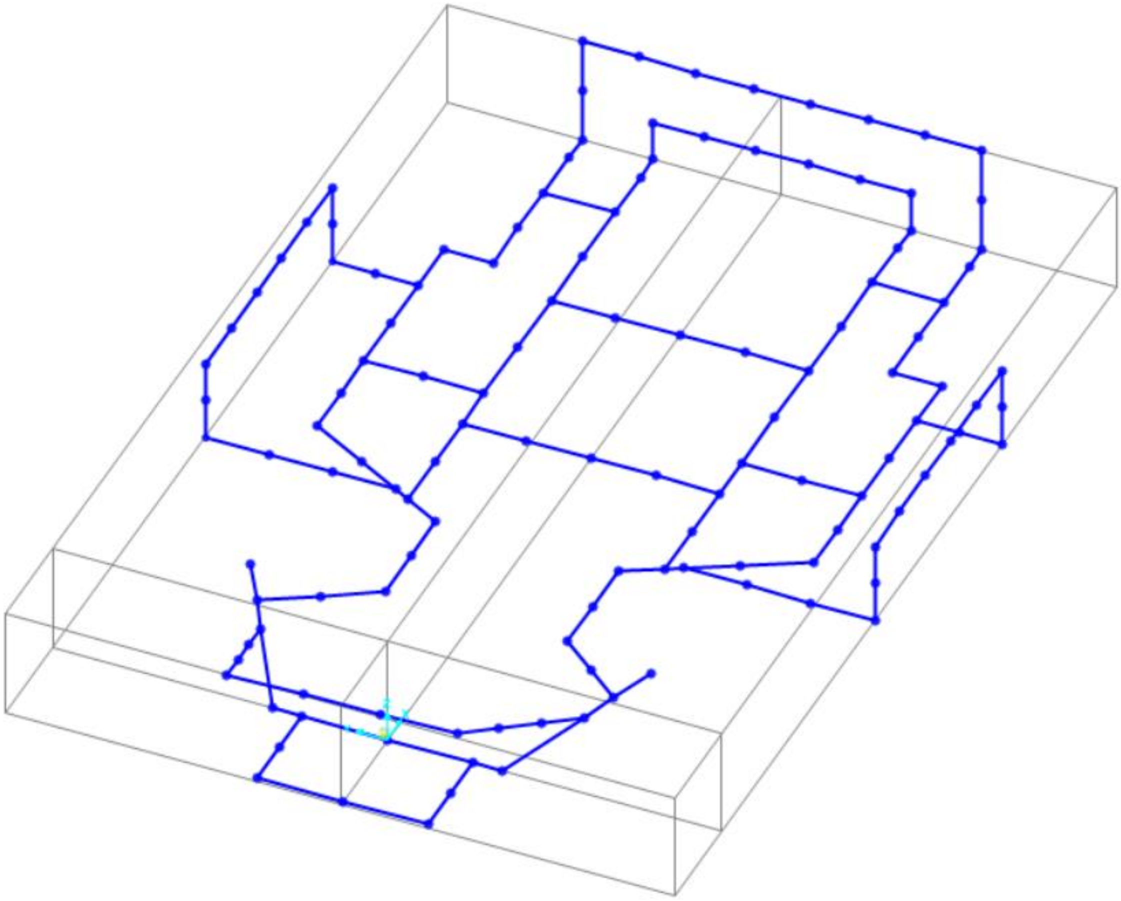


Figure 39: SAP2000 Model

With this model the frequencies of the structure are obtained; for the six rigid modes and the 3 principal dynamic modes:

Mode		f [Hz]
Static	1	8,00E-06
	2	2,00E-05
	3	2,50E-05
	4	2,60E-05
	5	2,70E-05
	6	2,90E-05
Dynamic	7	28,4
	8	35,3
	9	52,1

Table 23: Natural frequencies

In order to know if it will be a problem of resonances in the kart; to the model are introduced 4 frames that represents the pneumatics; but like it is said before, this program works transforming the frames; due this the pneumatic is designed like:

$$K_n = \frac{E \cdot A}{L} \quad (\mathbf{8.9.1})$$

E = Young modulus

A = Area of the frame

L = Length of the frame

The K of a normal pneumatic it is around 20000 N/m; in that case, like the pneumatics are smaller than in a normal kart; I am going to use a K of 15000 N/m. So, if the frames introduced are of 0,2 m with an area of 1 m^2 (this numbers are not reals; they are only used to be able to introduce “spring” in the model); the modulus have to be:

$$\mathrm{E} = \frac{15000 \cdot 0{,}2}{1} = 3000\ \mathrm{N/mm^2}$$

Another important factor is the mass of these frames; like its volume it is known; and its mass have to be similar to 1/4 of the non-suspended masses; more or less 4 kg per frame; the density of these frames has to be:

$$\delta = \frac{4 \cdot 9{,}81}{1 \cdot 0{,}2} \simeq 200\ \mathrm{N/m^3}$$

Introducing these frames, the final model and the frequencies for the dynamic models are:

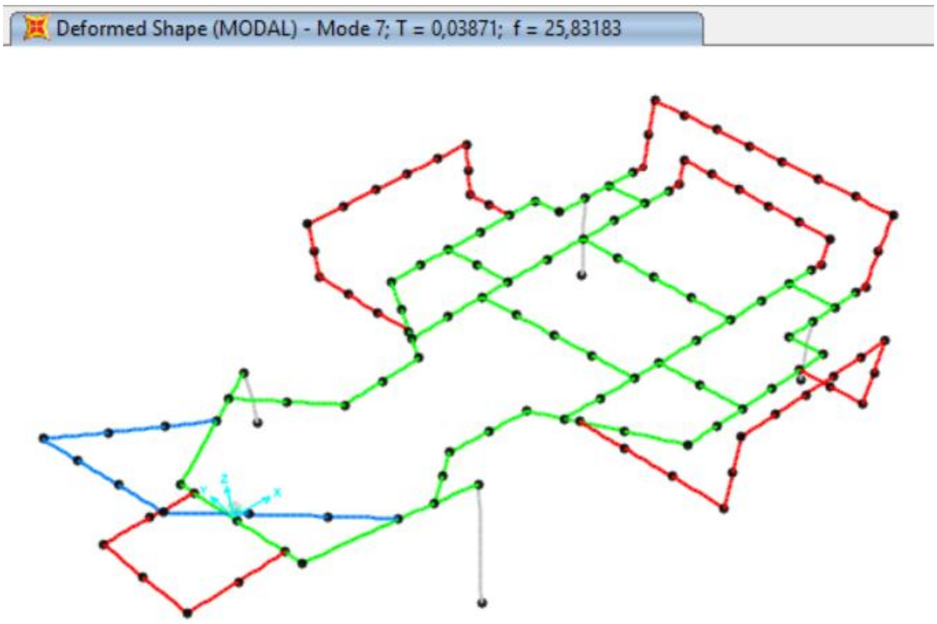


Figure 40: 1st dynamic mode

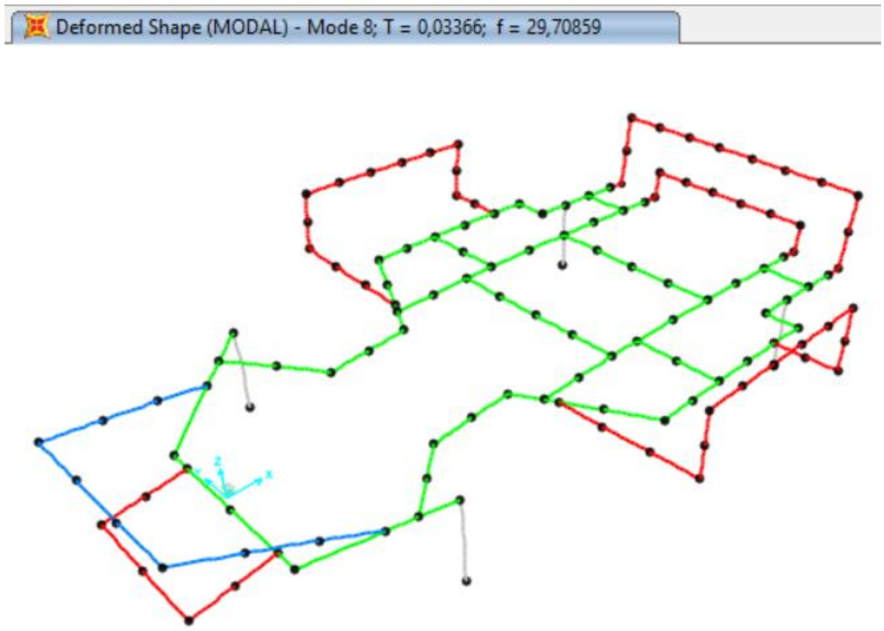


Figure 41: 2nd dynamic mode

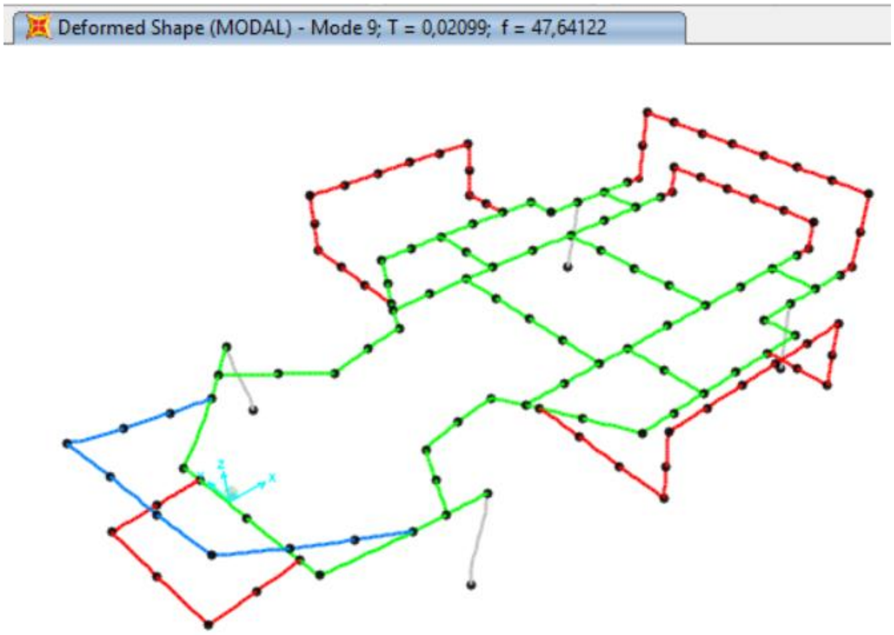


Figure 42: 3rd dynamic mode

Once that the natural frequences of the model are knowed the problem of the resonance could be analysed. This problem happens when an variable force (or a desplacment) exite the sistem at the same frequency of a natural frequence during a peride of time. The important frequencies of exitation are that produced for a wave length equal to 1,2,3… times the wheelbase; due to they exit the vertical mode; and for a wave length equal to 0,5; 1,5; 2,5… times the wheelbase; due to they exit the turn arround the transversal axis.

The frequences could be calculated like:

$$f = \frac{1}{\lambda} \cdot v \quad (\mathbf{8.9.2})$$

Like the problem of the resonance happens when the forces are appliqued during a period of time with the same frequency; this problem have to be analysed during a period of constant velocity. The sections in which the velocity is constant in a circuit are not usual; overall there could be find in some straight part when the kart goes at maximum velocity.

The maximum speed is limited in some countries for amateur drivers. This limitation is around the 70 and the 80 km/h. Due these three velocities are going to be analysed 70, 80 and 135 km/h (the maximum theorical speed). Comparing the natural frequencies of the kart; and the excitation frequencies, there are not any frequency that is an excitation and a natural frequency:

Speed	**[km/h]**	**70,0**	**80,0**	**135,0**
	[m/s]	**19,4**	**22,2**	**37,5**
Multiple	**λ [m]**	**f [Hz]**		
0,5	0,525	37,0	42,3	71,4
1	1,05	18,5	21,2	35,7
1,5	1,575	12,3	14,1	23,8

Table 24: Excitation frequencies

Mode		**f [Hz]**
Dynamic	7	25,8
	8	29,7
	9	47,6

Table 25: Natural frequencies of the dynamic modes

At this point a clarification has to be done; these calculations have done with medium values, not real values; so, in the case of the vibrations where a change could modify all the results it could be interesting to redo this process introducing the real values. If there cannot be done; for any reason; to changes could be done to reduce the effect of the resonances; introduce an additional mass; or change the tyres could be a solution.

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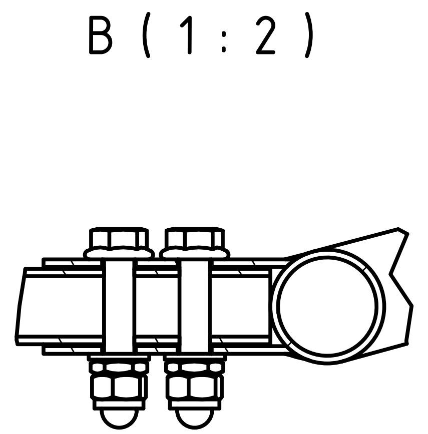
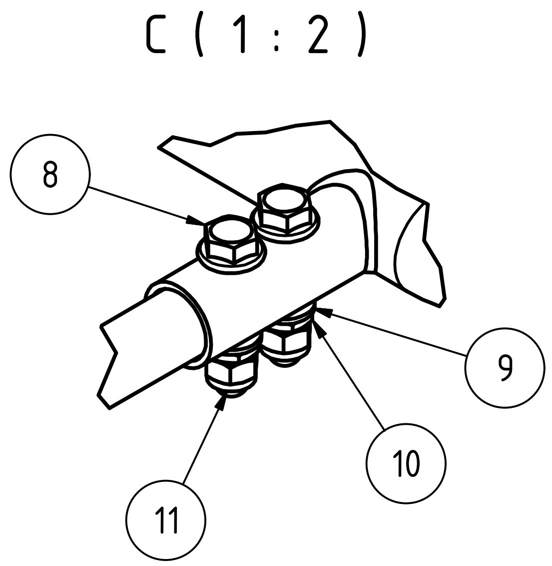
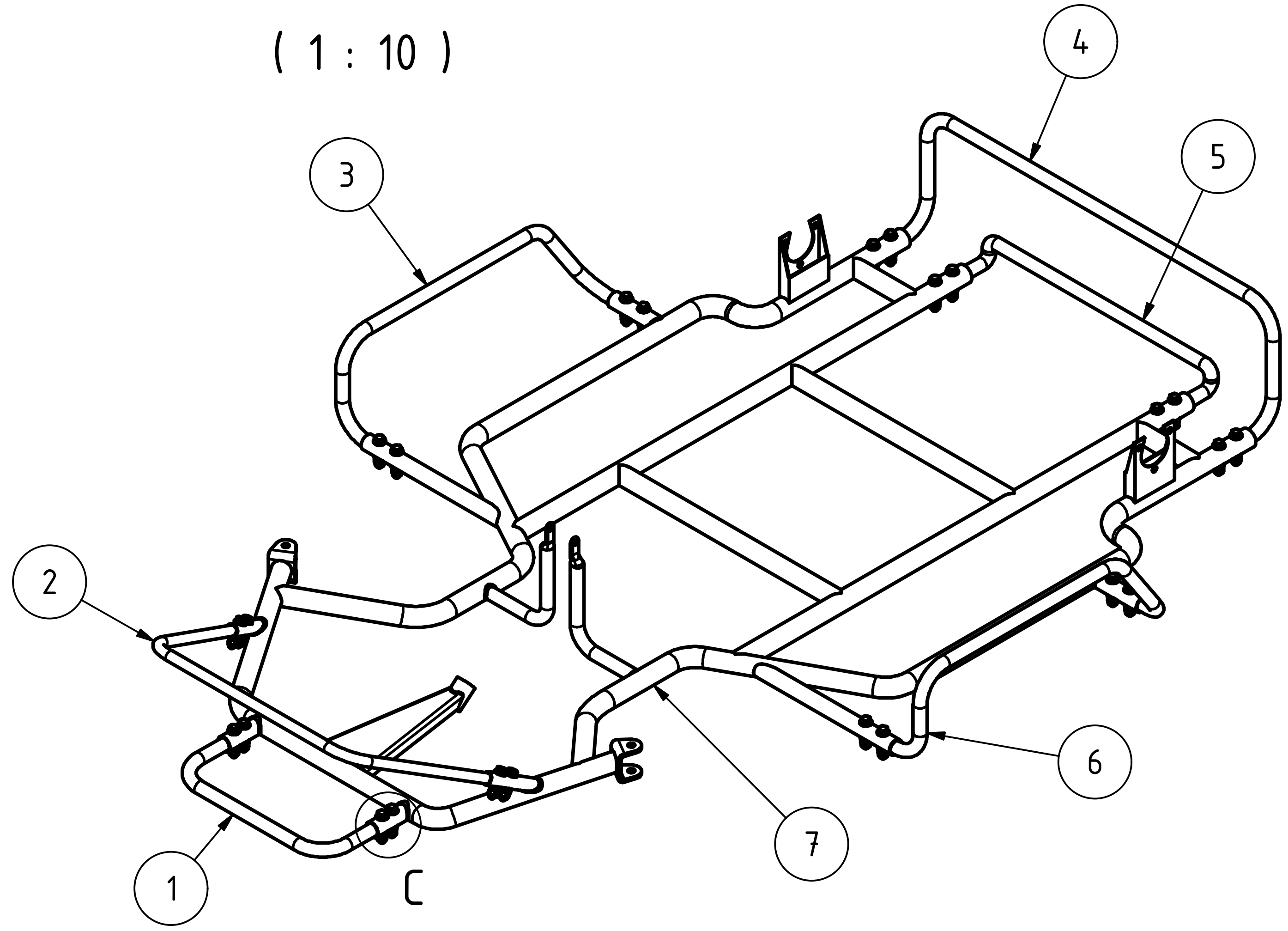
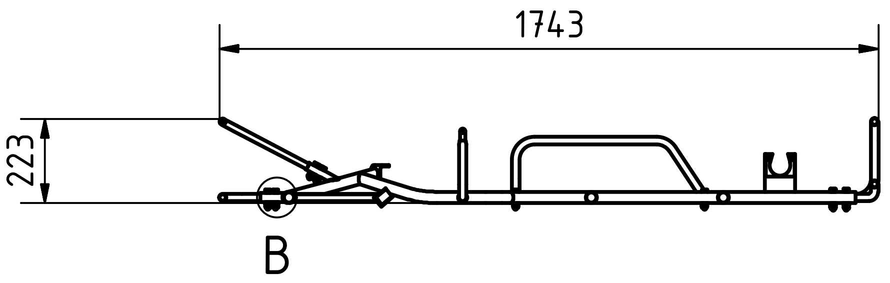
Design and analysis of the chassis of an electric kart

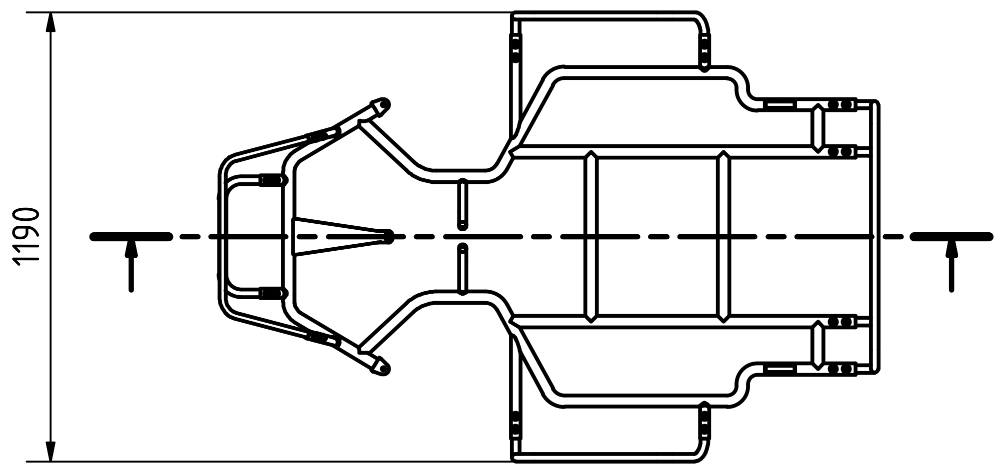
2. Technical drawings





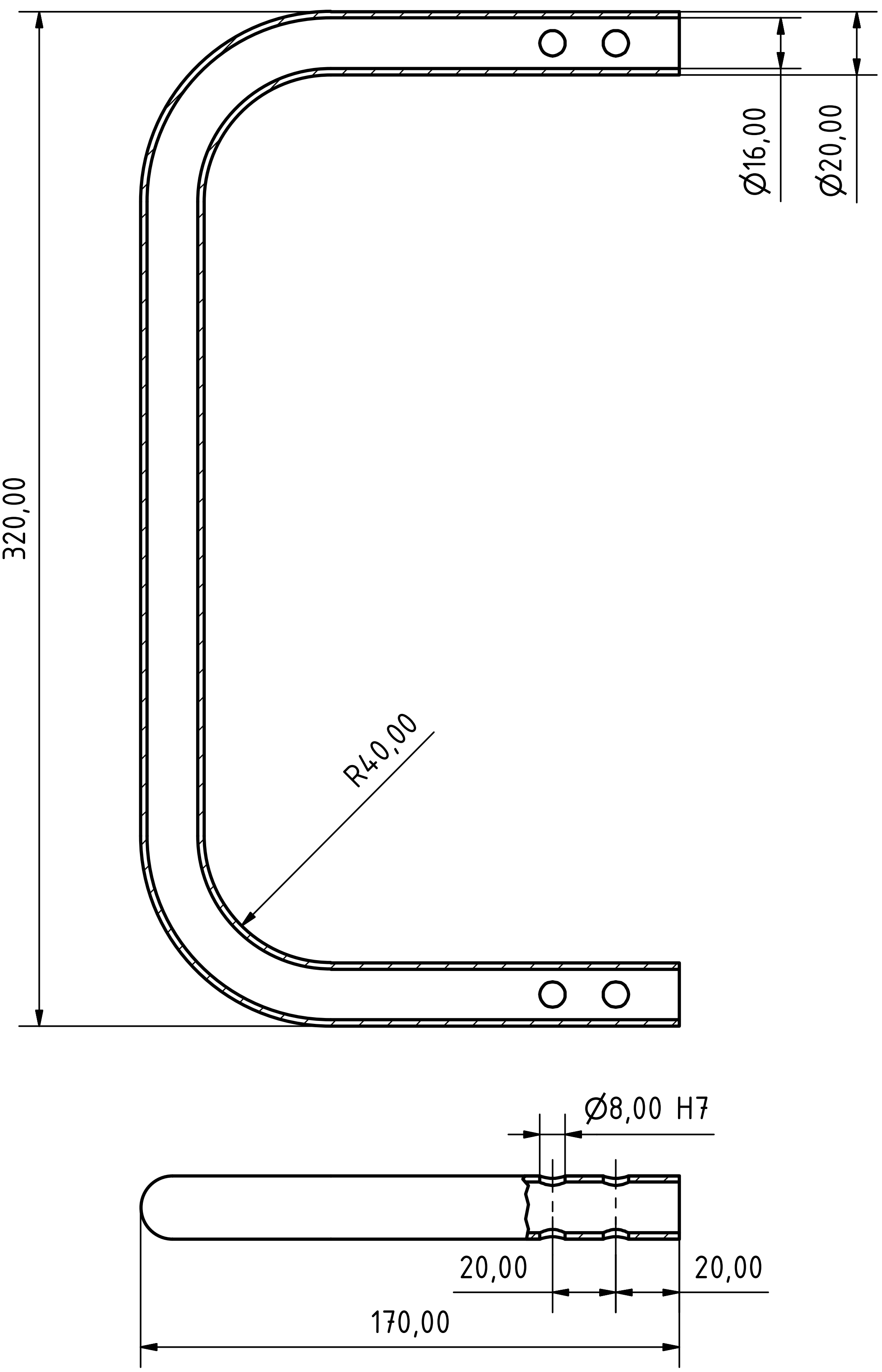




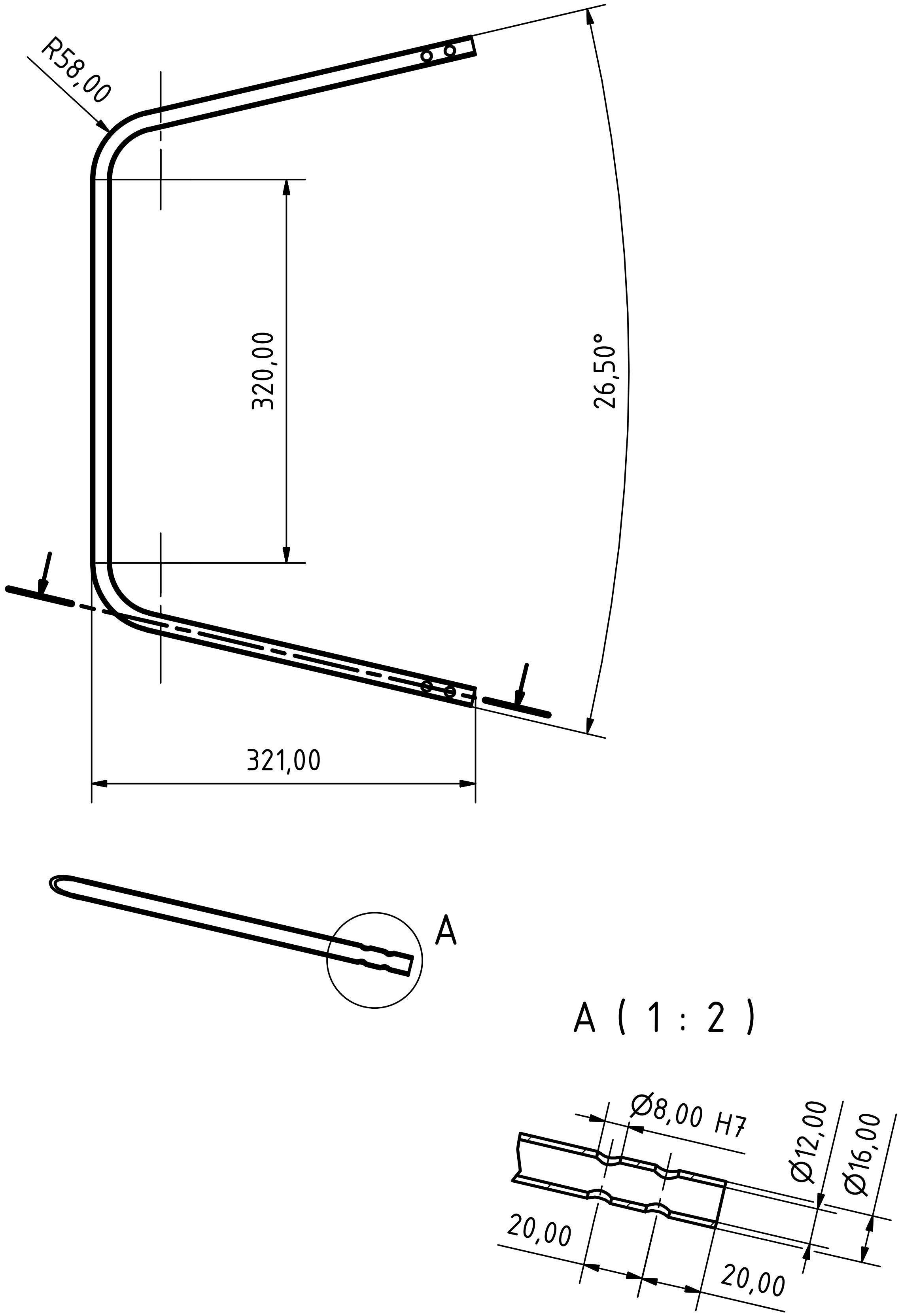


COMPONENTS			
ELEMENT	QUANTITY	COMPONENT	MATERIAL
1	1	Lower Frontal Bumper	Steel AISI 4130
2	1	Upper Frontal Bumper	Steel AISI 4130
3	1	Lateral Bumper R	Steel AISI 4130
4	1	Upper Back Bumper	Steel AISI 4130
5	1	Lower Back Bumper	Steel AISI 4130
6	1	Lateral Bumper L	Steel AISI 4130
7	1	Chassis	Steel AISI 4130
8	24	EN 1665 - M8 x 35	Mild Steel
9	24	DIN 126 - 9	Mild Steel
10	24	ISO 4035 - M8	Stainless Steel, 440C
11	24	DIN 986 - M8	Mild Steel

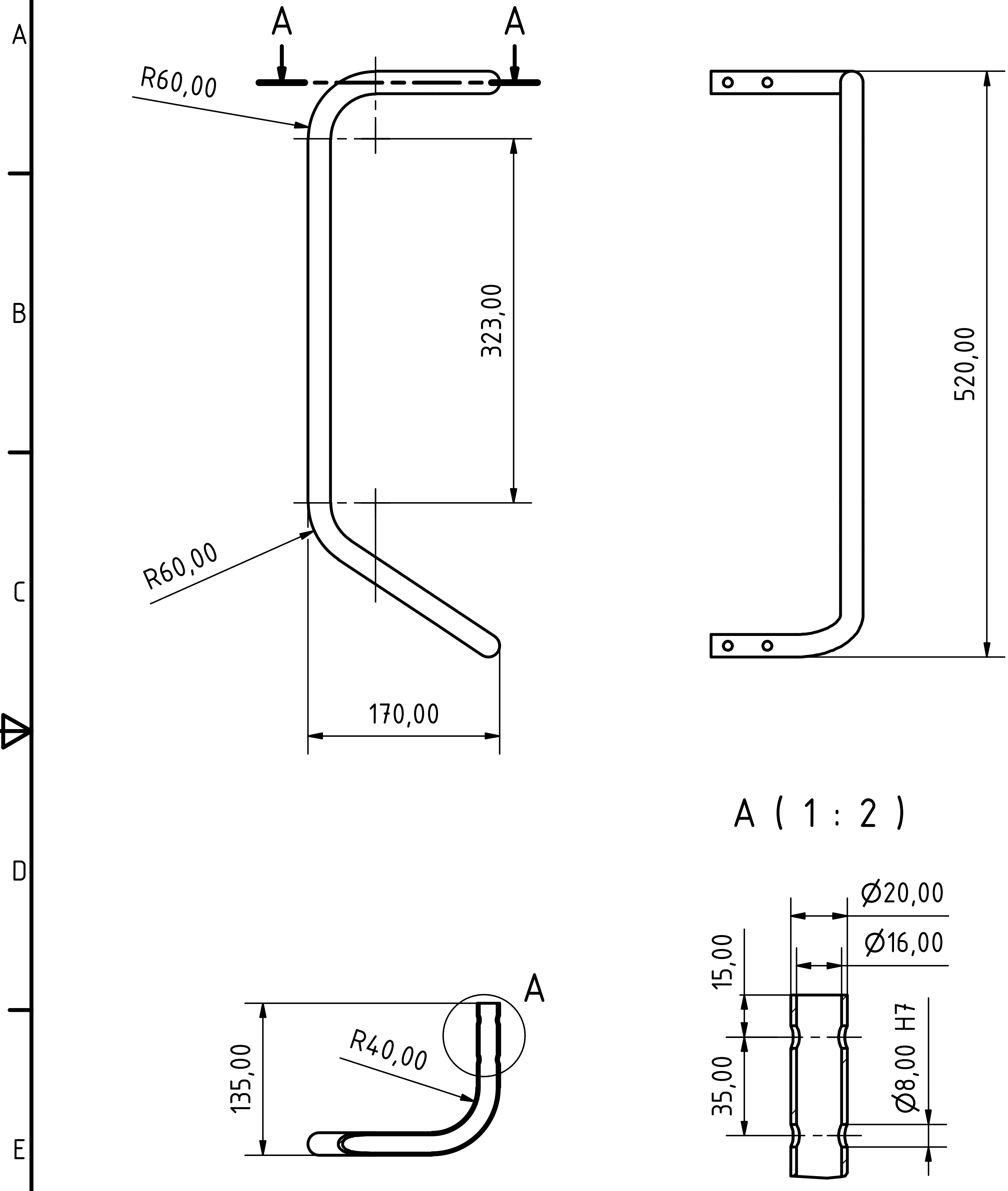
University: ETSID - UPV		Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 20	Project: Chassis EK1		Number: 1.00.A
			Replaces:
Format: A3	Plan: Overall Drawing		Replaced by:



University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 28/06/2019
Scale: 1 : 2	Project: Chassis EK1	Number: 1.01.A
Format: A4	Plan: Lower Frontal Bumper	Replaces:
		Replaced by:



University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 5	Project: Chassis EK1	Number: 1.02.A
Format: A4	Plan: Upper Frontal Bumper	Replaces:
		Replaced by:



University: ETSID - UPV

Name: Adrián Altamira Peña

Date: 29/06/2019

Scale: 1 : 5

Format: A4

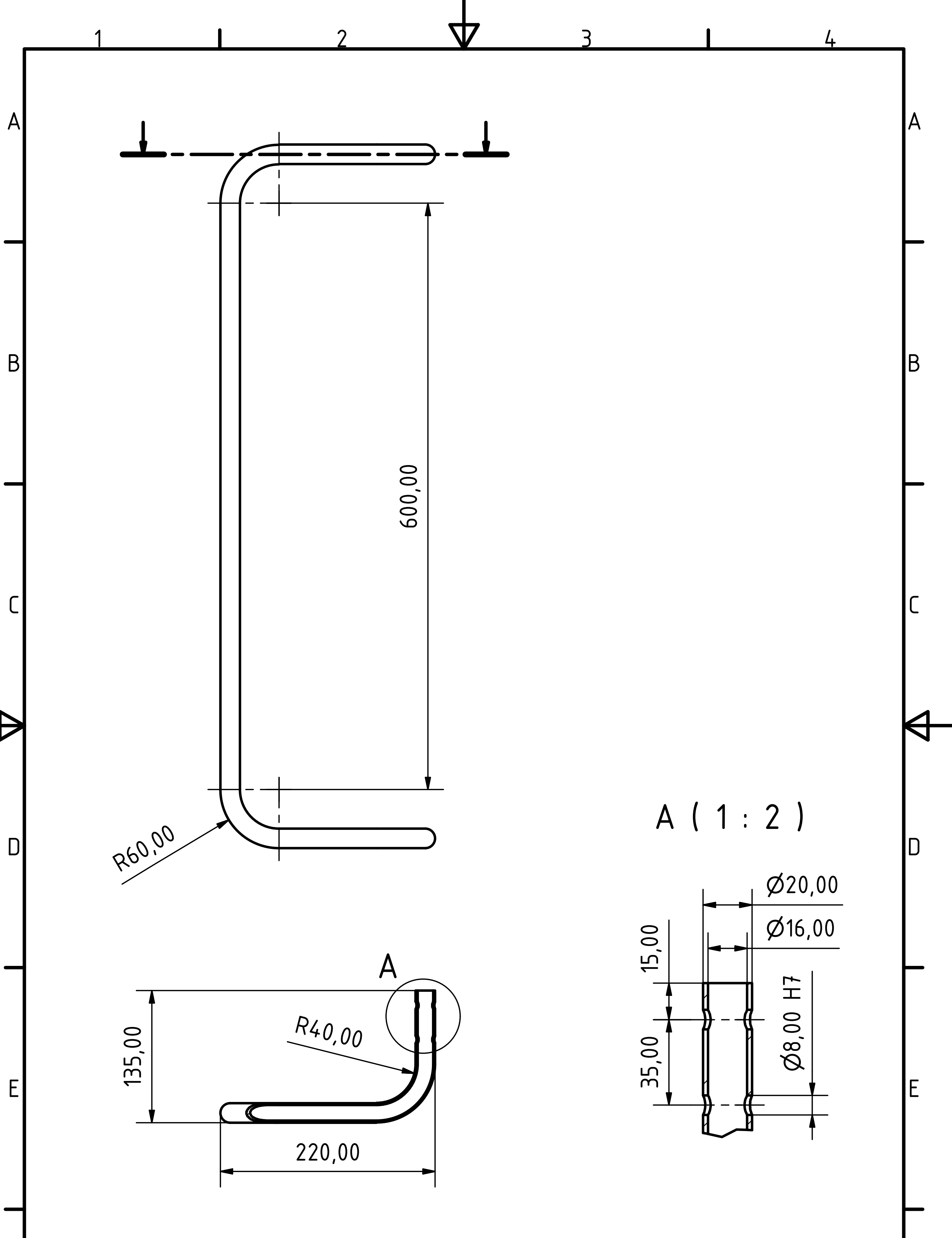
Project: Chassis EK1

Plan: Lateral Bumper R

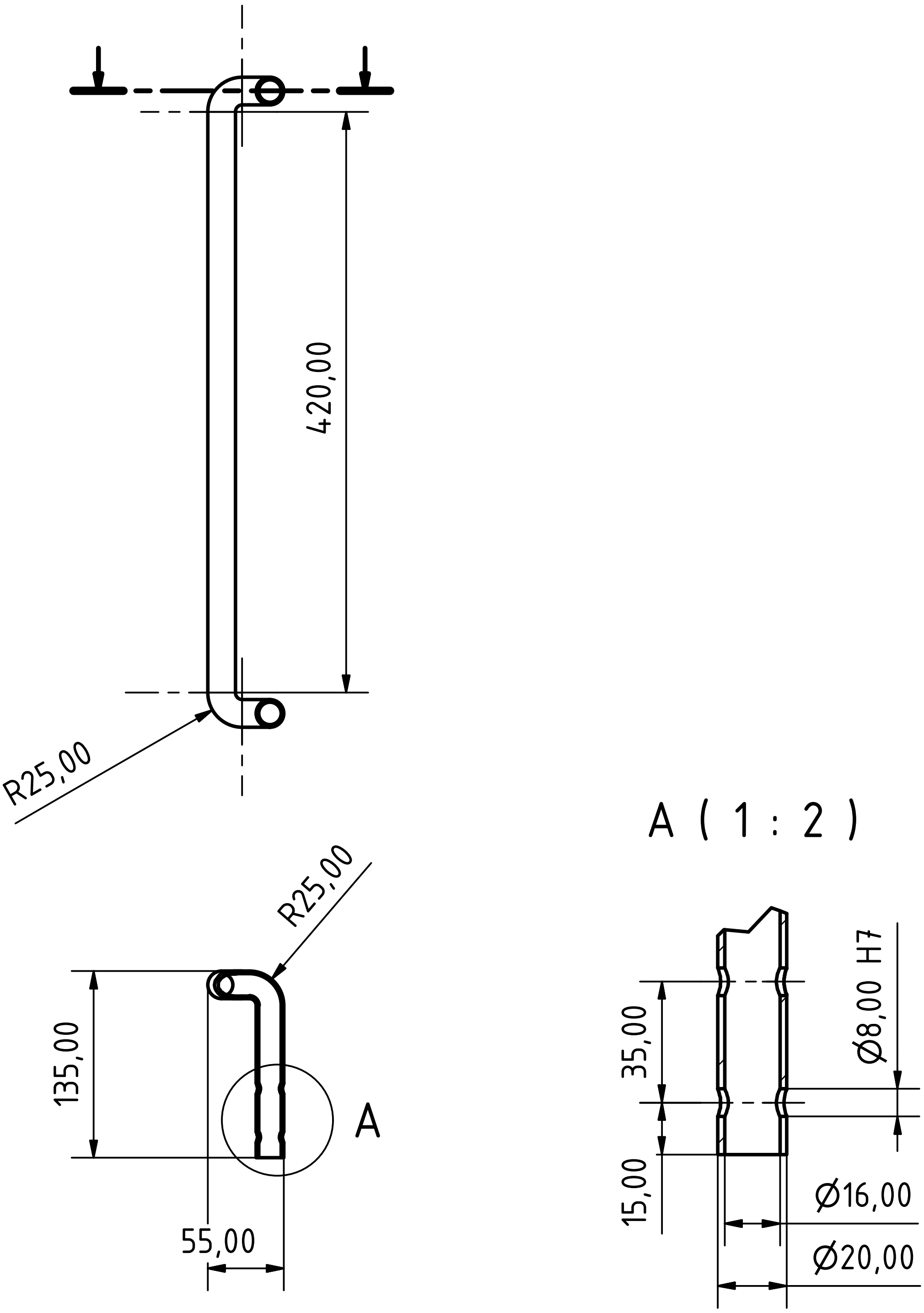
Number: 1.03.A

Replaces:

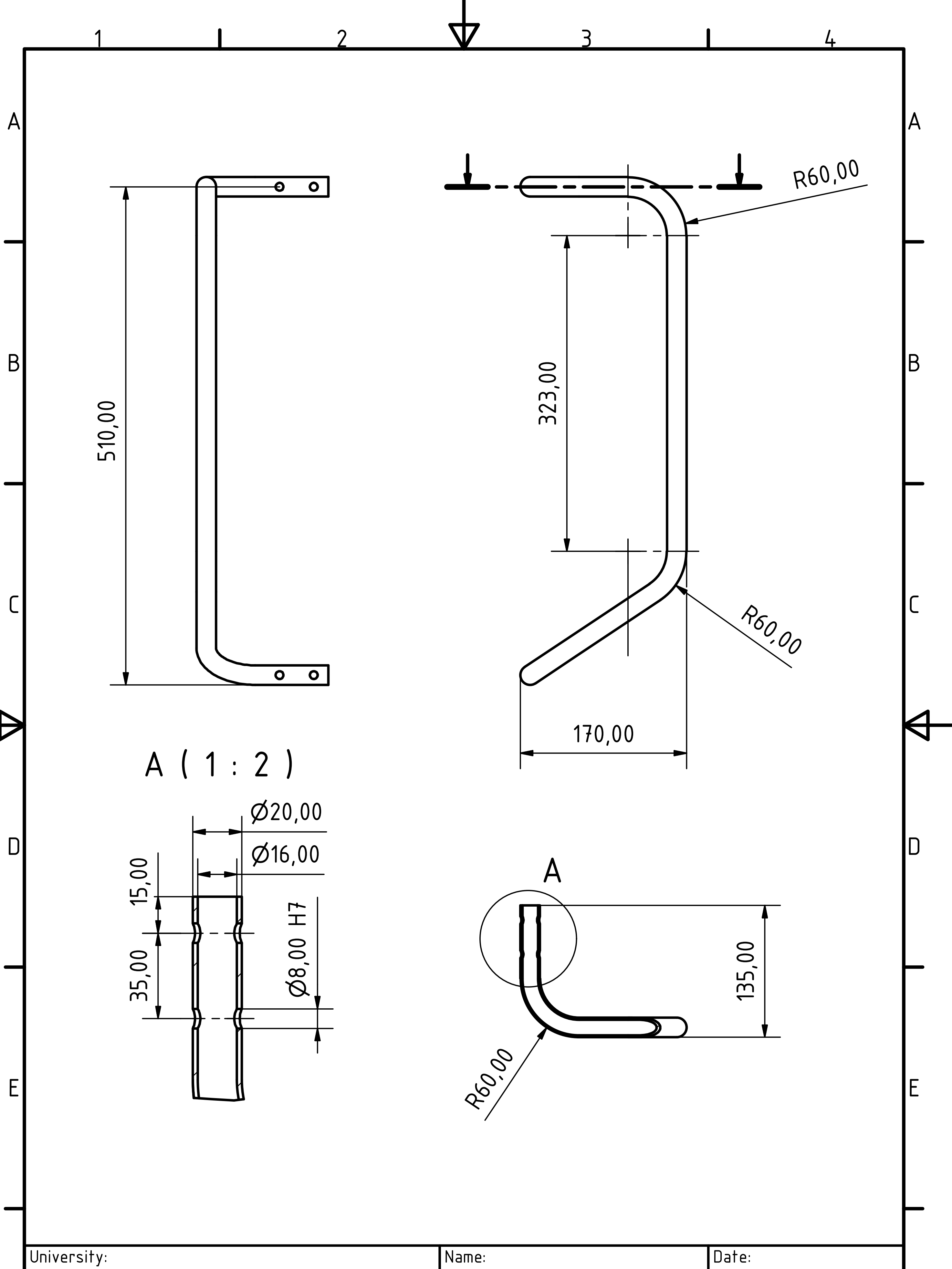
Replaced by:



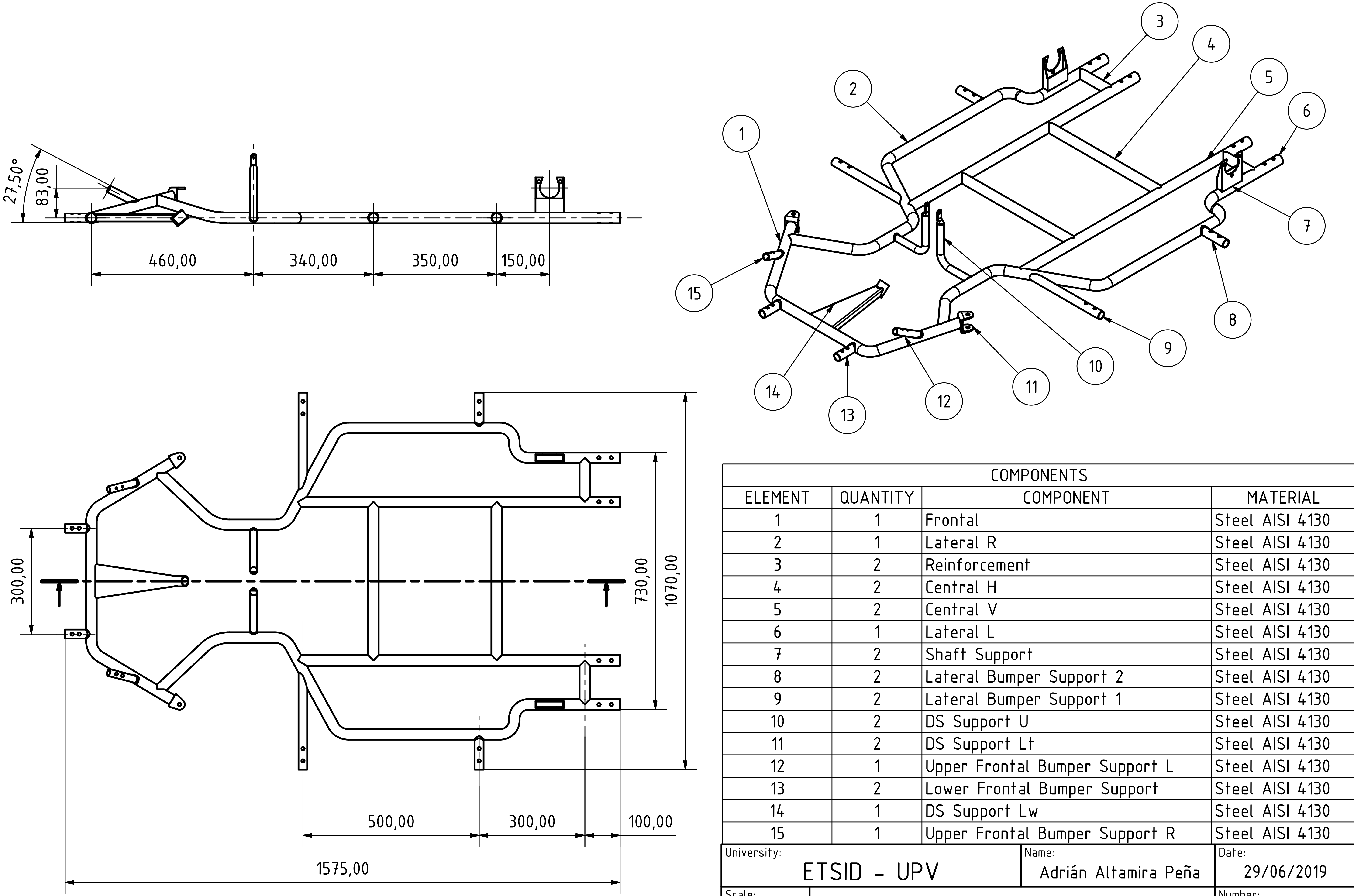
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 5	Project: Chassis EK1	Number: 1.04.A
Format: A4	Plan: Upper Back Bumper	Replaces:
		Replaced by:



University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 5	Project: Chassis EK1	Number: 1.05.A
Format: A4	Plan: Lower Back Bumper	Replaces:
		Replaced by:

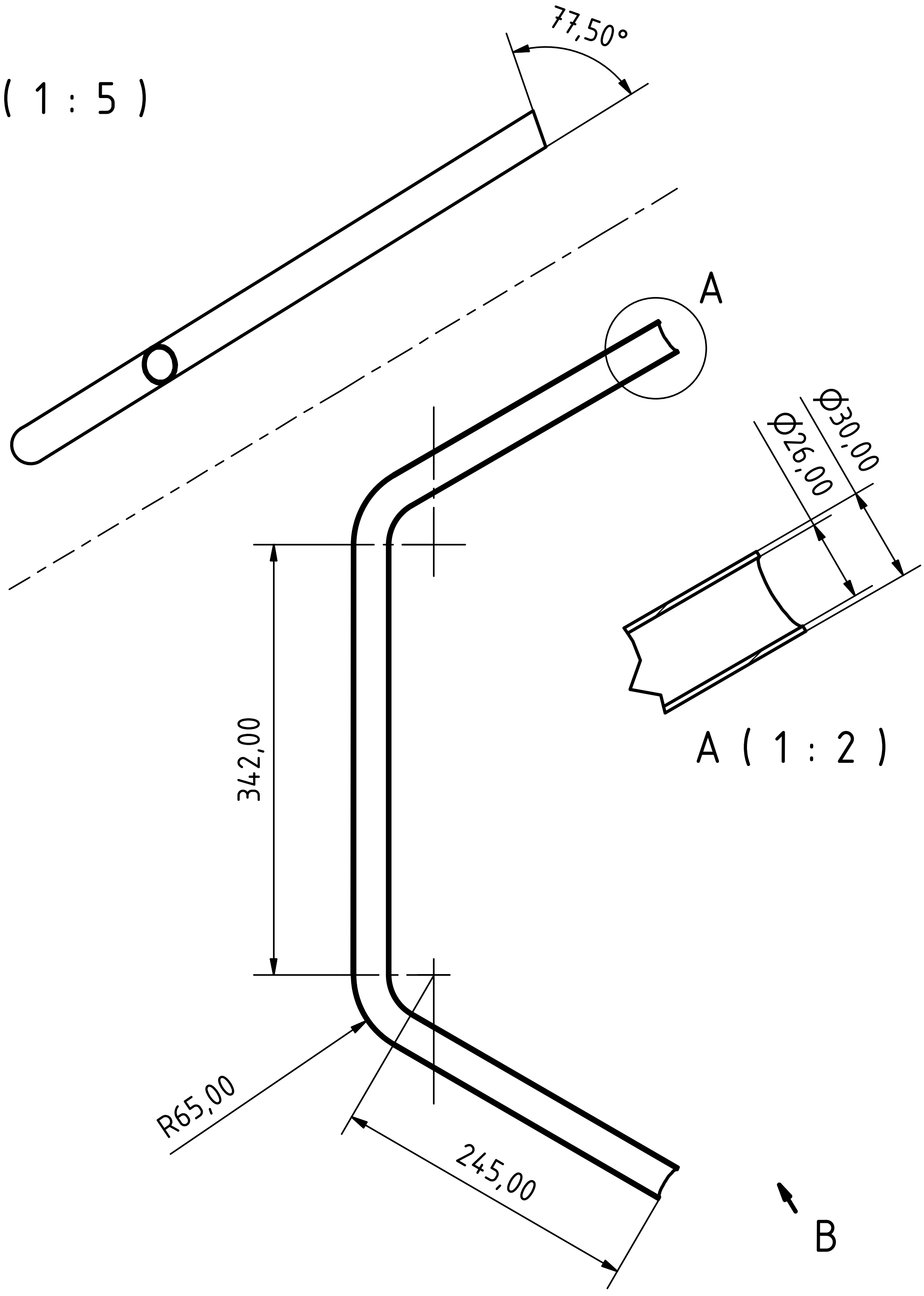


University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 5	Project: Chassis EK1	Number: 1.06.A
Format: A4	Plan: Lateral Bumper L	Replaces:
		Replaced by:

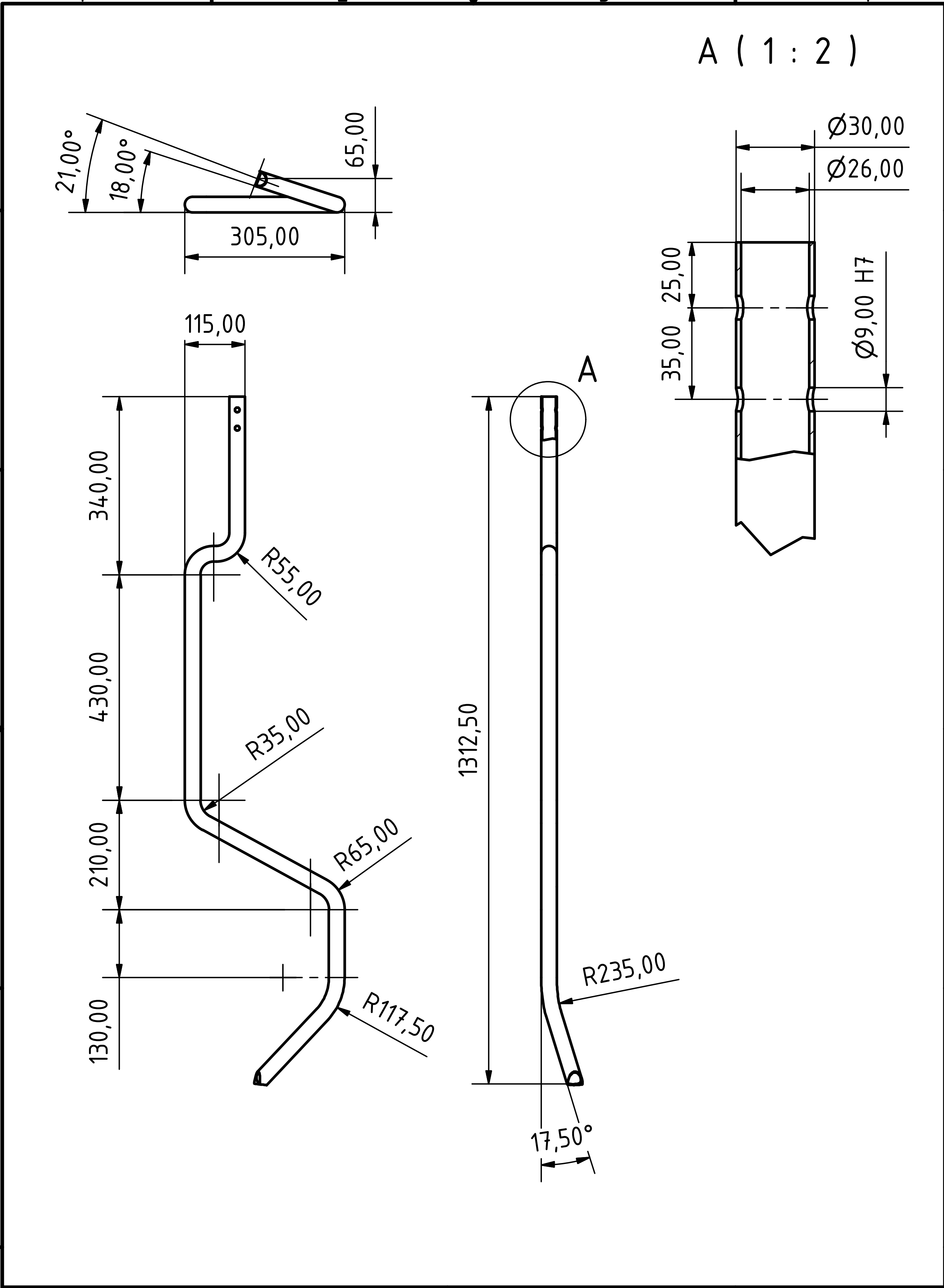


COMPONENTS			
ELEMENT	QUANTITY	COMPONENT	MATERIAL
1	1	Frontal	Steel AISI 4130
2	1	Lateral R	Steel AISI 4130
3	2	Reinforcement	Steel AISI 4130
4	2	Central H	Steel AISI 4130
5	2	Central V	Steel AISI 4130
6	1	Lateral L	Steel AISI 4130
7	2	Shaft Support	Steel AISI 4130
8	2	Lateral Bumper Support 2	Steel AISI 4130
9	2	Lateral Bumper Support 1	Steel AISI 4130
10	2	DS Support U	Steel AISI 4130
11	2	DS Support Lt	Steel AISI 4130
12	1	Upper Frontal Bumper Support L	Steel AISI 4130
13	2	Lower Frontal Bumper Support	Steel AISI 4130
14	1	DS Support Lw	Steel AISI 4130
15	1	Upper Frontal Bumper Support R	Steel AISI 4130

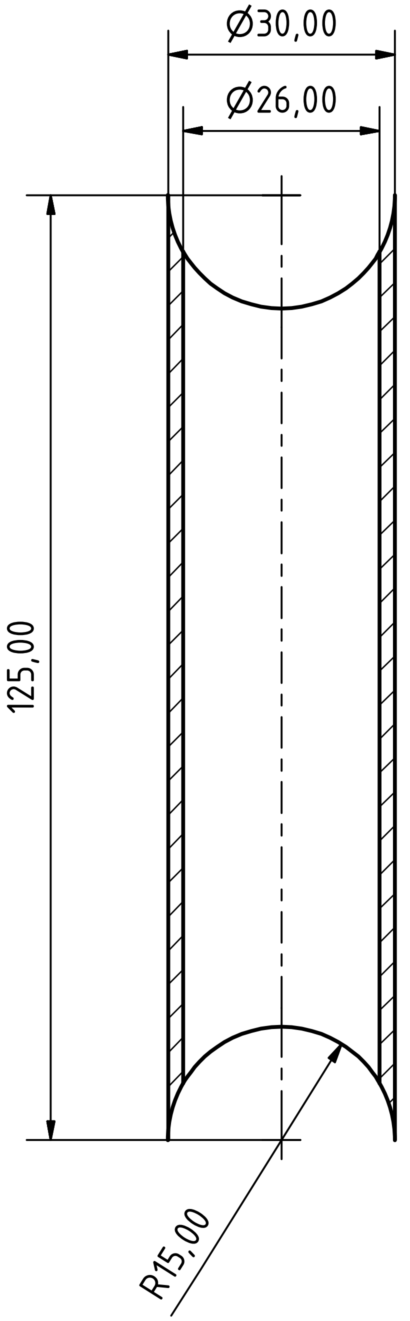
University: ETSID - UPV		Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 10	Project: Chassis EK1		Number: 1.07.A
Format: A3	Plan: Chassis		Replaces:
			Replaced by:



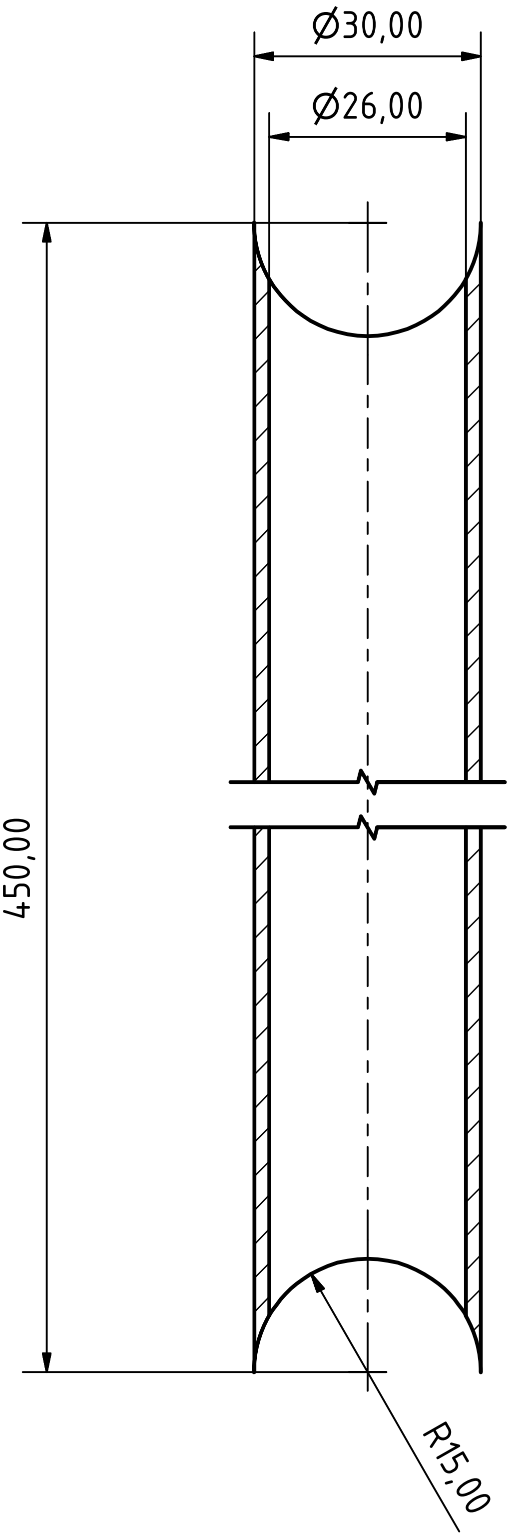
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 5	Project: Chassis EK1	Number: 7.01.A
Format: A4	Plan: Frontal	Replaces: / Replaced by:



University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 10	Project: Chassis EK1	Number: 7.02.A
Format: A4	Plan: Lateral R	Replaces:
		Replaced by:

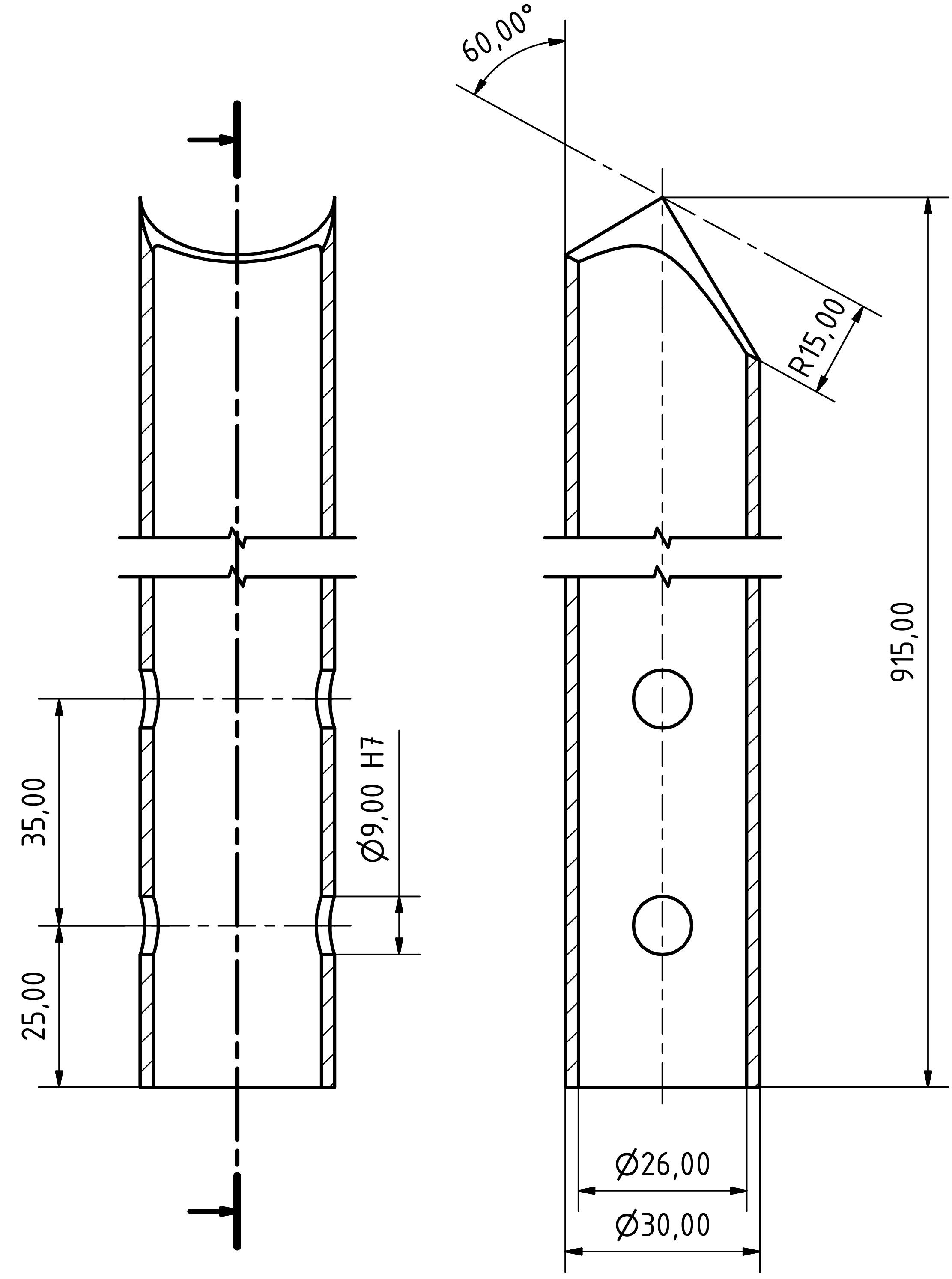


University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 1	Project: Chassis EK1	Number: 7.03.A
Format: A4	Plan: Reinforcement	Replaces:
		Replaced by:



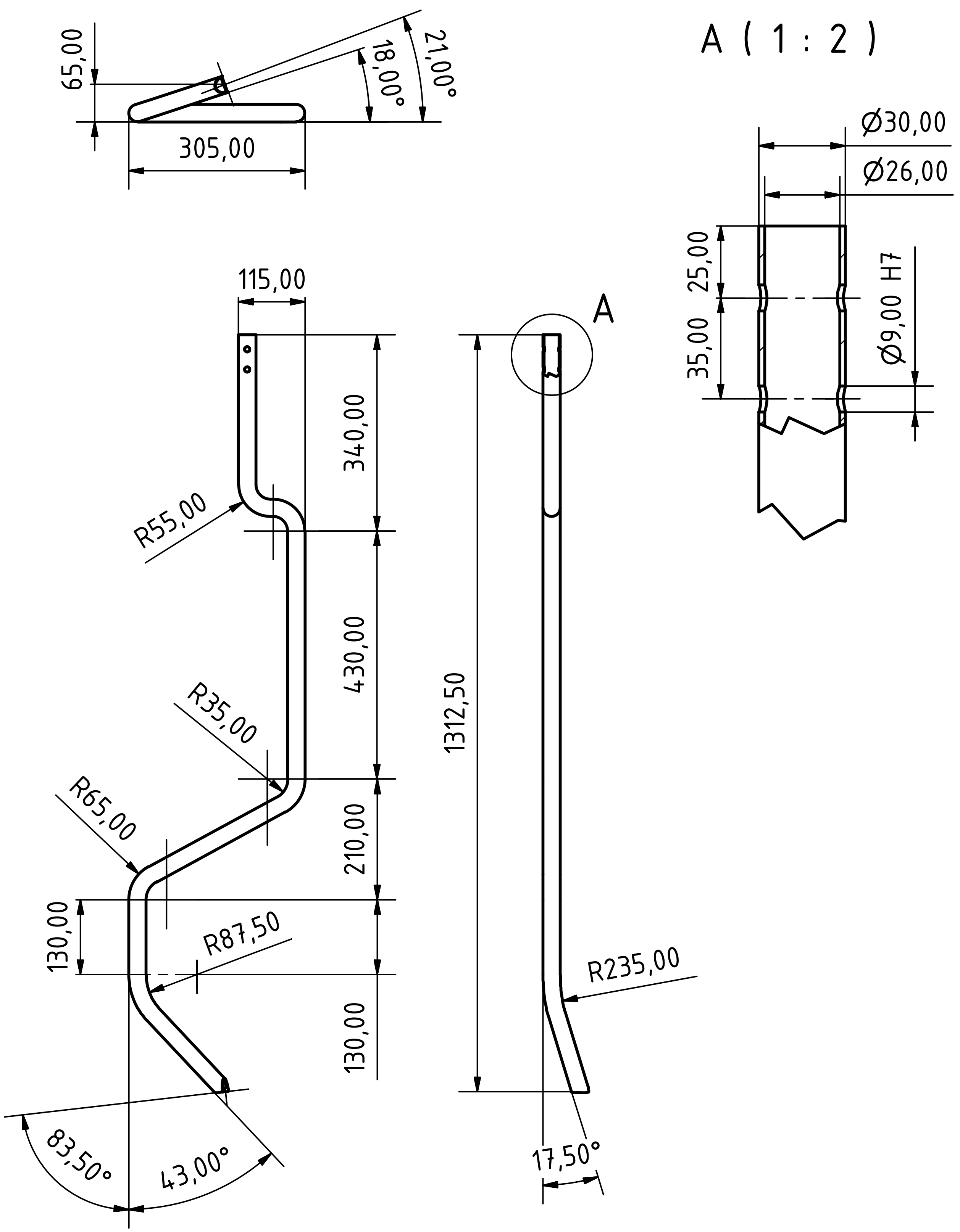
University:	Name:	Date:
ETSID - UPV	Adrián Altamira Peña	29/06/2019

Scale: 1 : 1	Project: Chassis EK1	Number: 7.04.A
Format: A4	Plan: Central H	Replaces:
		Replaced by:

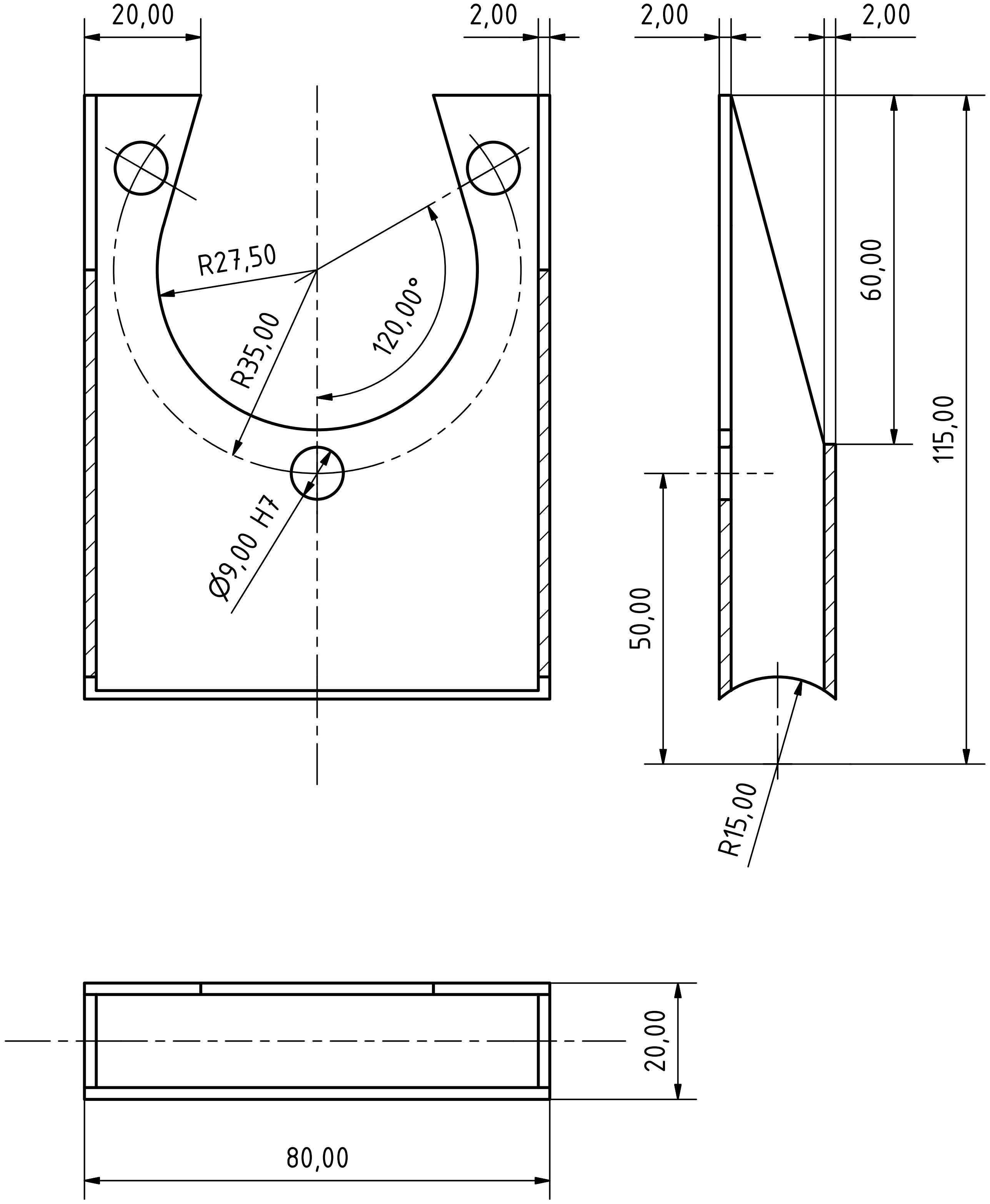


University:	Name:	Date:
ETSID - UPV	Adrián Altamira Peña	29/06/2019

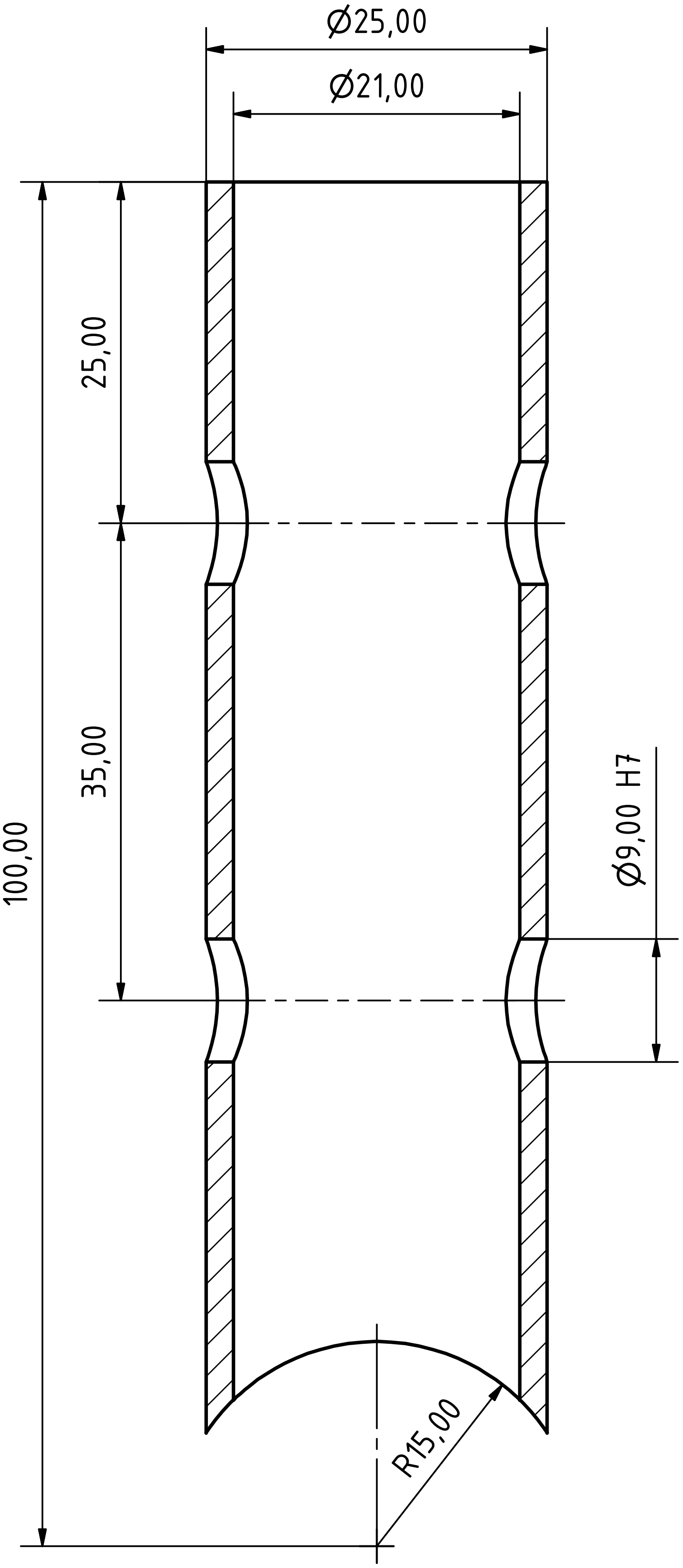
Scale: 1 : 1	Project: Chassis EK1	Number: 7.05.A
Format: A4	Plan: Central V	Replaces:
		Replaced by:



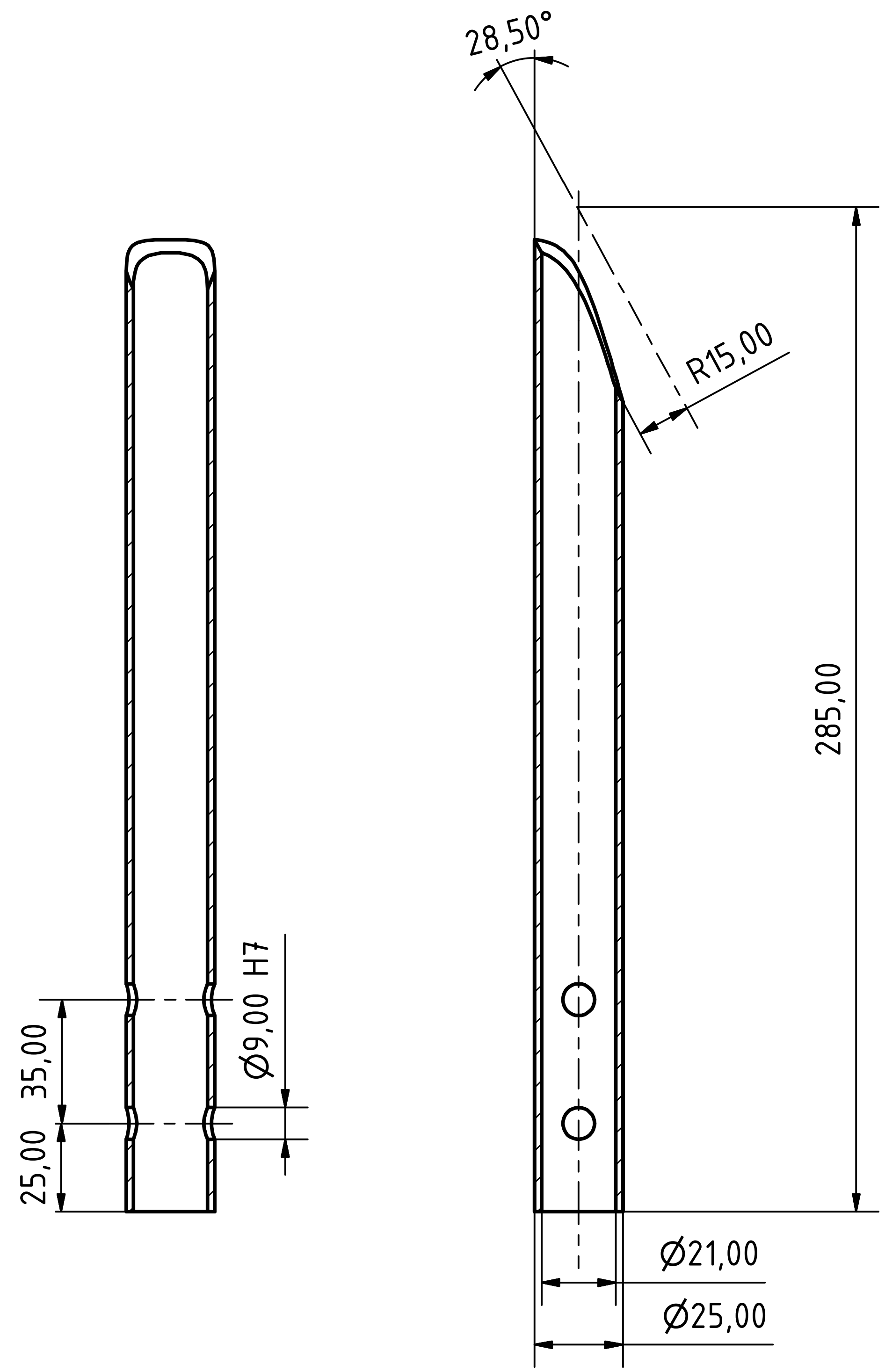
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 10	Project: Chassis EK1	Number: 7.06.A
Format: A4	Plan: Lateral L	Replaces:
		Replaced by:



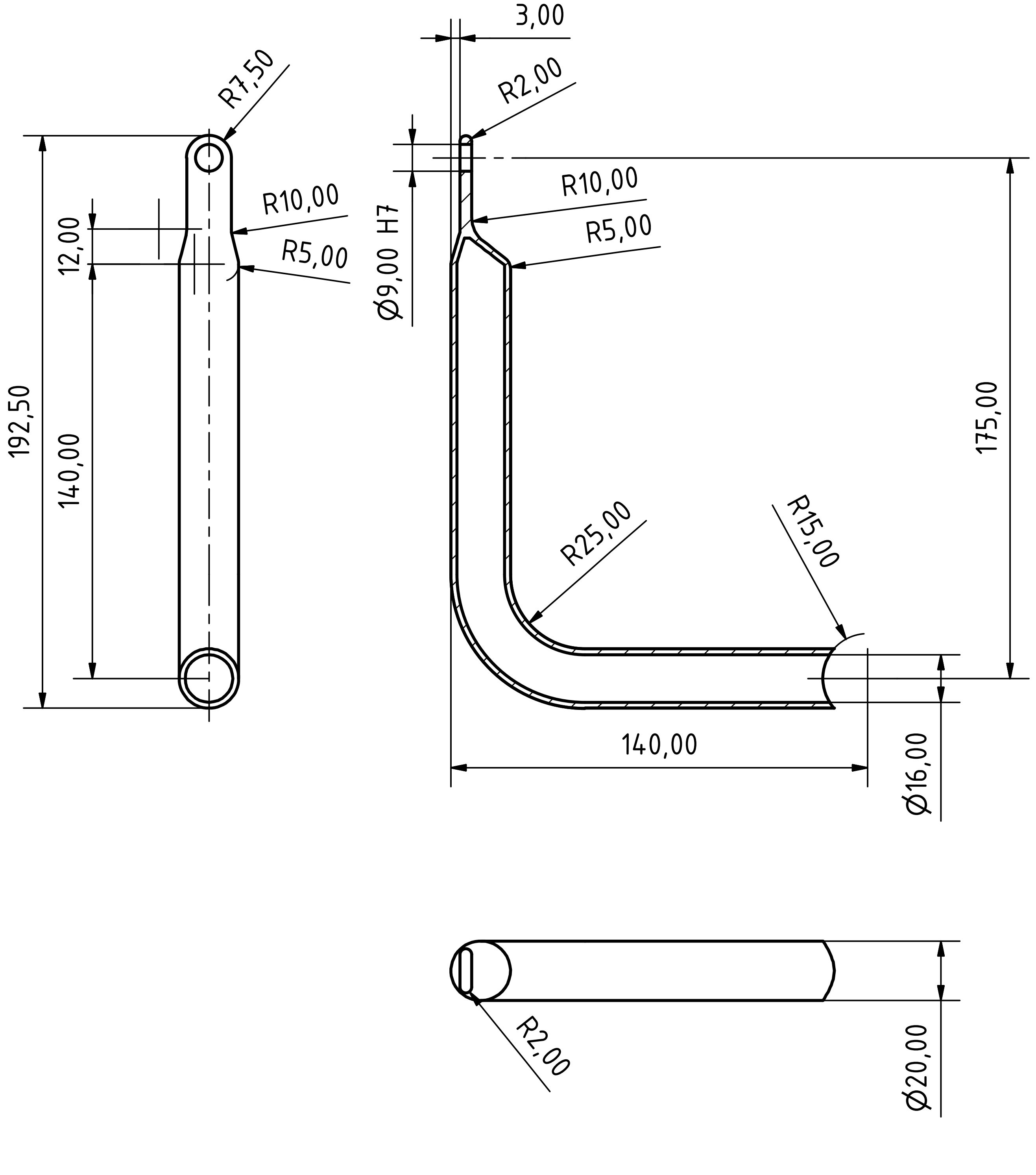
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 1	Project: Chassis EK1	Number: 7.07.A
Format: A4	Plan: Shaft Support	Replaces:
		Replaced by:



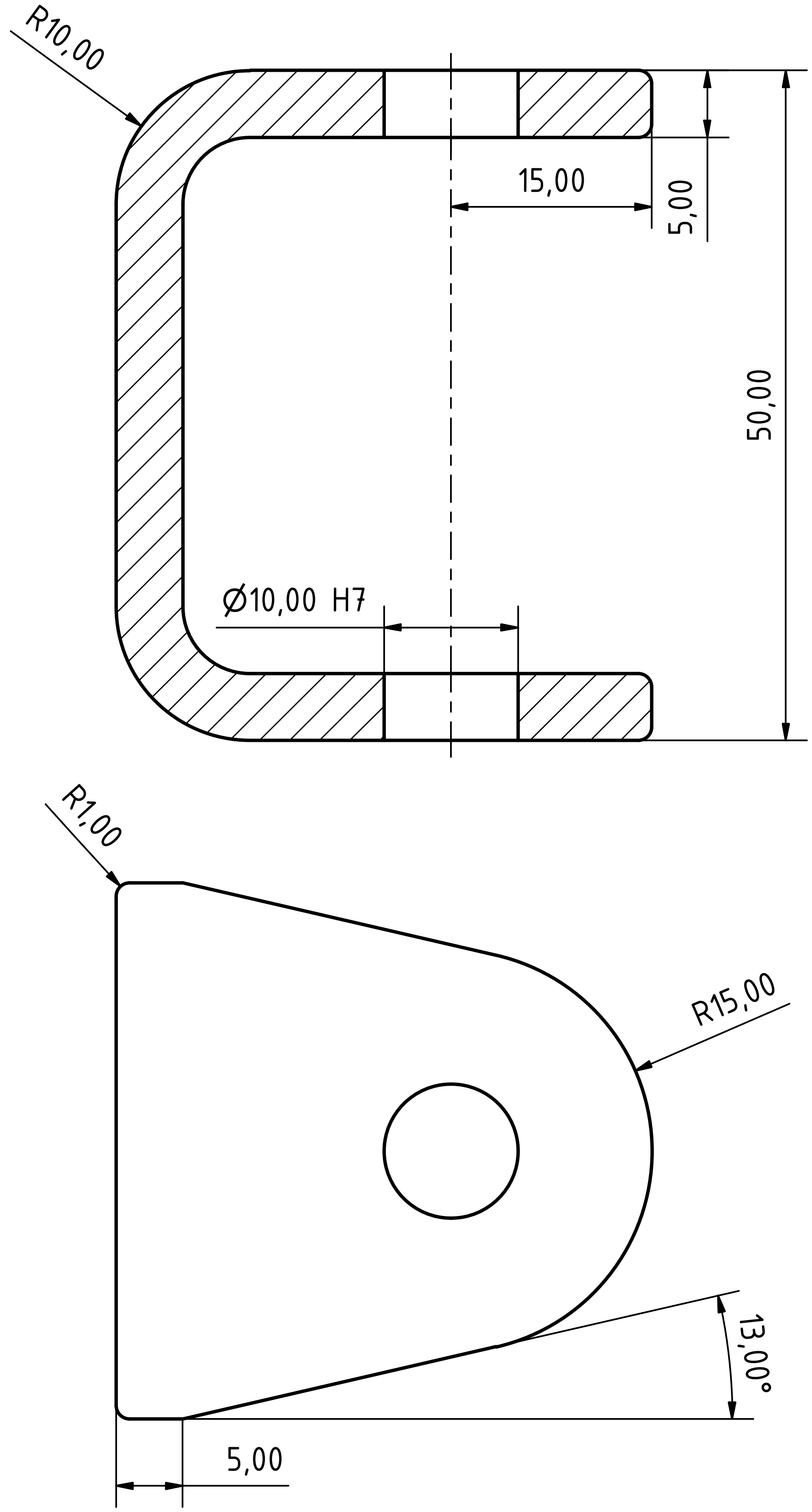
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 2 : 1	Project: Chassis EK1	Number: 7.08.A
Format: A4	Plan: Lateral Bumper Support 2	Replaces:
		Replaced by:



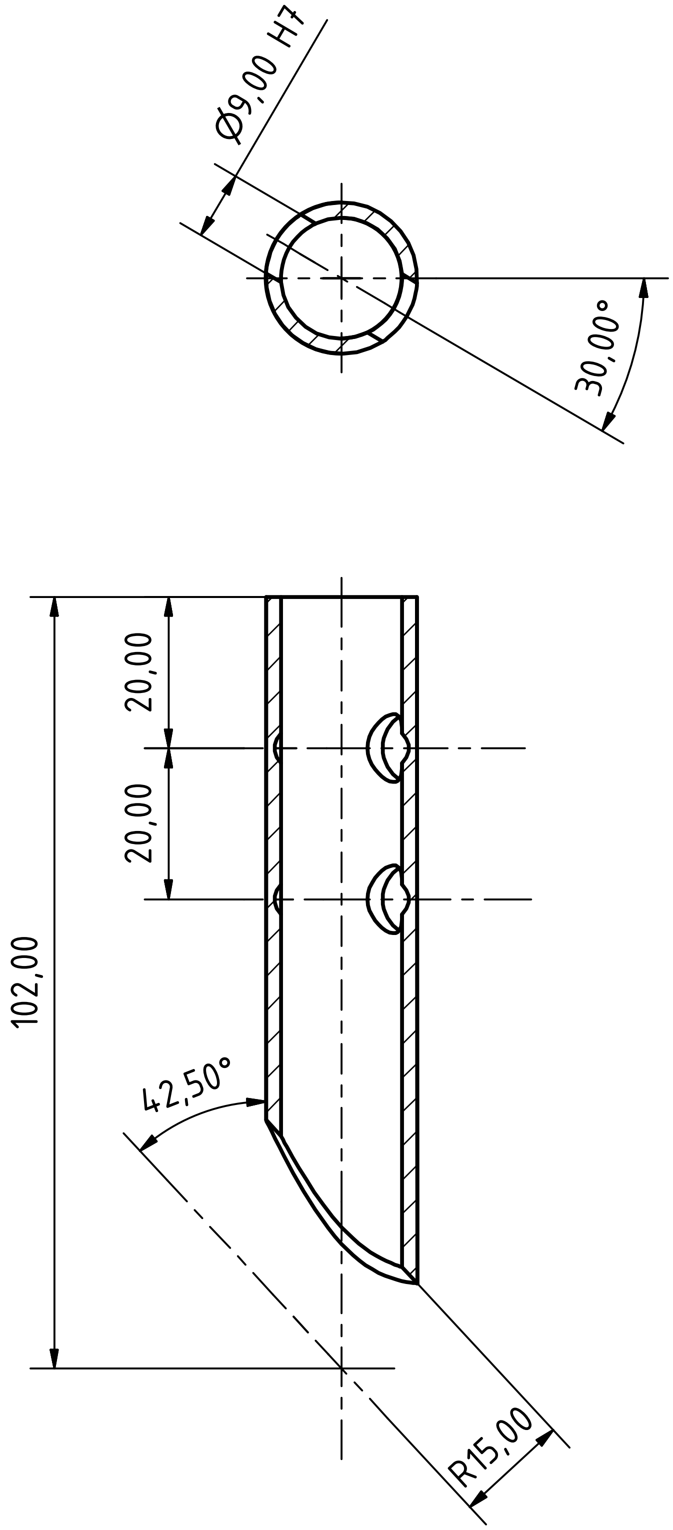
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 2	Project: Chassis EK1	Number: 7.09.A
Format: A4	Plan: Lateral Bumper Support 1	Replaces:
		Replaced by:



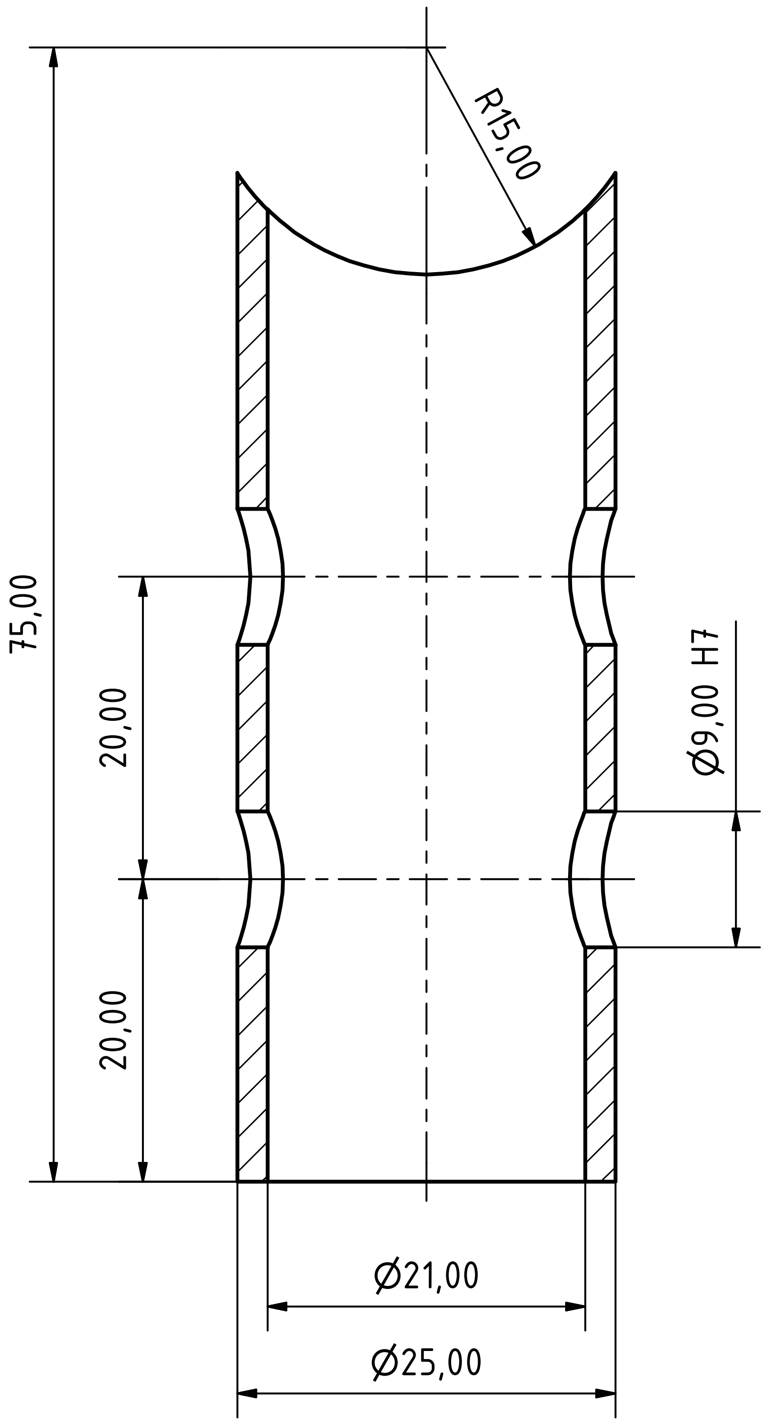
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 2	Project: Chassis EK1	Number: 7.10.A
Format: A4	Plan: DS Support U	Replaces:
		Replaced by:



University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 2 : 1	Project: Chassis EK1	Number: 7.11.A
Format: A4	Plan: DS Support Lt	Replaces:
		Replaced by:

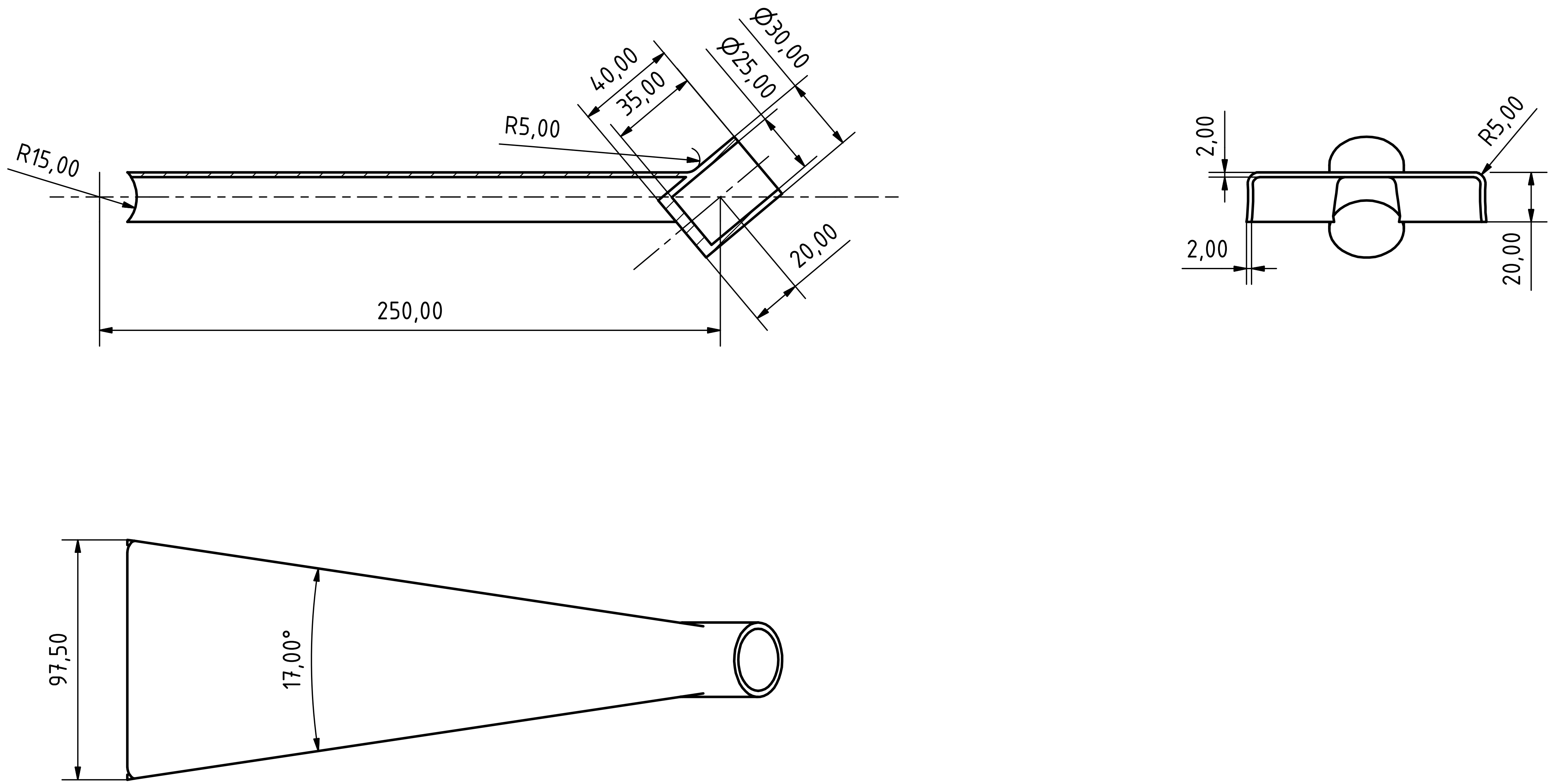


University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 1	Project: Chassis EK1	Number: 7.12.A
Format: A4	Plan: Upper Frontal Bumper Support L	Replaces:
		Replaced by:

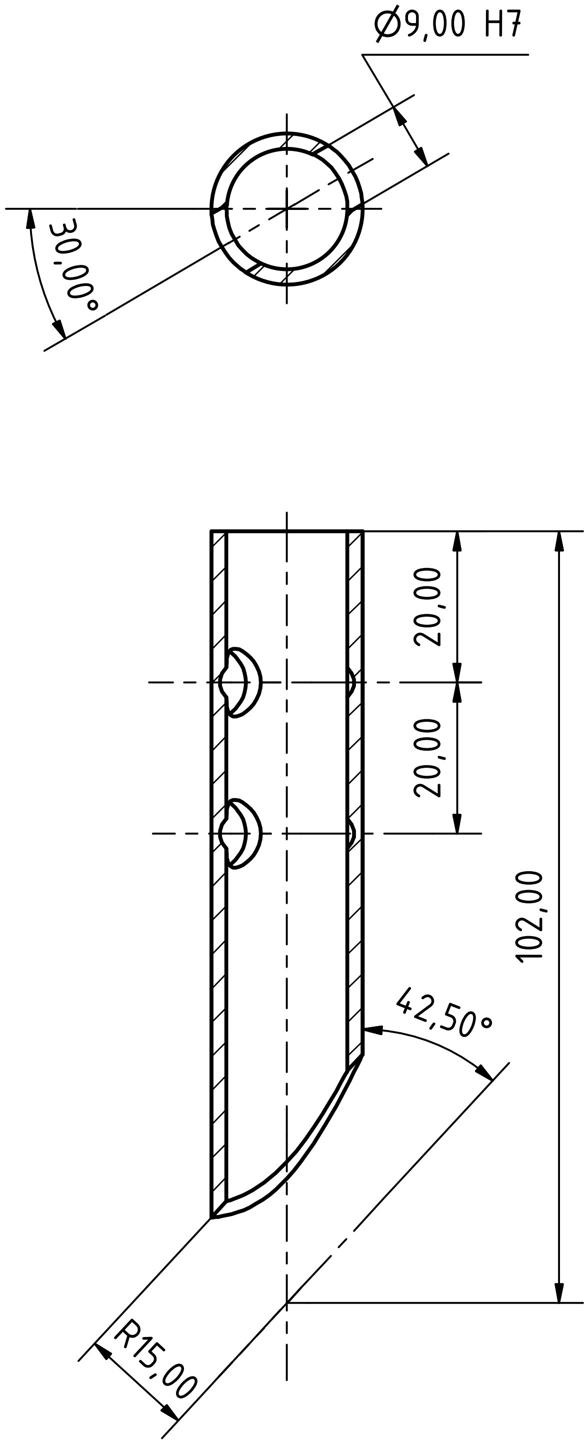


University:	Name:	Date:
ETSID - UPV	Adrián Altamira Peña	29/06/2019

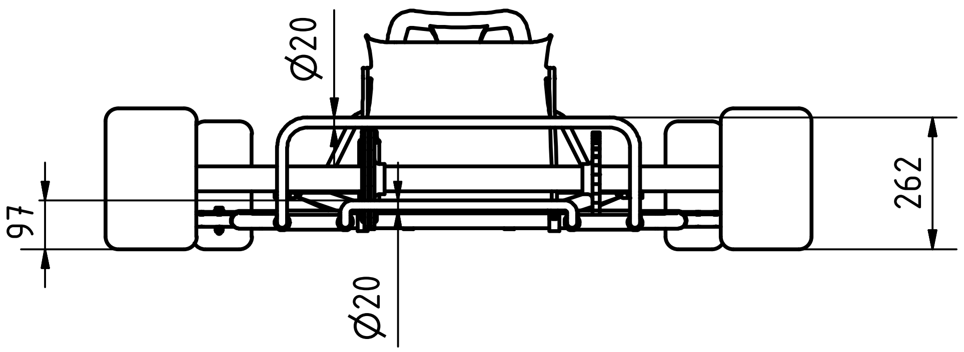
Scale: 2 : 1	Project: Chassis EK1	Number: 7.13.A
Format: A4	Plan: Lower Frontal Bumper Support	Replaces:
		Replaced by:

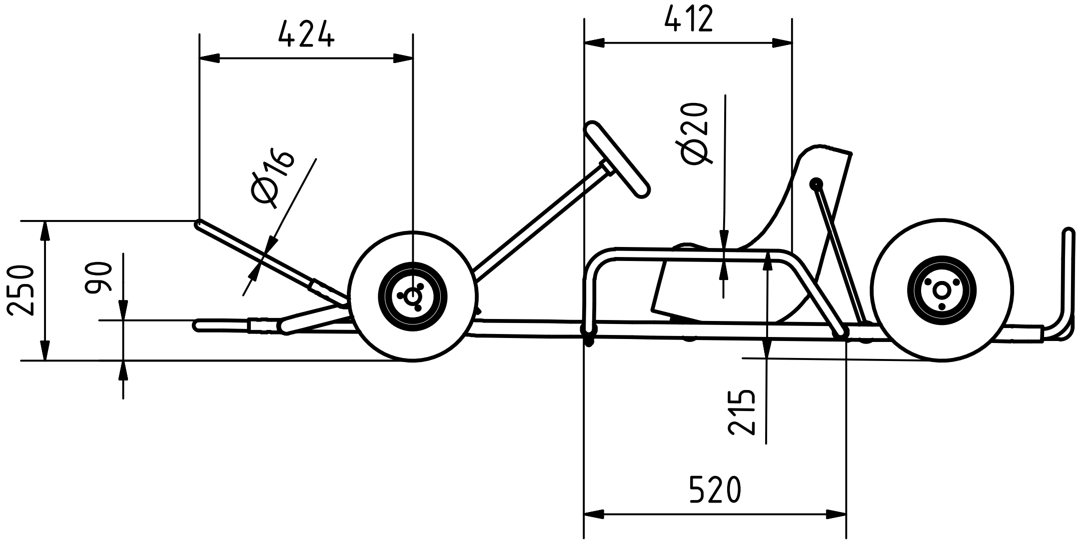


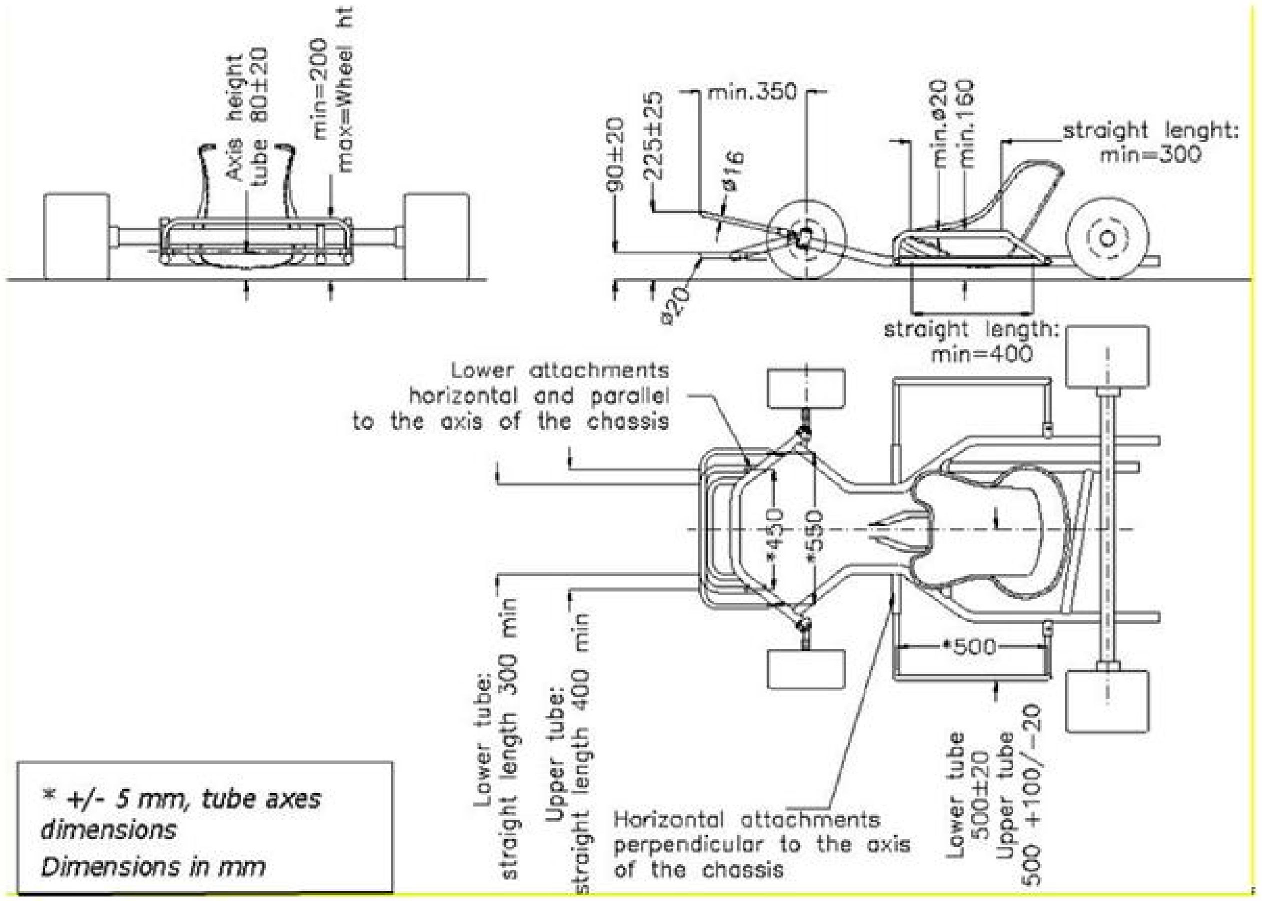
University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 2	Project: Chassis EK1	Number: 7.14.A
Format: A3	Plan: DS Support Lw	Replaces:
		Replaced by:

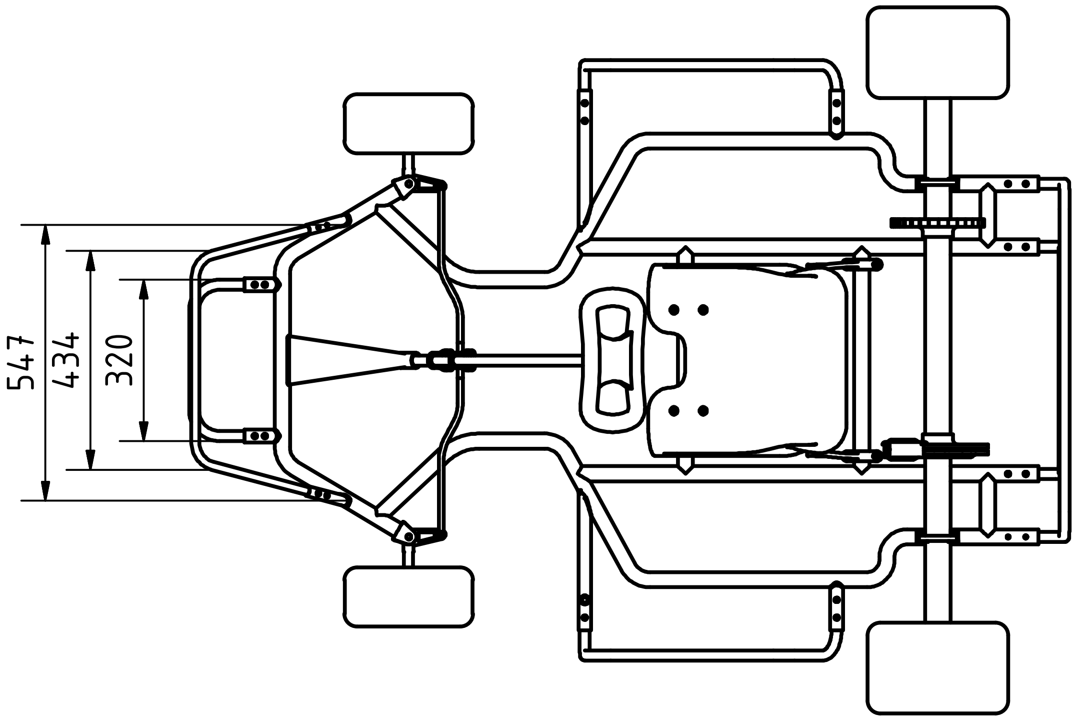


University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1 : 1	Project: Chassis EK1	Number: 7.15.A
Format: A4	Plan: Upper Frontal Bumper Support R	Replaces:
		Replaced by:









University: ETSID - UPV	Name: Adrián Altamira Peña	Date: 29/06/2019
Scale: 1:15	Project: Chassis EK1	Number: E.2.00.A
Format: A3	Plan: General bumpers dimensions	Replaces:
		Replaced by:

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Design and analysis of the chassis of an electric kart

3. Product conditions

3.1. Objective

The objective of this project was to design a chassis for an electric kart that an amateur driver or a company that rent karts in a circuit could use to make them karts.

This document gives some advises and conditions of use in order to guarantee the proper functioning of the kart.

3.2. Legal conditions and regulations

To do this design; the chassis and the bumpers, it is used the "Technical regulations for electric Karts (e-Karting)" of 2019. To corroborate that its design is made following these regulations it could be check that the dimensions of the bumpers follow the dimensions of the technical drawing Nº 2a. Check the draw E.2.00.A.

Other important point was the material; and in order to follow the ISO 4948, and ISO 4949 the normalized AISI 4130 was chosen.

3.3. Materials and components requirements

The tubes have to follow the dimensions indicated in the plans; but if for any the material have to be changed; it could be accepted any material that complies the ISO 4948, and ISO 4949.

The certificate ASTM A500 grade A is going to be required to the fabricant. It is an habitual certificate required in structural steel tubes. Overall it is used in edification; but like the steel in this application is structural steel, this certificate is going to be required.

3.4. Conditions of fabrication

3.4.1. Tubes cutting

This process is going to be done to ambient temperature, because a good surface finish is required in the surfaces that are going to be melted. The surfaces that are going to be machining have to be smooth.

3.4.2. Tube bending

This process is going to be made without heating the tubes; with a machine that let bending the tubes using 3 rollers. In this process the critical point is the occurrence of fissures and cracks. So, it is necessary to do a statistic check to guarantee the no occurrence of fissures and the thickness.

3.4.3. Weld process

The welds have to be done with weld seam on 1,5 mm; and with a material with the same characteristics, or better, than the steel of the tubes. This process has to be done following the EN-ISO 3834 and EN-ISO 9001.

3.4.4. Assembly

After cut, make the holes, and bend all the tubes; the different components of the chassis are going the be weld; starting for the frontal and lateral tubes; following with the central tubes; and ends with the supports.

After weld all the tubes, the bumpers could be screwed, like in the 1.00.A drawing.

This last pass could be before or after assembly the other components of the chassis; it is recommendable to do after, in order to assembly the other components easier.

3.5. Quality controls

The critical point of this structure could be the weld points; overall in the union between the frontal and lateral tubes; so, in order to guarantee that them are well a control using X-ray is going to be done in all the weld points.

This control is going to be done following these steps:

1. Visual control: after the 48 h of the weld process a visual control of this points is going to be done in order to look for imperfections or irregularities.

2. Penetrating liquids: if the previous steep cannot guarantee the quality of the process this analysis is going to be done.

3. X-ray: if the structure passes the first or the second control a third one is going to be done; and is an industrial X-ray.

3.6. Tests

3.6.1. Static analysis

In all the chassis a static analysis is going to be done applying a load of 2 kN in the tube 1; and checking that the vertical deformation is lower than 5 mm.

3.6.2. A recommendable test

Well, this test cannot be made sins all the kart is assembled; but it is highly recommendable. It is just a performance test in order to guarantee that the kart not suffer a problem of resonance. If the kart suffers this problem, there are some solutions:

1. Changing the tyres: this is the best solution because not change the characteristics of the kart; but it could solve the problem.

2. Changing the batteries: it's the most expensive solution; but if the mass of the batteries is reduced, the maximum speed, the maximum velocity, and the autonomy are going to be increased.

3. Introduce an additional load: this could be a solution if the mass of the kart is lower than the maximum one; but the maximum acceleration, speed and autonomy is going to decrease.

3.7. Delivery terms

The chassis have to be delivered in one carton box; in this box there have to be the chassis with porexpan protections; the bumpers in other boxes inside the first one; and with porexpan protections too; and a plastic bag with the bolds, and nut divided in three bags.

3.8. Use conditions

To use this chassis the design of the kart has to follow the technical regulations of the FIA.

The maximum mass of the pilot (protections included) is of 100 kg.

There is not limit in the power of the engine; like it can be check in the point 1.8.7. of the annex of calculus; concretely in the art B2, as higher is the acceleration lower is the deformation; so, there is not limitation in the power of the engine.

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Design and analysis of the chassis of an electric kart

4. Budged

4.1. Materials

Into the following table there is a breakdown of the price of any component of the chassis.

Ref.	Tube	Ø (mm)	t (mm)	m (kg)	Unities	Price (€/kg)	Price (€)
T.1.	Frontal	30	2	1,29	1		
T.2.	Lateral (L/R)	30	2	2,16	2		
T.3.	Reinforcement	30	2	0,14	2		
T.4.	Central H	30	2	0,59	2		
T.5.	Central V	30	2	1,24	2		
			Total (+5%):	10,03	Total:	0,9	9,02
T.6.	Lateral bumper support 1	25	2	0,29	2		
T.7.	Lateral bumper support 2	25	2	0,1	2		
T.8.	Upper frontal bumper support (L/R)	25	2	0,07	2		
T.9.	Lower frontal bumper support	25	2	0,07	2		
			Total (+5%):	1,11	Total:	0,9	1,00
T.10.	DS support U	20	2	0,23	2		
T.11.	Lower frontal bumper	20	2	0,51	1		
T.12.	Lateral bumper (L/R)	20	2	0,8	2		
T.13.	Upper back bumper	20	2	1,11	1		
T.14.	Lower back bumper	20	2	0,65	1		
			Total (+5%):	4,55	Total:	0,9	4,09
T.15.	Upper frontal bumper	18	2	0,8	1		
			Total (+5%):	0,8	Total:	0,9	0,72
	Other components				**Unities**	**Price (€/U.)**	**Price (€)**
C.1.	Shaft support				2	0,20	0,40
C.2.	DS support Lt				2	0,15	0,30
C.3.	DS Support Lw				1	0,50	0,50
N.1.	EN 1665 M8x35				24	0,021	0,50
N.2.	DIN 126-9				24	0,005	0,12
N.3.	ISO 4035 - M8				24	0,011	0,26
N.4.	DIN 986 - M8				24	0,016	0,38
					Total:	**17,31**	

Table 26: Materials costs

4.2. Manpower and engineering costs

Into the following table there are the costs of the manpower (for a kart) and of engineering. The costs of engineering are divided for 500 chassis; because this work it is only done one time, it is not done for every kart. So, it is considered that a 500 chassis are going to be produced, at least.

Ref.	Manpower	Time (h)	Price (€/h)	Price (€)
M.1.	Tube cutting	2	20	40,00
M.2.	Tube bending	1	20	20,00
M.3.	Weld	5	50	250,00
M.4.	Assembly	1	15	15,00
			Total:	325,00

Table 27: Manpower costs

Ref.	Engineering	Time (h)	Price (€/h)	Price (€)
E.1.	Direction	0,02	50	1,00
E.2.	Desing	0,1	50	5,00
E.3.	Analyse	0,1	50	5,00
			Total:	11,00

Table 28: Engineering costs

4.3. Machinery

Into the following table there are the costs of the use of the machinery to make a kart:

Ref.	Machinery	Time (h)	Price (€/h)	Price (€)
Mch.1.	Machines	3	5	15,00
Mch.2.	Tube bending machine	1	5	5,00
Mch.3.	Weld equip	5	10	50,00
			Total:	70,00

Table 29: Machinery use costs

4.4. Amortization

In the following table there are the costs of the factory and buy the machines. These costs are calculated for 1 month; and divided between the expected production; 40 karts/month. 2 per every workday.

Ref.	Amortization	Time (months)	Price (€/month)	Price (€)
A.1.	Factory	1	3000	75,00
A.2.	Furniture	1	100	2,50
A.3.	Machinery	1	500	10,00
			Total:	87,50

Table 30: Amortization

Other of the costs is the software used. Like the design was done in 3 months this cost is only for this period. In that case, this cost by 500 other time; due this is the minimum production expected.

Ref.	Software	Time (months)	Price (€/year)	Price (€)
S.1.	Inventor	3	535	0,27
S.2.	SAP 2000	3	8000	4,00
S.3.	Excel	3	80	0,04
			Total:	4,31

Table 31: Software amortization

4.5. Final budged.

Once that al the prices are calculated the final budged could be calculated; adding all the expenses and applying the industrial earning rate (6%) and taxes (21% VAT)

Concept	Price (€)
Materials	17,31
Manpower & Engineering	336,00
Use of machinery	70,00
Amortization	87,50
Software	4,31
Total:	515,12
Industrial earning (6%)	30,91
Total:	546,02
Tax (VAT 21%)	114,67
Total:	**660,69**

Table 32: Final budged

The final budged is lower than the initial one (that was a prevision); so that prices could be adjusted:

Component	Price (€)
Chassis	410,00
Frontal Bumpers (2)	80,00
Side Bumpers (2)	85,00
Back Bumpers (2)	90,00
Total:	**665,00**

Table 33: Final components prices

A. References

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B. Tables

C. Figures