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DESIGN OF A TUBULAR CHASSIS FOR A KARTCROSS

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ABSTRACT

The objective of this project is to establish a method of calculation for designing a tubular chassis of a kart-cross in a manner in which can be measured all the parameters that are related to the design to get a chassis as light, strong and economical as possible, taking into account that the time available is limited.

It is first necessary to know when you can give as valid a chassis in their resistance. This is essential to know the criteria used by tubular chassis designers in different competitions such as Formula SAE/Student Car, Kart-Cross or NASCAR.

Once you are aware of the limits that must not be exceeded we must decide how it will be done the calculation. The theory of finite elements would be ideal if the geometry was easy to obtain.

For the purposes of these theories is necessary to make a rough drawing of the chassis. To carry out this approach correctly, it is essential to keep in mind the methods of manufacture which provides a worker when constructing a tubular structure such as this.

The usefulness of the project is evident. It is absolutely necessary to know how carry out the calculation of a chassis to design a tough vehicle, that, behave properly and be as light as possible.

One of its most immediate applications is to assist in the design of a competition car Formula SAE / Student and Kartcross racing, which are used chassis tubular.

Nowadays, in the world of automobile competition, the solution for the construction of a chassis could be: the cheapest one, the lightest one or the most resistant one. A lot of money is being spent on the construction of a vehicle, big teams don't care about the money, and they care about the weight and strength of the vehicle. In this project the desired final solution would be a combination of the 3 previous solutions.

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1. INTRODUCTION

Increasingly, many people who are fond of the motor world want to put in practice their driving skills without endangering their life or people's life on the roads.

A kart is an ideal way to enjoy motor sport, with a safe and small investment in relation to other forms of motorsport.

These are the reasons why I decided to make this project. The design of a chassis could be a great idea if you love motorsport world.

The history of the sport says that it was born in 1951 in an aviation base in the United States of America. In the 60's the sport was introduced into Europe through France and England

The first kart was made with heating tubes a lawn mower engine and the steering wheel of an old plane into disuse.

Early in the history of karting, karts were some basic equipment that barely reached 50 km/h. Since the beginning of the history of the evolution of the karting it has been steadily increasing. The chassis has been gaining in stability and braking. Today a kart can reach speeds exceeding 150 km/h but with the confidence of a Formula 1 car.

In 50 years karting has gained the respect and recognition as a sport and a wonderful school of driving. Men like Senna, Prost, Schumacher, Alesi, Hakkinen, Coulthard, Hamilton or Alonso and many others collaborated at the beginning on the sport of karting to be the most important basis of current motorsport.

The objective of this project is to establish a method of calculation for designing a tubular chassis of a kart-cross in a manner in which can be measured all the parameters that are related to the design to get a chassis as light, strong and economical as possible, taking into account that the time available is limited.

Nowadays, in the world of automobile competition, the solution for the construction of a chassis could be: the cheapest one, the lightest one or the most resistant one. A lot of money is being spent on the construction of a vehicle, big teams don't care about the

money, and they care about the weight and strength of the vehicle. In this project the desired final solution would be a combination of the 3 previous solutions.

It is first necessary to know when you can give as valid a chassis in their resistance. This is essential to know the criteria used by tubular chassis designers in different competitions such as Formula SAE/Student Car, Kart-Cross or NASCAR.

1.1. WHAT IS A KART?

According to the International Karting Regulation a kart is a land vehicle with or without bodywork; it has four wheels not aligned in constant contact with the road. Two of the wheels provide the traction and the other provides the direction.

The kartcross are agile, fast and powerful experience large accelerations due to their low 300kg to 100 horsepower developed by the powerful, lightweight motorcycle engines

Basically the operation is very simple from the beginning of the history of karting. The engine transmits movement to the rear axle by a chain. At one end of the crankshaft is located a pinion gear teeth on the chain that sits. When the engine moves it moves the chain which movement occurs on the rear axle, the two rear wheels always rotate in unison, and obviously makes the movement of the kart.

2. AIMS OF THE PROJECT

The objective of this project is to establish a method of calculation for designing a tubular chassis of a kart-cross in a manner in which can be measured all the parameters that are related to the design to get a chassis as light, strong and economical as possible, taking into account that the time available is limited.

Nowadays, in the world of automobile competition, the solution for the construction of a chassis could be: the cheapest one, the lightest one or the most resistant one. A lot of money is being spent on the construction of a vehicle, big teams don't care about the money, and they care about the weight and strength of the vehicle. In this project the desired final solution would be a combination of the 3 previous solutions.

The objectives in this project are:

1. Static calculation: Will be held the load distribution when the kart is stationary. The total weight of the chassis is equal to the sum of reactions at the supports, in our case, the axles.

2. Dynamic calculation: Will be held the load distribution when the kart accelerates, when the kart brakes and when the kart is cornering. When braking or boot a vehicle appears an inertial force that opposes the force that tends to set in motion or stop the vehicle and modifying the loads on axles.

When starting a vehicle the inertial force is higher on the rear axle than when the car is stationary. The opposite happens on the front axle, because the weight is transferred from the front axle to the rear axle.

For braking, the inertial force is higher on the front axle than when the car is stationary. The opposite happens on the rear axle, because when the kart brake, the weight is transferred from the rear axle to the front axle.

Finally, when the kart is cornering there is also a load transfer. This aspect will be studied later. When the kart is cornering, it is changing its trajectory, so it is changing also the loads that the kart is subject.

3. Torsion and flexion. Then it will be studied how the chassis behaves in flexion and torsion analysis. With the software Ansys, one end of the structure will be fixed, and on the other end will be applied a charge and a torque.

With these studies it will be known which the parts of the chassis that suffer more are.

After that, the material chosen has to be analyzed, and, if is there other material with better properties, this new material will be chosen to do the calculations again.

3. DESIGN OF A TUBULAR CHASSIS

One of the most important parts of the kart is the chassis. A chassis can be defined as a structure whose purpose is to rigidly connect the front suspension and rear grip and provide points for the various vehicle systems, as well as protect the driver against the collision. The chassis should be rigid to deform slightly and so does not alter the characteristics of driving. It is analogous to an animal's skeleton. It is the most crucial element that gives strength and stability to the vehicle under different conditions.

Chassis is used mainly for cargo vehicles such as vans, trucks and buses, and also as reinforcement in the race cars such as Formula SAE / Student kartcross and NASCAR. It is known as a metal frame chassis, which are mounted on all components of the vehicle. The body is installed on the chassis once your application has been determined (in our case, kartcross). The installation proceeds bolting or welding the car body to the frame, if bolting is known as "coachwork independent" if welded known as self-supporting.

In competition vehicles it could be said that the two major types are monocoque and tubular frame.

In monocoque chassis the difference between the chassis and the body is diffuse, because the chassis is part of the bodywork.

Tubular chassis are the most commonly used as reinforcement of competition vehicles because their design is simpler and the determination of the stresses to which they can be subjected.

Regarding the mechanical behavior it must be said that the chassis is more rigid than the bodywork. It is interesting at the time of a crash, that the bodywork is deformed everything it can to not transmit the energy of the collision on passengers but the chassis should be deformed slightly so as not to alter the driving characteristics

The functions realized by the chassis are:

- It is the main support element of the vehicle: as it was said before, the chassis is analogous to an animal's skeleton; it is the element that supports all of the stress in the vehicle.
- It is the rigid connection between the rear axle and the front axle. In a vehicle, the distance between the front and the rear axle, should be constant.
- It gives to the kart the necessary rigidity for the possibilities that can occur when the forces are in motion.
- It protects the pilot against the collision. Is designed so that if an accident occurs, the frame will not break or deform very much and the pilot will have the less damage.

The chassis construction is the compromise between stiffness, weight and space, taking into account all the final cost. It should be considered static and fatigue resistance, stability of structural members, the load bearing capacity of the unions, manufacturing and assembly. In this project only take into account the efforts that may be considered static. For a complete measure of a chassis it should also be fatigue calculations and collision, which gives for a couple of other projects.

3.1. STIFFNESS CRITERIA

The overall expression of the stiffness is: $K = P / \Delta$

Where: P: applied load

Δ : deformation

The stiffness supplies the followings proportionalities: $K \propto E \times I$ and $K \propto E \times A$

Where: E: modulus of elasticity or Young's modulus

I: moment of inertia

A: sectional area

Of these proportionalities follow that the higher modulus of elasticity, moment of inertia and / or sectional area, the greater the stiffness.

In the stiffness of a chassis, it takes into account two aspects: Bending stiffness and torsion rigidity.

Bending stiffness: Refers to how much the chassis bends due to the weight of the different elements of the vehicle. Experience tells us that it is not really a problem in the chassis design.

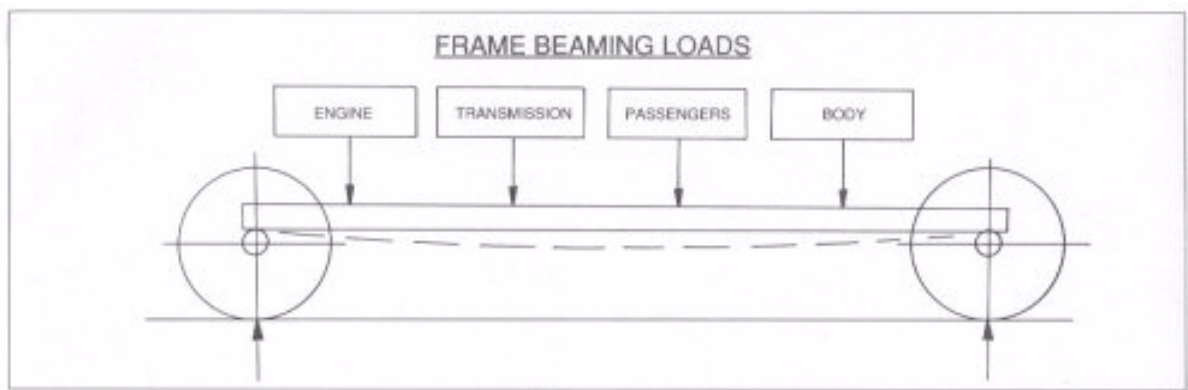


Fig. 3.1: Loads that can bend the chassis

Torsion stiffness: This refers to how much a chassis deforms due to an asymmetric load, for example, when a front wheel goes over a bump while the other do not. This is the characteristic that should take care to validate in terms of chassis rigidity. According to the competition which is aimed to design the car is available for torsion rigidity or another. This depends on the maximum torque that can be subjected. This torque comes from the combined forces of the shock absorber. We should have to decide how many degrees we want it to be deformed when we applied the maximum torque, for example, one degree, an amount invisible to the human eye. Finally, the stiffness should be maximized to provide a safety margin and to obtain a round number. In a competition of Kartcross, a torsion rigidity of $150 \text{ kg} \cdot \text{m} / ^\circ$ is an acceptable amount. According to sources, in a race of Formula SAE/Student a reasonable torsion rigidity would be $500 \text{ kg} \cdot \text{m} / ^\circ$. According to other sources, for a NASCAR race would require $1500 \text{ kg} \cdot \text{m} / ^\circ$ of torsion rigidity.

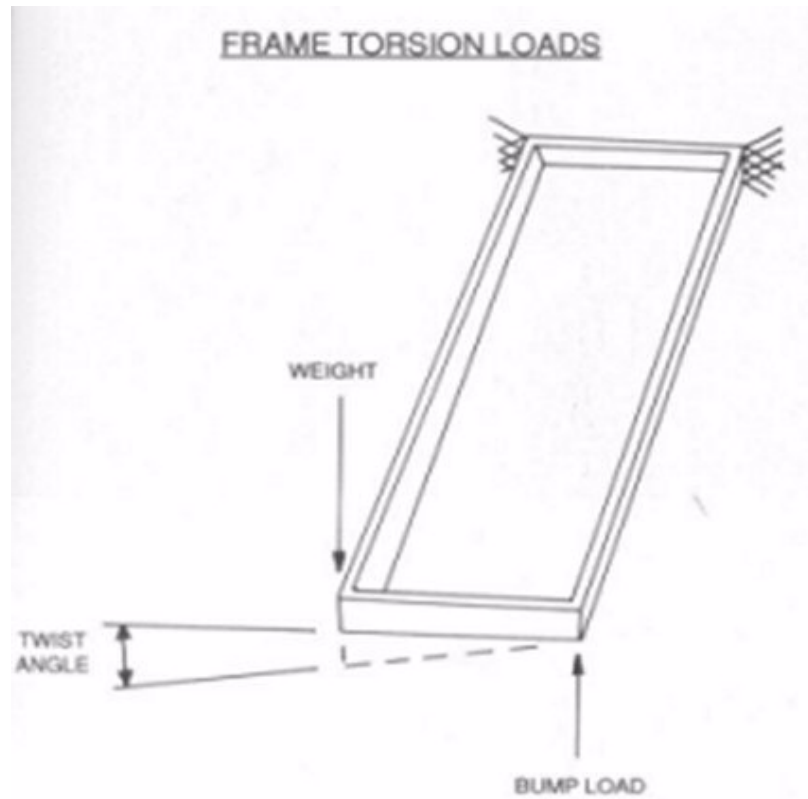


Fig. 3.2: Efforts under chassis torsion

Triangulation: For example, we have a rectangular structure to which a load is applied as is shown in the figure below.

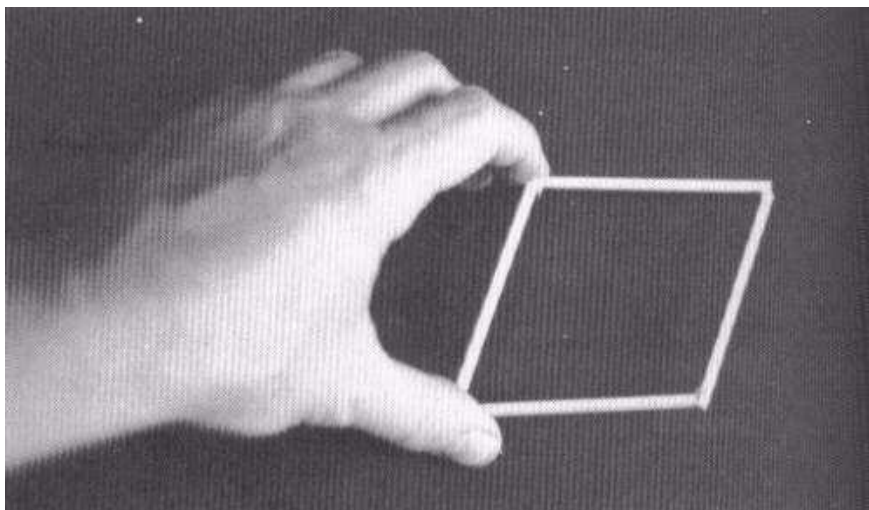


Fig. 3.3: Rectangular structure under torsion

It is found that the torsion stiffness is not much since the nodes must absorb much of the efforts in the form of bending moment. If you put a bar as shown in the figure below, the bar is working on axial stress (tension or compression) so that the node is suffering a smaller bending moment. Several studies show that the deformation due to axial stress is much lower, by orders of magnitude, due to bending moments and torsion. Therefore it is preferable that the bars are made to work to axial stress rather than bending moment or torque. This is achieved with triangulated structures. As for the type of axial stress, it is preferable traction to compression to avoid problems of buckling.

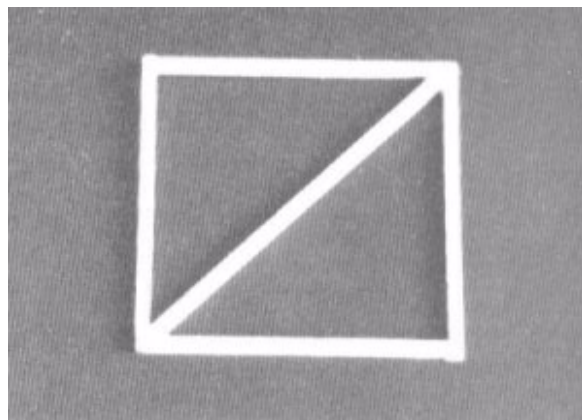


Fig. 3.4: Triangulated structure

In the design of a chassis, in terms of stiffness it must be taken into account the following points:

- There are elements that are not part of the structure but also provide rigidity, often not insignificant, for example, the engine. It must be taken into account when calculating.
- By decreasing the elastic modulus E because, for example, by choosing titanium or aluminum instead of steel, to not lower the overall stiffness the moment of inertia I must be increased and also the area A must be increased by increasing the diameters of the tubes.
- The elements that produce high load such as motor and suspension should be tied into the chassis triangulated points.
- Driving controls should be set as best as possible so the chassis will not deform while driving.

- The bars with a higher distance between supports need a greatest moment of inertia for increase rigidity.
- To increase the torsion stiffness could be added to the basic structure a side pod giving a greater moment of inertia. These side pods also increase the Lateral Impact Protection.

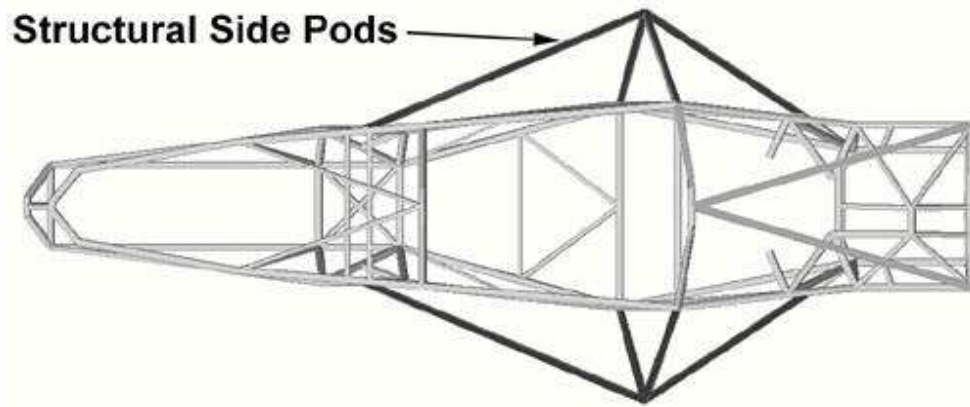


Fig.3.5: Structural side pods

- The proper subjection of motor components is very important for a long life of the chassis.
- The seat belt brackets should not deform significantly during the crash.
- Although in a crash the bodywork should be deformed as much as possible, the part that protects the driver's feet should be rigid.

3.2. CRITERIA ON THE WEIGHT AND ITS DISTRIBUTION

In the design of a chassis, in terms of weight and weight distribution must be considered the following points:

- The less the chassis weights, respecting the rigidity, the best engine power the kart will have.
- Regarding the studies done for the suspension, the center of gravity should be as low as possible to reduce rolling.

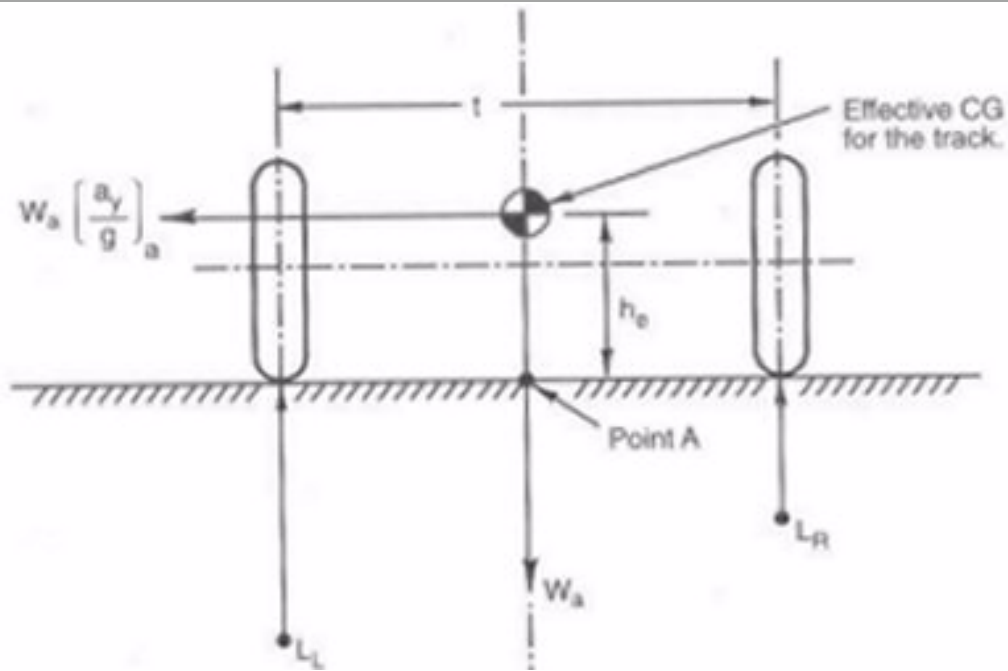


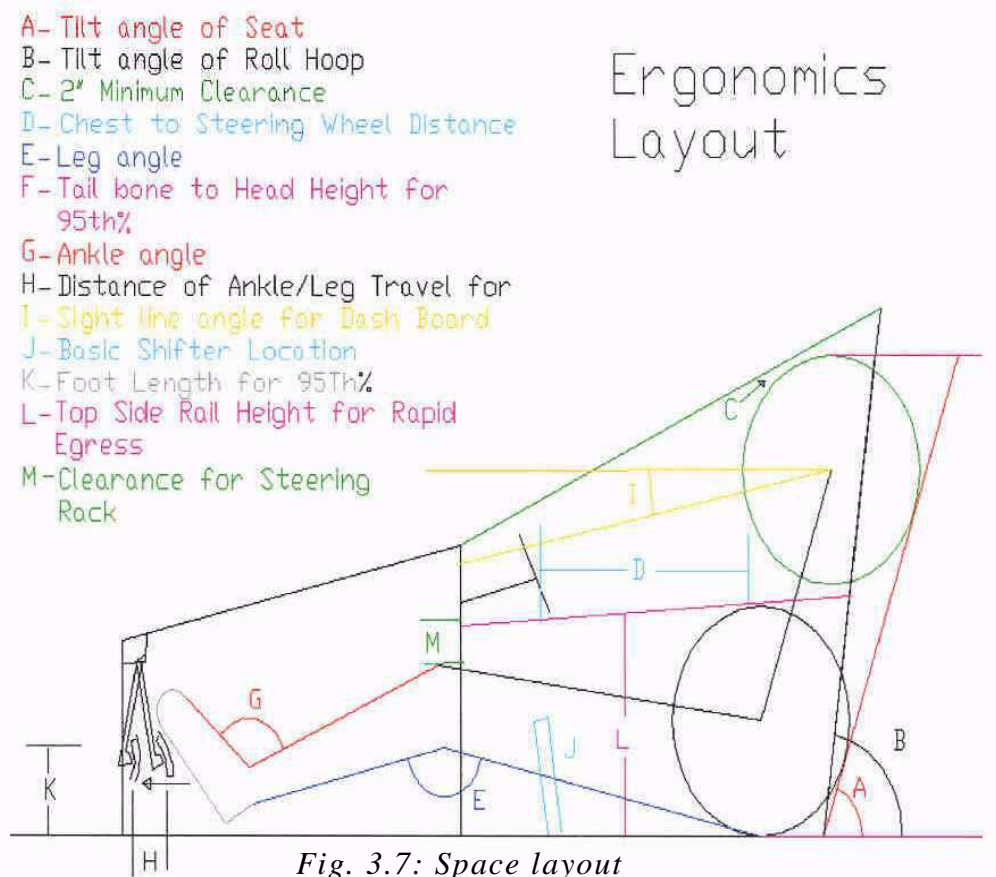
Fig. 3.6: Weight distribution

- Regarding the studies done for the aerodynamics, the center of gravity should be forward of the center of lateral pressure to avoid driving instabilities due to sudden changes in a crosswind, for example, overtaking a truck. It is known that the center of lateral pressure is further back if the side surface is greater in the rear than up front.

3.3. SPACE CRITERIA

In the design of a chassis, in terms of space needs should take into account the following points:

- In the design of the structure, if there is transmission by chain, there should be enough space to mount a sprocket size range acceptable.
- It should be considered the ease of access for maintenance of propulsion elements.
- The 95 percentile man should be able to comfortably enter the car with the helmet. 95% percentile means that 95% of men are smaller than this model and that only 5% are bigger. The structure should not interfere with the driver in the movement for it to perform driving. A particular problem in this is the arms of driver.



- Measures of the pedals, the length and angle of the feet determine the height of the front of the chassis.
- The angle of the legs and body dimensions determine the length the seat.
- The sight line is used to determine the height of front rim.
- It is very important a rapid evacuation of the driver if there is an accident. For example, competition in Formula SAE / Student must be less than five seconds.



Fig.3.8. Situation of the driver and the engine

3.4. COST CRITERIA

In the design of a chassis, should be taken into account the following aspects to decrease the cost:

- The selection of bars should be the less varied possible in diameters.
- The number of bent bars should be as small as possible.
- The number of joints should be the minimum.
- In a welded construction of a tubular structure, almost all costs manufacturing bars correspond to the filling bars. It has been shown that with K-type joints could be obtained the minimum number of bars and filling joints. The spaced knots are easier to manufacture, because they simply apply a single cut on each end of the filling bar.
- The number of welds should be the minimum. This is achieved asking tubular profiles in extra long lengths.

3.5. APPLIED LOADS

The principal Efforts applied into a chassis are the bending and torsion. The bending is not as important as the torque because the bending does not affect the loads of the wheels, which are most affecting in the chassis. The car is also subject to stresses due to aerodynamics. The chassis must be formed that the air push the car down and the chassis should be rigid to deform. In a Kartcross tubular chassis type is not really important down force because the speeds are not large enough to have influence.

- The design efforts are the worst conditions:
 - Kart at maximum cornering speeds.
 - Hard acceleration.
 - Braking sharply in both, straight and curved.
- The points of application of the efforts include:
 - Brackets of the suspension (suspension forces).

- Brackets where heavy loads are applied (and weight forces inertia).
- The structure itself (weight and inertia forces)

The loads mentioned can be classified by their variation in time, as follows:

- Permanent loads G, for example, the weight of the structure, the weight of fixed equipment and driver.
- Varying loads Q, erg loads from the suspension or the inertia during acceleration, braking or cornering

The variable loads are considered as quasi-static. It begins with the 'characteristic value' of the load, which is the average value of the load in a space of time. For example, is assumed that the kart is cornering. While it is cornering, the lateral acceleration will probably change as it will do the curve, because surely the driver will change the speed or the radius of curvature. This lateral acceleration produces an inertia loads that change in the same proportion as does the acceleration. It would be taken as the characteristic value of the load of inertia the average during the maneuver. It should be taken into account the variability of the acceleration with time during the maneuver and not only the average arithmetic between the maximum and minimum value. Generally the characteristic value is called F_k .

The characteristic value is multiplied by a partial safety coefficient γ_k adopted for the loading considered. This partial safety coefficient takes into account possible adverse deviation of the magnitude of the loads, an inaccurate modeling of the same or some uncertainty in the evaluation of the effects of the loads or limit state considered.

Thus, we have what is called 'load's calculus value':

$$F_d = \gamma_k \times F_k$$

In theory, there are two ways to determine the numerical values of partial coefficients:

- 1- For calibration of a long history and tradition constructive. This is the basic principle of most of the coefficients proposed in the current Eurocode.

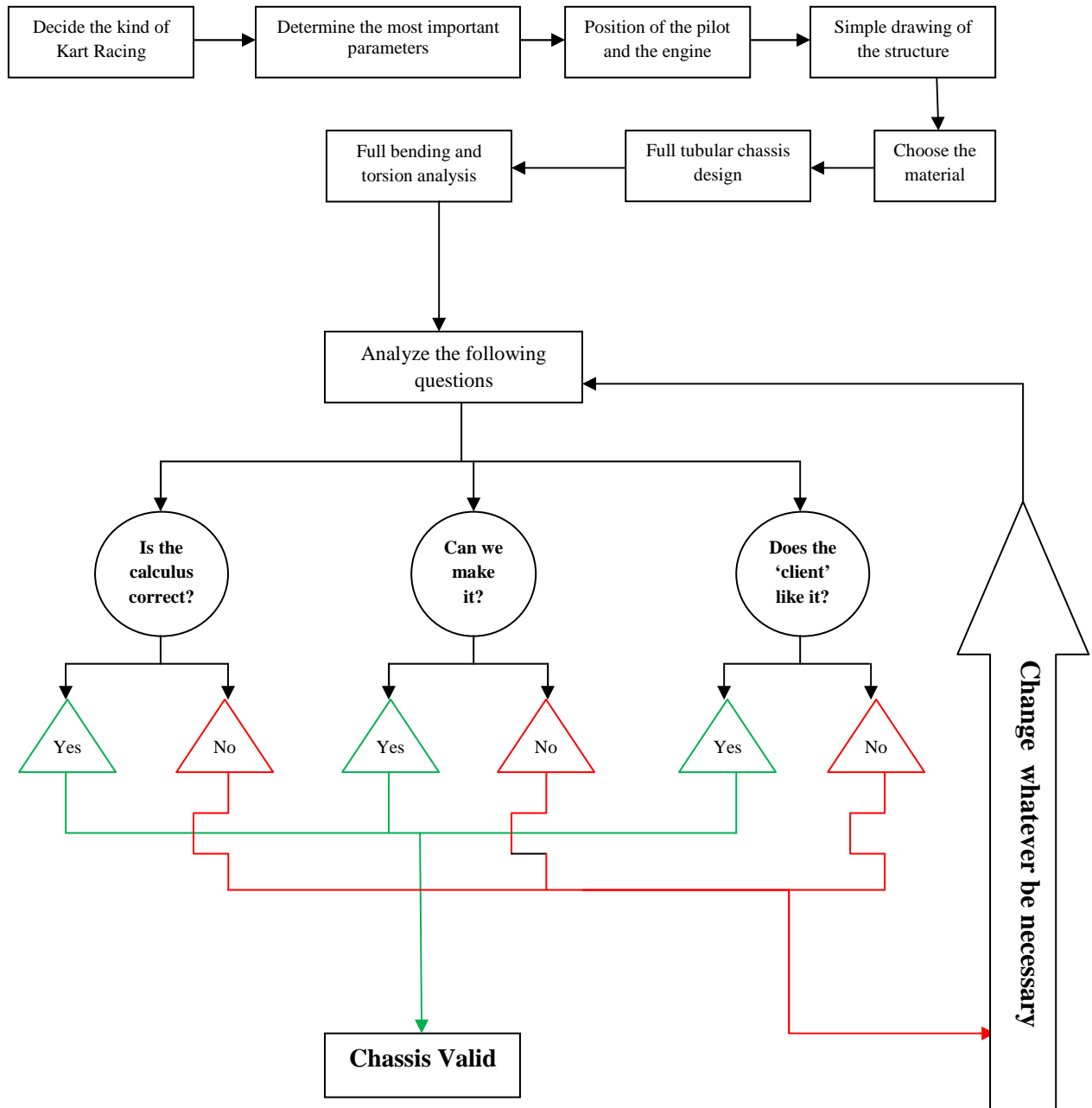
- 2- By the statistical evaluation of experimental data and field observations, and should be performed within the probabilistic theory of reliability.

Actually, to refine the calculated profiles, must be taken into account more factors, but usually, is taken the following values that are more conservative in order to simplify the calculation:

Permanent loads will have a coefficient $\gamma_G = 1.33$

Variable loads will have coefficient $\gamma_Q = 1.15$

3.6. DESIGN ALGORITHM



4. PROJECT TIMETABLE

Activity/week	October				November				December				January				February				March				April			
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
Decide kind of kart																												
Legislation																												
Parameters																												
First drawing																												
Interim Report																												
Full drawing																												
Analysis																												
Final Report																												
Revision																												

Fig. 4.1: Project Timetable

5. MATERIALS

5.1. INTRODUCTION

Not infrequently in designing structures, the designer makes the mistake of ignoring the material that most suits from the beginning, and its possibilities to be modeling. The engineer must know the tools available to tinker, its costs and experience. He must know whether the material is made up in cold or hot, the behavior of welded joints varies. Also it is very important to know which are the mechanical properties of materials such as its elasticity modulus E , the shear modulus G , density ρ and its yield stress f_y . In theory, the section of the tubes that are available can be circular or rectangular, and may be hollow or solid.

5.2. TYPES OF STEEL

The first question is, why steel? The chassis could also be made of titanium, aluminum or carbon fiber. The truth is that chassis can be made of almost any material but it must also be taken into account cost, mechanical behavior and formability possibilities it has. The steel has the following advantages:

- Its price is relatively cheap
- Good weld ability
- Ductile material
- Its elasticity modulus is higher than many other materials such as titanium or aluminum. Thus the size of the section of the tube needed to have the same stiffness is lower.

As is mentioned in the introduction, the designer always needs to specify whether the material is cold-finished or warm-finished. The cold-finished tubular profiles are always welded, and the warm-finished tubular profiles, although most of them are also welded, may not have seam. In the case of the construction of a tubular chassis the most usual is to use cold-finished tubular profiles.

The types of steel are specified by the International Organization for Standardization (ISO) in the following rules:

- **ISO 630**: Structural steels.
- **ISO 4951**: High yield strength steel bars and sections.
- **ISO 4952**: Structural steels with improved atmospheric corrosion resistance.

The chemical composition and mechanical properties of cold-finished tubular profiles comply with the recommendations of **ISO 630**. The mechanical properties of steels are generally characterized by the yield strength f_y , tensile ultimate strength f_u , and elongation δ_u . These properties are determined by tensile tests and can achieve diagrams σ - ϵ . The regulation **ENV 1993-1-1** prescribes the following minimum value for the relationship between ultimate strength f_u and tensile yield strength f_y :

$$f_u/f_y = 1.2 \text{ (based on the nominal values of } f_u \text{ and } f_y\text{)}$$

A structure made of tubular profiles filled with predominantly static loads should, in principle, be designed so as to provide a ductile behavior. This means that if the bars are critical, they should ensure capacity of rotation, or if the joints or connections are critical, they should also ensure sufficient rotation capacity. The ductility is measured in the Charpy V- test, in which a small steel piece with standard dimensions and a standard V-cut is subjected to a shock load in an environment with a given temperature. Charpy value represents the minimum breaking energy of the pieces in the trial that can withstand when they are set out to a given temperature, expressed in Joules. The values of the steels standardized by the ISO and by the CEN obey the minimum requirement of 27 Joules prescribed by the Eurocode 3.

Other aspect in the description of the mechanical properties is defined by the resistance and the ductility of the tubular profiles when they are loaded in the thickness direction. If during the test appear a crack (a lamellar tearing), it may be avoided by using steel with low sulfur content or adding sulfur with other elements such as, for example, calcium.

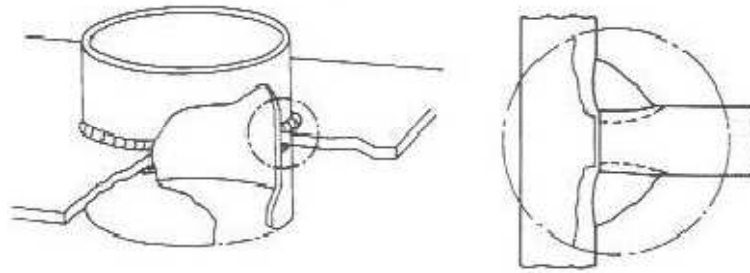


Fig.5.1: Lamellar tearing

The following types of steel correspond to the hot-finished tubular profiles and to the cold-finished tubular profiles. The designations of the types of steel in the table are fixed in the EN 10025, but they can be different in other regulations.

Type of Steel	Minimum elastic limit f_y (N/mm ²)	Tensile strength f_t (N/mm ²)	Minimum percentage elongation $L_0=5.65\sqrt{S_0}$	
			longitudinal	transverse
S235	235	340...470	26	24
S275	275	410...560	22	20
S335	335	490...630	22	20
S460	460	550...720	17	15

Table 5.1: Types of steel

In the first choice of material was used an alloy of carbon steel (0.25%) with a tensile strength of 490 MPa and a yield strength of 340 MPa. This option was quite expensive knowing that with other type of steel will be cheaper.

One of the aims of the project was the choice of the best material taking into account stiffness, cost, and weight. The lightest material would be one kind of carbon fiber, but the regulations do not allow the use of it.

It is recommended, taking into account the behavior of the joints, to chose the S335 steel with a elastic limit of $f_y = 335 \text{ N/mm}^2$. So de material chosen is S355 steel.

5.2.1. Physical properties of structural steels

Then, it is showed the recommended physical properties, valid for all structural steels.

Elastic modulus $E = 210,000 \text{ N/mm}^2 \text{ (MPa)}$

Shear modulus $G = \frac{E}{2(1+\nu)} = 81,000 \text{ N/mm}^2 \text{ (MPa)}$

Poisson ratio $\nu = 0.33$

Linear expansion coefficient $\alpha = 12 \times 10^{-6} /^\circ\text{C}$

Density $\rho = 7850 \text{ Kg/m}^3$

5.2.2. Types of bars for tubular chassis

First of all it must be answered this question, which bars do we prefer, solid or hollow bars? Is known that thin-walled tubes hold up well in flexion and buckling because the moment of inertia 'I' is higher than the moment of inertia in a solid tube of the same weight. In conclusion, the best bars to make a chassis are the hollow bars, in other words, tubes.

The following question is; circular section tubes (CHS) or rectangular section tubes (RHS)? The CHS are especially attractive and they offer an effective steel distribution around the center axis. This profile opposes the minimum resistance to wind and water loads. The problem is that when they have to join with others circular forms it may be required special profiling. Moreover, it is known that the geometric properties of the bars influence the capacity of resistance of the union. It can only be got the best design if the designer understands the behavior of the union and takes it into account from the conceptual design. Is known the properties of the joints between CHS and the joints between RHS, but is not known the properties of mixed unions. In the case of chassis, are preferred the CHS against RHS because of; esthetic, aerodynamic, axial bending and because the number of joints is not very high, so that is not decisive in the total cost.

5.2.3. Increase in the elastic limit caused by the cold deformation

This increase can be used only in RHS profiles in traction elements but not in bending. The best option is the CHS profiles so that increase does not matter in the elastic limit.

5.2.4. Consideration about the weld-ability of the materials

Basically, the chemical composition of a type of steel is which determines their weld-ability. For the weld-ability of unalloyed steels, generally used for the construction of a chassis, are critical carbon content ($C \leq 0.22 \%$) and purity of the steel indicated by the sulfur content ($S \leq 0.045 \%$), phosphorus ($P \leq 0.045\%$) and nitrogen ($N_2 \leq 0.009 \%$). The weld-ability improvement, not only because of the low percentage of carbon ($C \leq 0.20 \%$), but also because of the fine-grained microstructure of the material, which reduces the susceptibility to brittle fracture. The chemical composition, which influences the susceptibility to breakage in cold of the zone affected by heat, is often measured by the Carbon Equivalent Value CEV as follows:

6. CALCULATIONS OF A TUBULAR CHASSIS

6.1 INTRODUCTION

The ideal calculation would have taken place if the chassis could be accurately modeled, with the thickness of the each bar and with the joints well drawn. In this way, the chassis could be calculated by finite element method. The fact is that the means at our disposal at present do not allow us to do it completely. For this reason, to carry out the calculation, the design of the structure has to be approximate, based on manufacturing conditions, in order to apply structural calculation theories that exist today.

Start by defining the limit state which shall not be exceeded to give the approval to the structure. Must be also calculated whether the system is statically stable, that is, if each bar compressed and / or bent supports buckling or not. It must be also proved that every joint withstand all the loads which are subjected.

6.2. CALCULUS THEORIES

6.2.1. First-order theory or stiffness method

The method is based on Castigliano's first theorem, which states that: *“If the strain energy of an elastic structure can be expressed generalized as a function of displacement q_i ; Then the partial derivative of the strain energy with respect to the generalized displacement gives generalized force Q_i ”* Alberto Castigliano (1847-1884).

This method is applicable to elastic systems, provided that the elastic energy can be expressed in terms of deformations.

In this case it is assumed that the structure behaves in a linear way, that is, an increase of external load corresponds to a proportional increase of deformations and internal forces. This behavior is originated in two suppositions: the material has a linear behavior and the deformations are near zero (the deformed position is near the initial position). While these conditions are carried out, it is possible to increase the loads and every solutions are valid. This drives to linear problems which can be denominated like first-order theory.

$$\mathbf{F} = \mathbf{K} \mathbf{A} \quad \mathbf{K}: \text{conventional stiffness matrix in first-order theory}$$

6.2.2. Second-order theory

Suppositions:

- The deformations are not near zero. Therefore the equilibrium equations must be considered in the deformed position, not in the initial position.
- The behavior of the material is linear-elastic. In some cases, particularly in ones where the slenderness is lower than a critic value, it should be considered that the behavior is not elastic (is plastic).

All of this leads to non-linear problems often referred to a second-order theory.

$$\mathbf{F} = (\mathbf{K} + \mathbf{K}_{\sigma}) \cdot \Delta$$

\mathbf{K}_{σ} is the geometric stiffness matrix of the whole structure, obtained by assembling all the relevant elements. The matrix depends on the axial stress in all cases.

6.3. APPLICATION OF THE THEORY TO THE CALCULATION OF A TUBULAR CHASSIS

As previously stated, the main objective in terms of structural calculations is the rigidity, that is, the deformation should be very small. From this is deduced that the theory to be applied is the first-order or the stiffness method, due to that their requirements are met and this theory is much easier to use than the second order.

6.4. LIMIT STATE

This section is going to try to determine the limit that should not be exceeded to give the approval to a chassis in terms of rigidity. In principle every bars of the chassis should be able to withstand all the loads that are applied. If one of them fails, that means that the *ultimate limit state* has been exceeded. Our intention is not only not to exceed the *ultimate limit state* but also not to reach a *limit state of deformation*.

When does one bar goes beyond the *ultimate limit state*?

When does one bar goes beyond the *deformation limit*?

6.4.1. Ultimate limit state

To calculate the ultimate limit state, the designer is faced with four methods of calculation:

Plastic-plastic procedure, transverse section type 1:

In this procedure is considered that is possible to develop total plasticity in the transverse section forming in this way a plastic ball-and-socket joint. The ultimate limit state is reached when the number of ball-and-socket joints is enough to produce a mechanism.

Elastic-plastic procedure, transverse section type 2:

In this procedure is considered that the ultimate limit state is reached when the first ball-and-socket joints appears.

Elastic-elastic procedure, transverse section type 3:

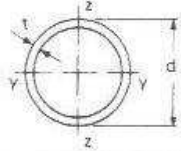
In this procedure is considered that the ultimate limit state is reached when the fibers of a transverse section start flowing.

Elastic-elastic procedure, transverse section type 4:

The transverse section is composed of thinner walls than those of type 3. It is necessary to consider, explicitly, the effects of local buckling when determining the last moment or the capability of resistance to compression of the transverse section.

The elements calculated by the first 3 methods should not bend locally before reaching their ultimate load limits; this means that the cross sections should not be thin-walled.

To satisfy this condition, the ratio **d/t** for circular tubular sections should not exceed certain maximum values. These ratios are different for transverse sections 1, 2 and 3. A transverse section must be classified according to the type less favorable of the elements under compression and/or flexion.



type of transverse section	compression and/or bending				
1	$d/t \leq 50 \epsilon^2$				
2	$d/t \leq 70 \epsilon^2$				
3	$d/t \leq 90 \epsilon^2$				
$\epsilon = \sqrt{\frac{235}{f_y}}$	f_y (N/mm ²)	235	275	355	460
	ϵ	1	0,92	0,81	0,72
	ϵ^2	1	0,85	0,66	0,51

Table 6.1: d/t ratio

For a quick determination of the kind of transverse section of a tubular profile, is shown below limiting values d/t for CHS profiles with different distributions of stress.

d/t ≤	Type of structural steel			
Type of transverse section	235	275	335	460
1	50	42,7	33,1	25,5
2	70	59,8	46,3	35,8
3	90	76,9	56,9	46

Table 6.2: Maximum limits d/t for transverse sections 1, 2 and 3 subjected to compression and/or bending

For the application of the procedures 'plastic-plastic' (type 1) and 'elastic-plastic' (type 2), the relationship between the ultimate tensile strength f_u and yield strength f_y must not be less than **1.2**.

$$f_u / f_y \geq 1.2$$

For the application of the procedure 'plastic-plastic', the unitary deformation ϵ_u corresponding to the ultimate tensile strength f_u must be at least 20 times the unitary deformation corresponding to the elastic limit f_y .

The types of steels for tubular profiles (CHS) cold or hot formed meet the requirements mentioned.

Type of cross section corresponding to a tubular chassis

The profiles used in a chassis have a maximum outer diameter of about **40 mm** and a thickness of about **2 mm**. This means that the ratio **d/t** will be at most around **20**. On the other hand, it is recommended the use of a high strength steel $f_y = 355 \text{ N/mm}^2$. The maximum upper limit of **d/t** more restrictive, with this type of steel, is the section corresponding to the type 1 with a value of **33.1**. As can be seen, it will not exceed the maximum limit using typical diameters and thicknesses for the construction of a chassis. Therefore, it is deduced that, it can be used to calculate even the section of type 1.

It has been stated earlier that significant deformation of the chassis is not wanted. This implies that the calculation will be done through a procedure elastic-elastic. For simplicity of calculation and because it also meets the requirement of the ratio d/t, it will be used the method of the transverse section (type 3), that is, the elastic-elastic which does not take into account the local buckling.

6.4.2. Deformation limit

It would be ideal to be able to impose to the chassis a maximum deformation limit on every node. But like everything in life, reality is only a vague reflection of the ideal world. It should be asked this question, how can the deformation be measured? For example, the deformation in bars could be measured by strain gauges, but what happens with the nodes? Firstly it would have to measure their initial position (without applying any loads) regarding to a fixed reference. When some loads are applied on the chassis simulating a driving state (very difficult if it is not driving), the new position of those nodes has changed regarding to the fixed position. The difference between the initial position and the final position must be measured. This cannot be made with the strain

gauges which can measure deformations in bars. The conclusion is that the current technology does not allow us to do this kind of measures.

Chassis builders rely on a concept that has already been mentioned above: the torsion stiffness. All the bars must withstand the efforts and must not be in local buckling, and, apart from that, the entire structure must have satisfactory torsion rigidity. In theory, the minimum torsion stiffness required would be one with which the relative deformation between the front and rear suspension would not be large enough to alter the driving characteristics provided while designing the suspension. The design of the suspension is made supposing that the chassis is a rigid solid, something that does not match with reality.

How to measure the torsion rigidity? The torsion stiffness of interest is the measure between the front and rear suspension. Ideally, it would have two infinite rigid plates, the first one joins the brackets of the trapezoids of the rear suspension and the other one joins the brackets of the triangles above the front suspension. For example, it would be fixed the rear plate, and, in the front plate, it would be applied a torque. It would be measured the angle rotated; therefore, it could know the torsion stiffness by the following equation:

$$\text{Torsion rigidity (Kg}\cdot\text{m)} = \frac{\text{Applied torque (Kg}\cdot\text{m)}}{\text{Angle rotated (}^\circ\text{)}}$$

The image below shows how to measure the torsion rigidity of a chassis.



Fig. 6.1: The rear part of the chassis is fixed and, with a beam, one worker applies a torque using his own weight

6.5. VERIFICATION OF THE BARS

6.5.1. Creep test in the bars

In this section is assumed that the operation of the method of stiffness is known. There are many programs to calculate structures using this method. The results obtained are axial forces, bending moments and torsion moments applied at the bars.

The first thing to check is if the bars reach an ultimate limit state. As was agreed earlier, the ultimate limit state is in which none of the points of any section reaches the creep stress. It can be applied the Von Mises criteria. From a state of stress in a determined point, this method finds one stress which can be compared with the limit stress of the material. This stress is called stress comparison and is equal to the following expression:

$$\sigma_{VM} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}}$$

Where σ_1 , σ_2 , σ_3 are the main stresses of the stress state, with its own sign. These main stresses are obtained by the Mohr's circle. In the case of the lengths of the bars used in a chassis, the shear stresses created from the shear stress Q are much lower than those produced from the axial force N , the torque T and the bending moment M . Given this, the worst stress state would be the next.

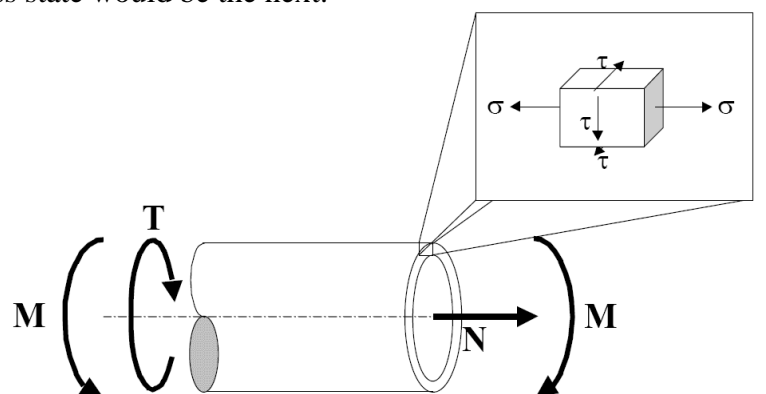


Fig. 6.2: Stress state of the bars.

Where $\sigma = \frac{N}{A} + \frac{M \cdot D}{I}$

$\tau = \frac{T \cdot D}{I_p}$

One of the planes of the element under consideration will have the following stress state:

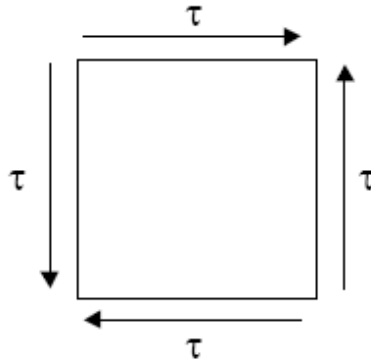


Fig. 6.3: Stress state, plane 1.

If it is made a 45 degrees turn, in the same plane, the equivalent stress state will be the next:

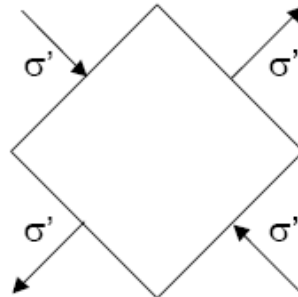


Fig. 6.4: Oblique's view of the stress state.

With this plane and these two pictures it can be found the first Mohr's circle.

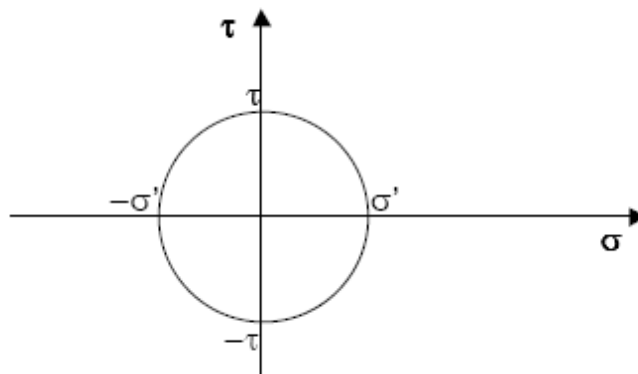


Fig.6.5: First Mohr's circle

The stress state of the perpendicular planes is the next:

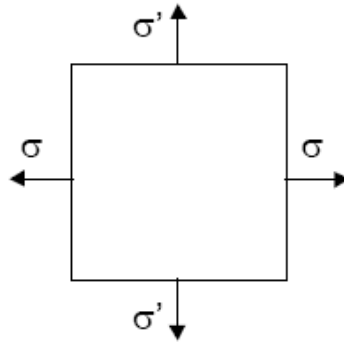


Fig. 6.6: Stress state, plane 2.

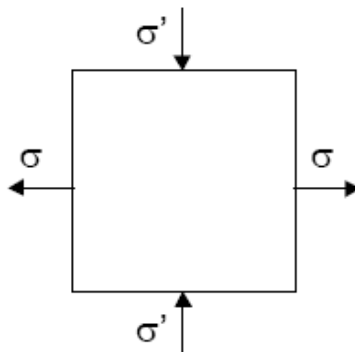


Fig. 6.7: Stress state, plane 3.

These stress states have their following corresponding Mohr circles.

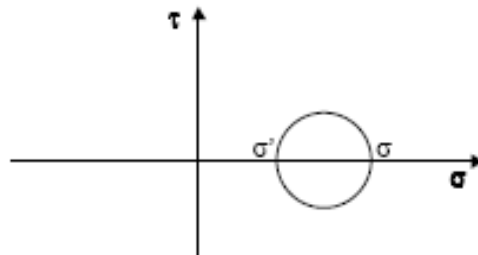


Fig. 6.8: Second Mohr's circle

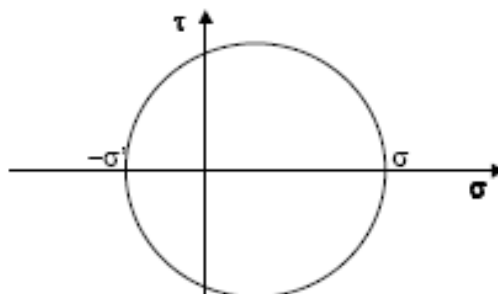


Fig. 6.9: Third Mohr's circle

Therefore, it follows that the set of Mohr circles is:

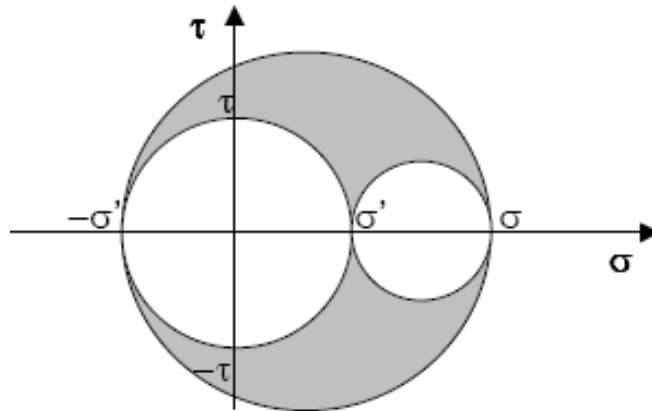


Fig.6.10: Set of Mohr circles

The possible stress states are those belonging to the shaded area. Hence it is obtained:

$$\sigma_1 = \sigma = \frac{N}{A} + \frac{M \cdot \frac{D}{2}}{I} \qquad \sigma_2 = \tau = \frac{T \frac{D}{2}}{I_p} \qquad \sigma_3 = -\tau = -\frac{T \frac{D}{2}}{I_p}$$

If these expressions are substituted in the Von Mises (σ_{VM}) expression:

$$\sigma_{VM} = \sqrt{\left(\frac{N}{A} + \frac{M \cdot \frac{D}{2}}{I}\right)^2 + 3\left(\frac{T \frac{D}{2}}{I_p}\right)^2}$$

If it is considered high resistance steel, the creep limit would be $f_y = 355 \text{ N/mm}^2$. This creep limit must be reduced by a partial safety coefficient of the material properties γ_M . In the calculation of the ultimate state for the class sections 1, 2 and 3, γ_M takes the following value $\gamma_M = 1.1$. Thus, it is found that the creep limit needed to make the calculation is $f_{yd} = 322.73 \text{ N/mm}^2$. If the Von Mises stress in the bar does not exceed this limit, the solution can be given as valid in that it does not reach the ultimate limit state.

6.5.2. Checking of the stability of the bars

A bar that does not reach ultimate limit state still has the possibility of 'failure'. From a given load the bar can no longer be stable, or, in other words, can be buckling, depending on their slenderness and boundary conditions.

Generally, is not necessary to test the resistance to lateral buckling with torsion for tubular profiles which in practice, are commonly used. This is because the torsion module I_t is very large compared with open profiles. With this in mind, the most general case for bars, will be one in which they are subjected to compressive and bending moments. For transverse sections of class 3, the relationships that must be satisfied are:

$$\frac{N_{sd}}{\chi \cdot A} + K_y \cdot \frac{M_{y,sd}}{W_{el,y}} + K_z \cdot \frac{M_{z,sd}}{W_{el,z}} \leq f_y$$

Where:

N_{sd} : Value of axial compression.

$M_{y,sd}$, $M_{z,sd}$: Values of absolute maximum bending moment around the **y-y** or **z-z** axis according to the theory of first order.

A: Section area.

$W_{el,y}$, $W_{el,z}$: Elastic modulus for **y-y** or **z-z** axis. For CHS bars the value is $\frac{2 \cdot I_y}{D}$ and $\frac{2 \cdot I_z}{D}$ respectively.

χ : Reduction factor for buckling curves. It takes the minimum between χ_y and χ_z depending on the **y-y** or **z-z** axis.

K_y , K_z : Expansion coefficient of the bar.

$f_{yd} = f_y / \gamma_M$: Creep limit of calculation.

Historically, the buckling of a column under centered compression is the oldest stability problem which was already studied by Euler. Currently, the calculation of buckling of a steel element subjected to compression is performed in most European countries, using the 'European buckling curves'. These curves are based on extensive researches, both experimental and techniques, which take into account, specially, the mechanical imperfections (residual stress, elastic limit distribution) and geometric (linear deviation) existing on the bars. It is mainly used multiple buckling curves (Eurocode 3). From these curves is obtained the factor χ if it is known the slenderness λ of the bar. As shown in the figure below there are 4 buckling curves **a₀, a, b, c**.

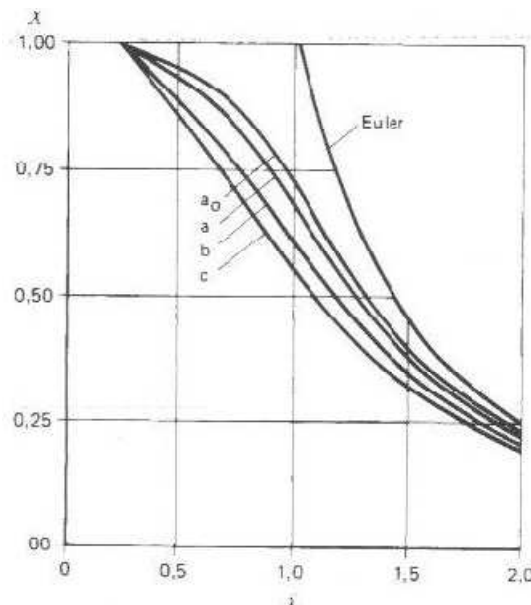


Fig. 6.11: European buckling curves and Euler curve.

To make a exhaustive calculation of buckling it should be determined as closely as possible the reduction factors χ and the expansion coefficients K_y and K_z for each bar. This involves hard work, not at all automatable, given the fact that it is required, among other things, that the engineer must decide often, with dubious criteria, the buckling length of each bar. To expedite the design process, it has been decided to do the calculations in a way that may be exaggerated but in reality it is not so. The bars work at most with a stress range around 35% of what they can really withstand. This is because, as it has been stated more than once, the primary objective in behavior and structure is rigidity. So even choosing the worst case of buckling, the bar should not fail. In case of failure of a buckling bar, it would be better to refine the calculation. It should be consulted on Eurocode 3.

Determination of the reduction factor χ for buckling curves:

The buckling curves a_0 , a, b, c depend on the type of transverse section. They are mainly based on the different levels of residual stresses that arise because of the different manufacturing processes:

Manufacturing process	Buckling curves
Hot formed	a
Cold formed (using f_{yb})	b
Cold formed (using f_{ya})	c

Table 6.3: Buckling curves for different manufacturing processes.

Where:

f_{yb} : elastic limit of the material (without cold forming)

f_{ya} : elastic limit of the material after cold forming

The 'a₀' buckling curve can be used instead of 'a' tubular shaped for hot formed profiles which use steel S460.

The most usual case in a tubular chassis is the one of the curve 'b'. The limits of the factor χ for this case are: $0.0672 \leq \chi \leq 1$. As it will be taken the worst case of all, it will be chosen $\chi = 0.0672$.

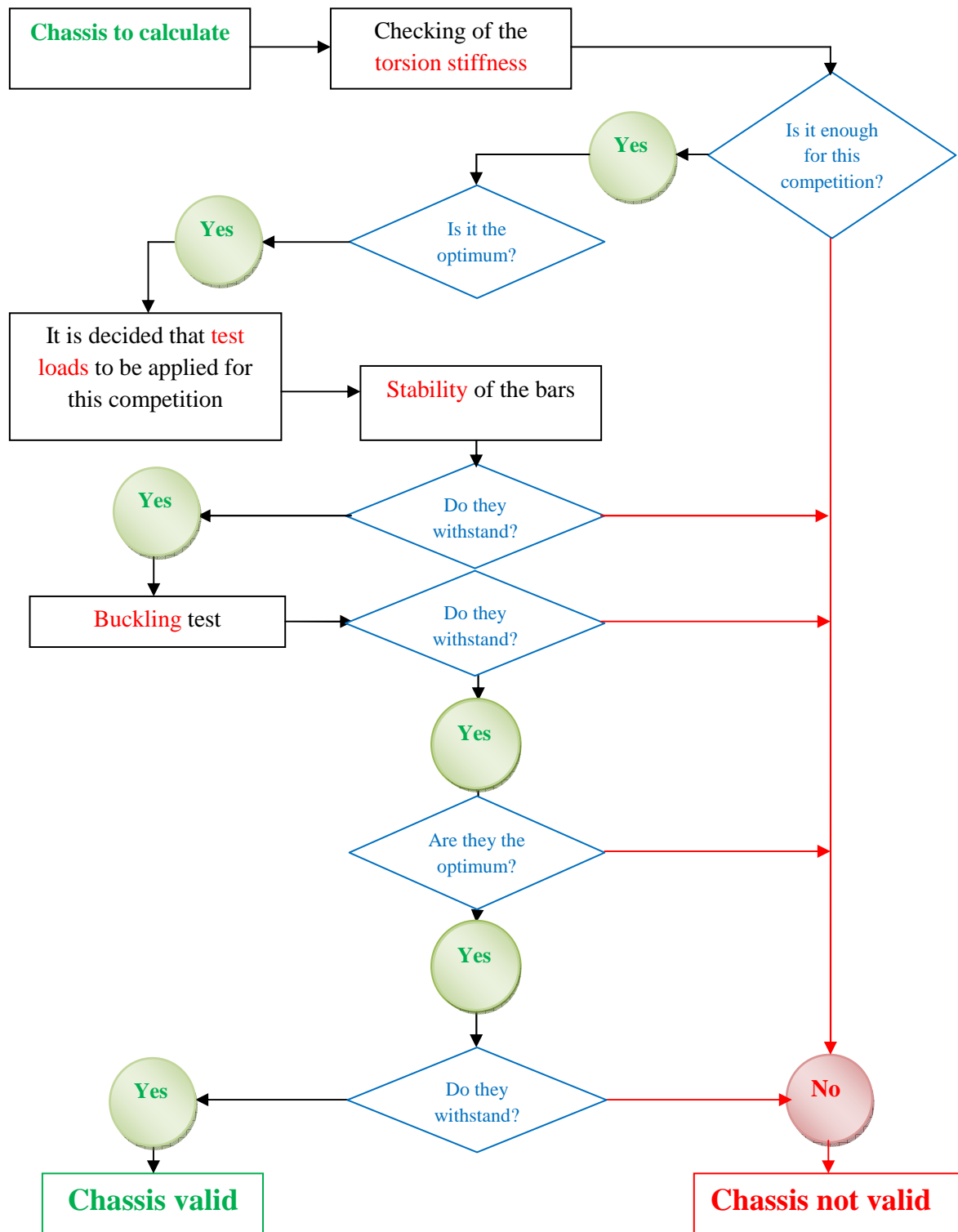
Determination of the coefficients of expansion of the bar, K_y and K_z :

These coefficients depend on the applied compression load, the χ_y and χ_z coefficients, the slenderness, the type of moment diagram, among others. This gives an idea of the complexity of its determination. Its upper limit is 1.5. Thus, it is taken $K_y=1.5$ and $K_z=1.5$.

Point of the bar where it will be applied the calculation:

The stiffness method will give us the stresses which it is subjected each bar. These stresses will vary along each bar, including the axial stress, given the fact that the loads will be applied along the bars and not only at the nodes. According to the theory of stability, the point of the bar, where the calculation of buckling will be made, depends on the boundary conditions. It has been taken the worst case of all, with that, it is not necessary to decide the boundary conditions of each bar, so in this particular case, is not clear where to apply the general expression of buckling. Because of that, the general expression of buckling will be applied in all of those points of each bar in which we will have the results of the stresses. This can be easily done with computer programs such as Excel. If any of them fails, the calculation of buckling of the concrete bar must be refined as it is shown in the Eurocode 3.

6.6. ALGORITHM FOR CALCULATION OF A TUBULAR CHASSIS



7. CALCULATION OF THE CHASSIS

7.1 INTRODUCTION

To begin the calculation of the chassis of a kart-cross racing is needed to make an evaluation of the needs that will cover the kart.

The main utility of the vehicle will be preparation for kart-cross circuits where it will be able to develop their full benefits. In its design is has been provided all mandatory requirements, as it is dictated by the technical regulations, for approval.

This project is focused on the design of the chassis, so the parameters that must be known are: weight of the chassis, engine, weight of the driver, wheel diameter, geometry and composition of the chassis, etc. In order to meet these needs we have chosen the following configuration:

- The engine that reflects the best the demands require an engine power of around 100 hp. The most commonly used engines are the ones from bikes such as Honda CBR600 or Yamaha R-6 ... The engine is a 4-stroke engine with 600 cc and maximum torque of 66 Nm at 11,250 rpm.
- The minimum weight of the vehicle can never be less than 310 kg without driver, without fuel, without water in the sprayer and running order. The use of ballast is forbidden.
- It is a steel plate placed between the driver's seat and the front of the vehicle, such as a mark of the regulations, for safety reasons related to the pilot's legs.
- The dimensions set by the regulation are:
 - o Length 2600 mm.
 - o Width 1600mm.
 - o High 1400mm. (Excluding siege plate located on the top of the chassis)
- Chassis: tubular. Built in metallic materials, usually carbon steel cold formed without welding. This tube has a high resistance in relation to their weight. The joints between the tubes are made by welding. The chassis must be strong in case of accident and as light as possible. It is mostly used 40x2 tubes for central

and front arches and 30x2 tubes for rear braces and rest of chassis. The properly triangulation carried provides excellent stiffness.



Fig. 7.1: 40x2 tubes

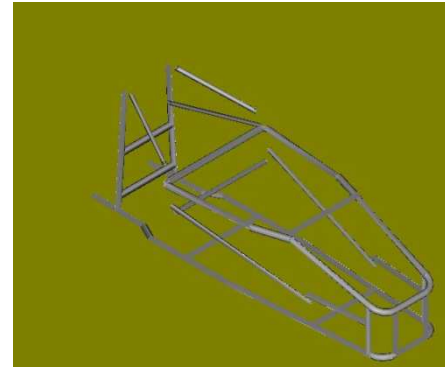


Fig. 7.2: 30x2 tubes

In a subsequent annex of this project, it will be shown all the features and technical regulations that must be obeyed with this type of karts.

7.2. LOAD HYPHOTESIS

In order to make the complete design of the chassis, it had to be subjected to the maximum stresses possible. To perform the calculations it has been made these assumptions:

1. Maximum acceleration: The maximum acceleration of a kartcross can be up to 2.15G. This measure is taken when the kartcross accelerates as possible in a long straight. To ensure the calculations, as the maximum longitudinal acceleration is taken $\mathbf{a_x = 2.5 G (m/s^2)}$
2. Braking. The maximum deceleration or braking is taken a value of $\mathbf{a_x = -1.5 G (m/s^2)}$ due to the maximum deceleration measured in a circuit by a kartcross is - 1.35 G.
3. Cornering: The value chosen for the calculations at this stage, when the kartcross is cornering, it will be $\mathbf{a_x = 1G (m/s^2)}$ due to the maximum lateral acceleration measured in a circuit by a kartcross is 0.9 G

These test loads have been made from the several data measured in the tests of a competition kartcross approved in a circuit. There are some efforts have not been measured because of lack of means, such as aerodynamics, although the speeds are not very large, so that in an estimate calculation they can be neglected. These hypotheses try to represent cases where the vehicle is suffering the most. Maybe in another type of car interest other load assumptions deemed appropriate. In the calculation of vehicle design, that is, not yet built, the interesting thing would be to have a history of data obtained from the largest possible number of tests of different cars and from these data to make the corresponding loading assumptions. Also in the calculation of a chassis design it should be taken into account a load increase coefficient for the loads as was indicated in the report.

7.3. GEOMETRIC MODELLING OF THE CHASSIS

The ANSYS program uses the stiffness method to calculate the structure. This requires that all the modeled geometry with straight lines. The lines drawn in ANSYS will generally pass through the neutral line of bars. The material used for steel bars is the S355, which has the following mechanical properties:

- $E = 2100000 \text{ kg/cm}^2$
- $G = 800,000 \text{ kg/cm}^2$
- $\sigma_{\text{calculus}} = 3293.14 \text{ kg/cm}^2$

7.4. LOAD DISTRIBUTION OF THE CHASSIS

In the following figure can be seen the distribution of forces in the front and in the rear axle. The total weight of the chassis is $W = 310 \text{ Kg}$

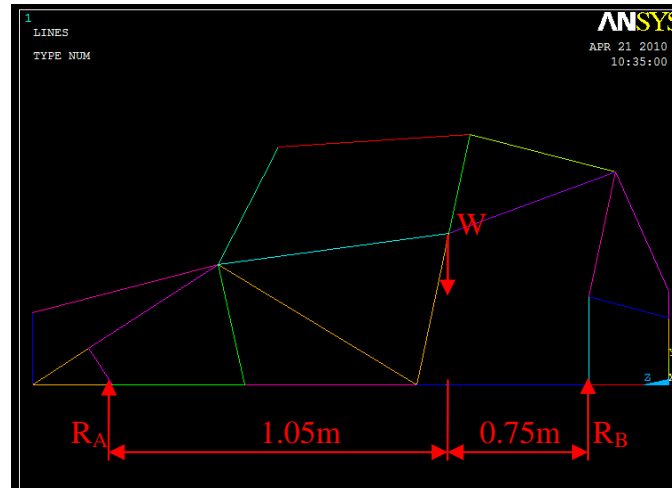


Fig. 7.3: Load distribution

By the following calculations it can be said that:

$$\sum F = 0 \quad \text{and} \quad \sum M_B = 0$$

$$R_A + R_B - W = 0 \rightarrow R_A + R_B = W = 310 \cdot 9.8 = 3038\text{N}$$

$$1.05 \cdot R_A = 0.75 \cdot W \rightarrow R_A = 1265.83 \text{ N} \quad R_B = 1772.17 \text{ N}$$

7.5. THE APPLICATION OF THE LOAD HYPHOTESIS IN THE CHASSIS

Previously has been said which the load hypothesis that will be applied on the chassis were, but they were not specifically described.

7.5.1. Maximum acceleration:

1. Inertia of the pilot on the seat's brackets:

The inertia will be the force exerted by the pilot, it will be its mass multiplied by the maximum acceleration and it is applied on the four seat brackets as is shown in the figure below:

$$F = (M \cdot a) / 4$$

$$= 75 \cdot 9.8 \cdot 2.5 / 4$$

$$= 460 \text{ N}$$

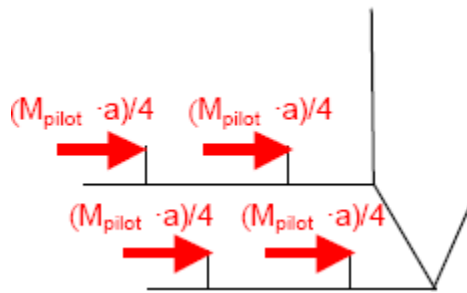


Fig. 7.4: Inertia of the pilot on the four seat's brackets

The 95% of the cases, it is chosen as the mass of the pilot a value of 75 Kg, therefore, in this example, the mass will be 75 Kg. And, as it was said before, the acceleration will be $a = 2.5g \text{ m/s}^2$.

2. Inertia of the engine on the engine brackets

The mass of the motor is approximately 95 kg and the centre of gravity has been placed like is shown in the figure. The figure below shows the balance of forces:

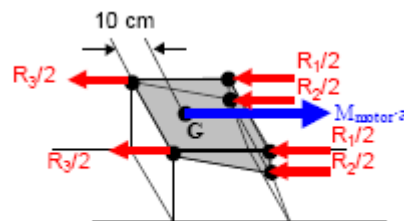


Fig. 7.5: Load balance of the engine inertia

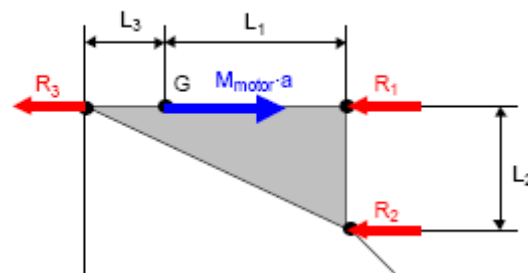


Fig. 7.6: Other view of the load balance of the engine inertia

From the equations of the static obtained:

$$R_1 + R_2 + R_3 = M_{\text{engine}} \cdot a$$

$$R_2 \cdot L_2 = 0$$

As it can be seen, is a statically indeterminate system with a degree of hyperstaticity. To solve it is assumed that in the top bar the deformation, from the centre of gravity, on one side and the other is the same:

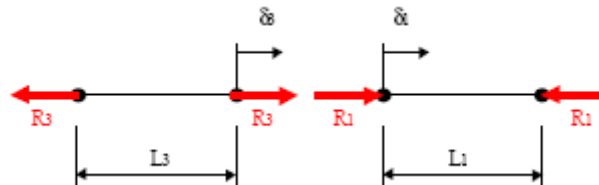


Fig. 7.7: The upper fiber of the engine modeling

The compatibility equation of movement is:

$$\delta_3 = (R_3 \cdot L_3) / (E \cdot A) = \delta_1 = (R_1 \cdot L_1) / (E \cdot A)$$

Replacing the values of dimensions and forces is to be applied to the chassis are:

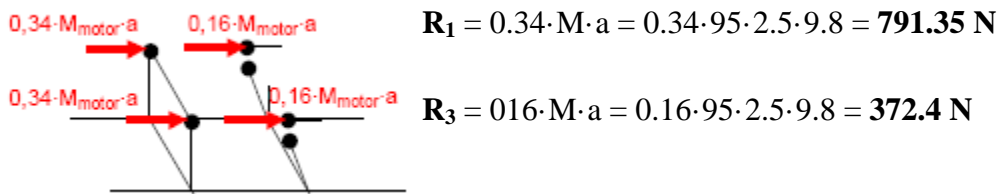


Fig. 7.8: The inertia reactions of the engine in the engine brackets

3. Inertia of the chassis itself

The density of steel is $\rho = 7850 \text{ kg/m}^3$. It must be applied a distributed load in all bars modeled less on which modulate the steering, the engine, the transmission and those that symbolize welded joints with an area of 10^5 cm^2 . The distributed load has the following value:

$$q = \rho \cdot A \cdot a \text{ [kg / cm]} = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (2.5 \text{ g}) = 45.9 \text{ Kg / m}$$

Being:

A: Section area; a: acceleration; $g = 9.8 \text{ m/s}^2$

The direction of the force is the same than the kart advance and with the opposite direction of acceleration of the vehicle as it is a reaction to progress.

4. Weight of the chassis itself

The density of steel is $\rho = 7850 \text{ kg/m}^3$. It must be applied a distributed load in all bars modeled less on which modulate the steering, the engine, the transmission and those that symbolize welded joints with an area of 10^5 cm^2 . The distributed load has the following value:

$$q = \rho \cdot A \cdot g \text{ [kg / cm]} = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (g) = 18.37 \text{ Kg/m}$$

Being:

A: Section area; $g = 9.8 \text{ m/s}^2$. The direction is always towards the ground.

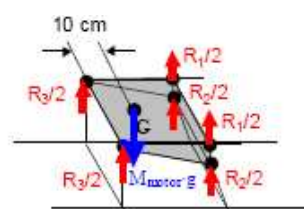
5. Weight of the engine

The equations of statics are:

$$R1 + R2 + R3 = F$$

$$F \times L1 = R3 \cdot (L1 + L3)$$

The system has a degree of hyperstaticity to be resolved in the following way: The R_1 brackets are more horizontal than those of R_2 . The deformation flexion is greater than the compression by orders of magnitude; therefore, it is considered that the effort is entirely absorbed in the vertical, by R_2 .



$$L_3 = 10 \text{ cm}, L_1 = 20 \text{ cm}$$

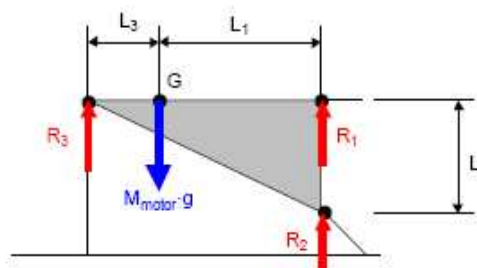


Fig. 7.9: Load balance of the weight of the pilot

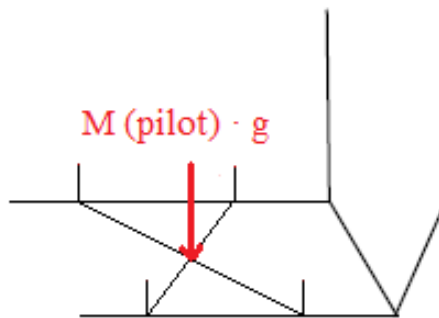
Then the reactions are:

$$R_1 = 0, R_2 = 31.66 \text{ kg}, R_3 = 63.33 \text{ kg}$$

Note that the direction of the force on the chassis is downward, not toward above. In the figures below the engine balance has been represented.

6. Weight of the pilot

As it was said before, the 95% of the cases the mass of the pilot is taken $M = 75\text{kg}$. In this design it will be taken $M = 75$ too. Therefore, the weight of the pilot will be $M \cdot g$ and it will be applied in the line which passes through the centre of gravity of the seat. It is shown in the following figure.



$$\text{With } F = Mg = 75 \cdot 9.81 = 736 \text{ N}$$

Fig.7.10: Weight of the pilot

Six load hypotheses were analyzed to design the chassis. But there are some more hypotheses which should be analyzed, including:

- The inertia of the transmission
- The inertia of the steering
- The force transmitted by the grip of the tyres to the asphalt.
- The inertia of the suspensions

In the section of futures lines of work is mentioned that it would be a good idea make the complete design of the chassis, including suspensions, transmission, steering, tyres...etc. In that future design, all the possible assumptions will be taken into account.

In the following figure is shown how would be the load system applied to the chassis when it is in maximum acceleration:

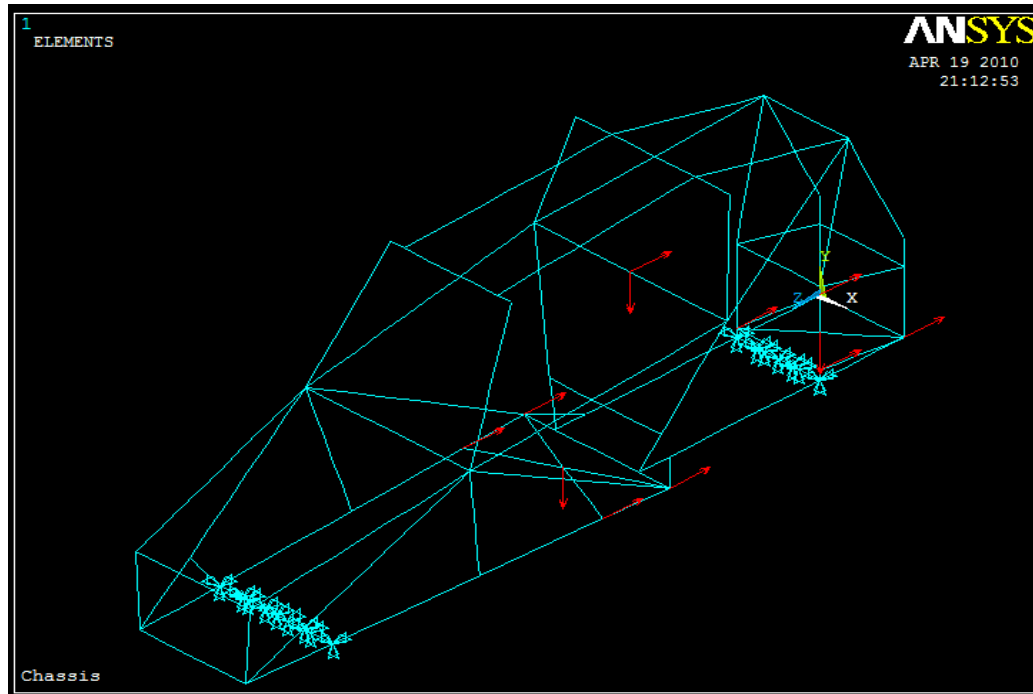


Fig.7.11: Load balance when accelerating

And the deformed shape in an excessive way is like it is shown below:

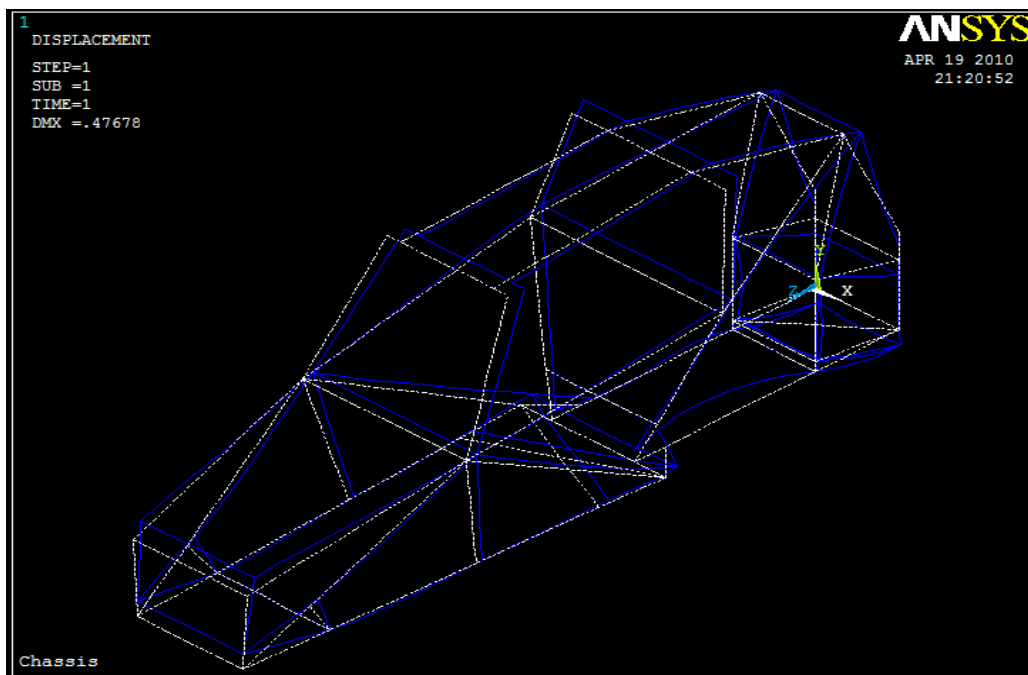


Fig. 7.12: Deformed shape of the chassis when it is accelerating with $a = 2.5g$

Here, it can be seen which part of the chassis suffers the most.

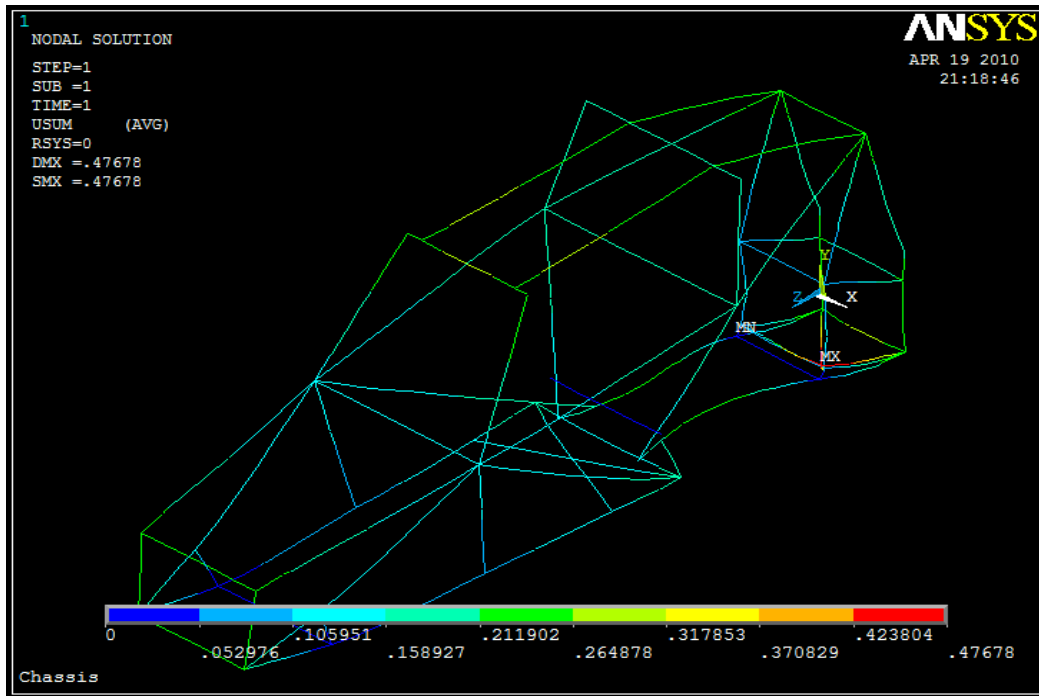


Fig. 7.13: Stress distribution when the kart accelerates

7.5.2. Maximum deceleration:

1. Inertia of the pilot on the seat belt brackets:

When braking, the driver lifts off the seat and is restrained by the seat belt. The load sharing between the different brackets of the belt is complex to determine, so it will be assumed that is equal for all.

$$F = (M \cdot d) / 4$$

$$= 75 \cdot 9.8 \cdot 1.5 / 4$$

$$= 276 \text{ N}$$

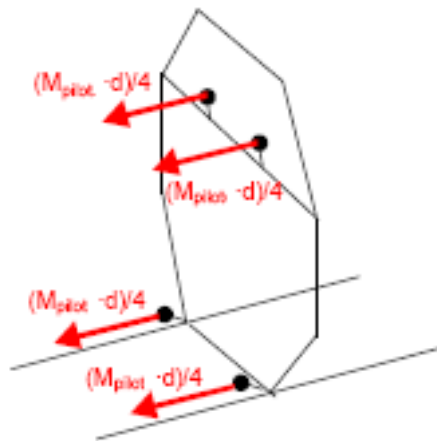


Fig. 7.14: Inertia of the pilot on the seat belt brackets

The 95% of the cases, it is chosen as the mass of the pilot a value of 75 Kg, therefore, in this example, the mass will be 75 Kg. And, as it was said before, the acceleration will be $d = -1.5g \text{ m/s}^2$.

2. Inertia of the engine on the engine brackets

The mass of the motor is approximately 95 kg and the centre of gravity has been placed like is shown in the figure. Taking a similar load balance than in the hypothesis of maximum acceleration, in this case we have the following reactions:

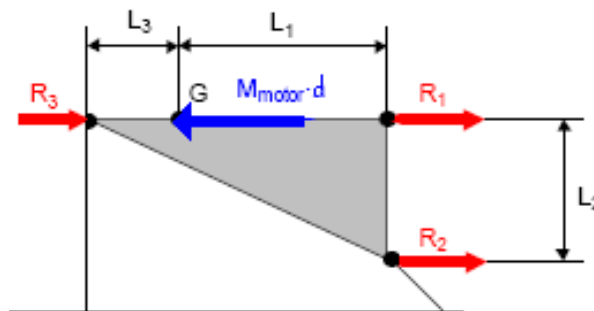


Fig. 7.15: View of the load balance of the engine inertia

From the equations of the static obtained:

$$R_1 + R_2 + R_3 = M_{\text{engine}} \cdot d$$

$$R_2 \cdot L_2 = 0$$

As it can be seen, is a statically indeterminate system with a degree of hyperstaticity. To solve it is assumed that in the top bar the deformation, from the centre of gravity, on one side and the other is the same:

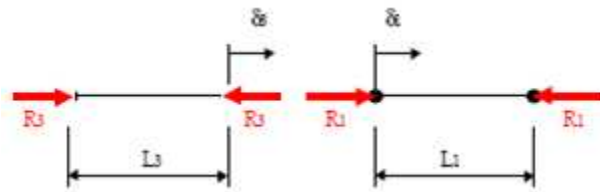
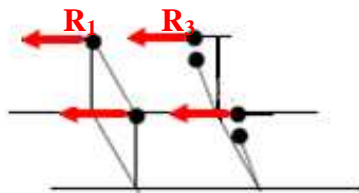


Fig. 7.16: The upper fiber of the engine modeling

The compatibility equation of movement is:

$$\delta_3 = (R_3 \cdot L_3) / (E \cdot A) = \delta_1 = (R_1 \cdot L_1) / (E \cdot A)$$

Replacing the values of dimensions and forces is to be applied to the chassis are:



$$R_1 = 0.32 \cdot M \cdot d = 0.32 \cdot 95 \cdot 1.5 \cdot 9.8 = 447 \text{ N}$$

$$R_3 = 0.68 \cdot M \cdot d = 0.68 \cdot 95 \cdot 1.5 \cdot 9.8 = 950 \text{ N}$$

(d is positive because the direction of the loads is in the same direction of motion)

Fig. 7.17: The inertia reactions of the engine in the engine brackets

3. Inertia of the chassis itself

The method of calculation is similar than in the case of maximum acceleration.

$$q = \rho \cdot A \cdot d \text{ [kg / cm]} = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (1.5 \text{ g}) = 27.54 \text{ Kg / m}$$

Being:

A: Section area; d: deceleration; g = 9.8 m/s²

The direction of the force is the same than the kart's advance and with the same direction of acceleration (deceleration, negative) of the vehicle.

4. Weight of the chassis itself

The method of calculation is similar than in the case of maximum acceleration.

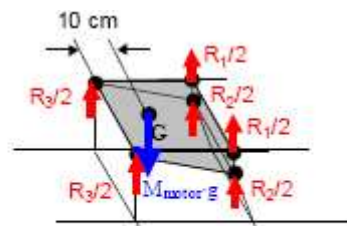
$$q = \rho \cdot A \cdot g \text{ [kg / cm]} = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (g) = 18.37 \text{ Kg/m}$$

Being:

A: Section area; $g = 9.8 \text{ m/s}^2$. The direction is always towards the ground.

5. Weight of the engine

The method of calculation is similar than in the case of maximum acceleration.



$$L_3 = 10 \text{ cm}, L_1 = 20 \text{ cm}$$

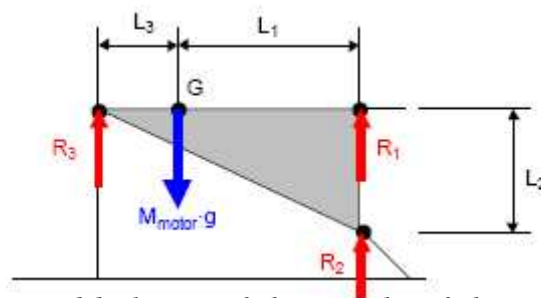


Fig 7.18: Load balance of the weight of the pilot

Is then:

$$R_1 = 0, R_2 = 31.66 \text{ kg}, R_3 = 63.33 \text{ kg}$$

Note that the direction of the force on the chassis is downward, not toward above. In the figures below the engine balance has been represented.

6. Weight of the pilot

The method of calculation is similar than in the case of maximum acceleration.

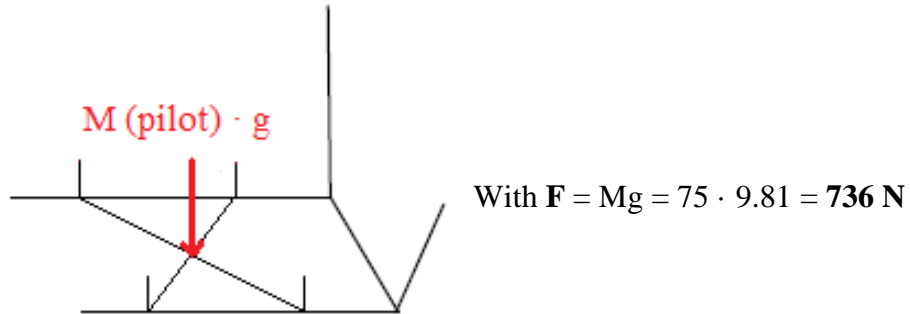


Fig 7.19: Weight of the pilot

In the following figure is shown how would be the load system applied to the chassis when it is in maximum deceleration:

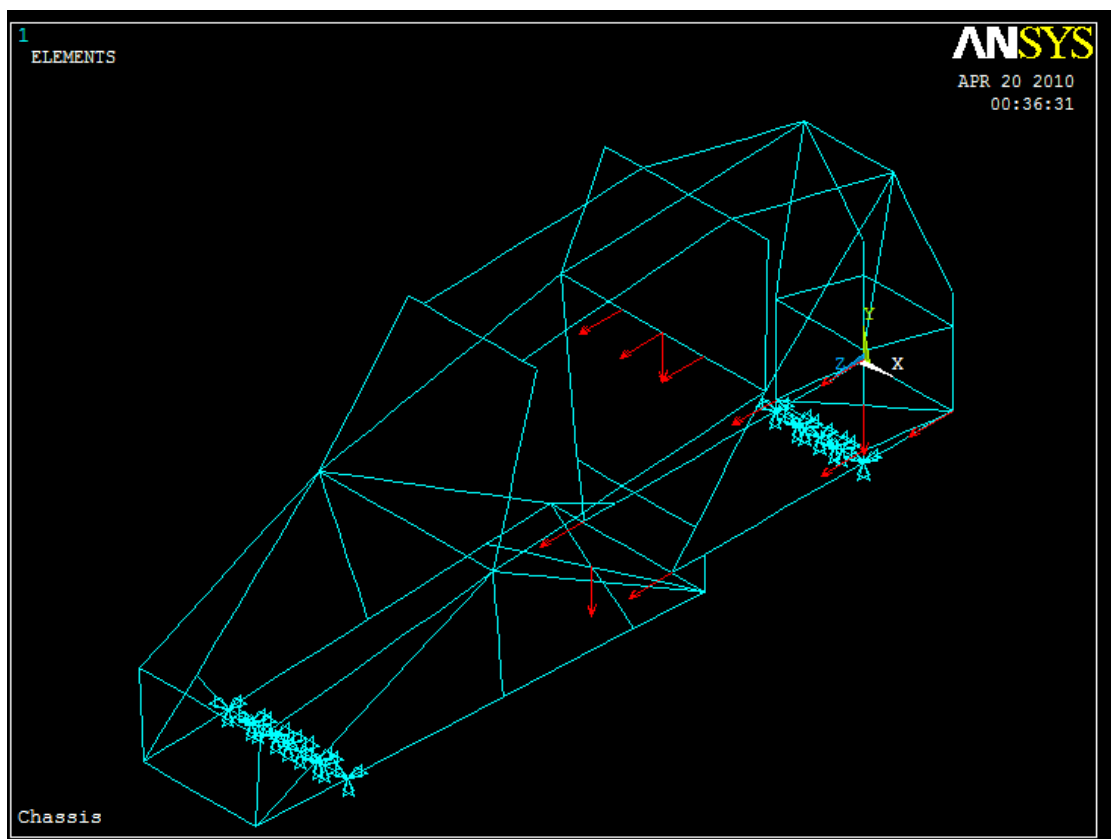


Fig.7.20: Load balance when the kart brakes

And the deformed shape in an excessive way is like it is shown below:

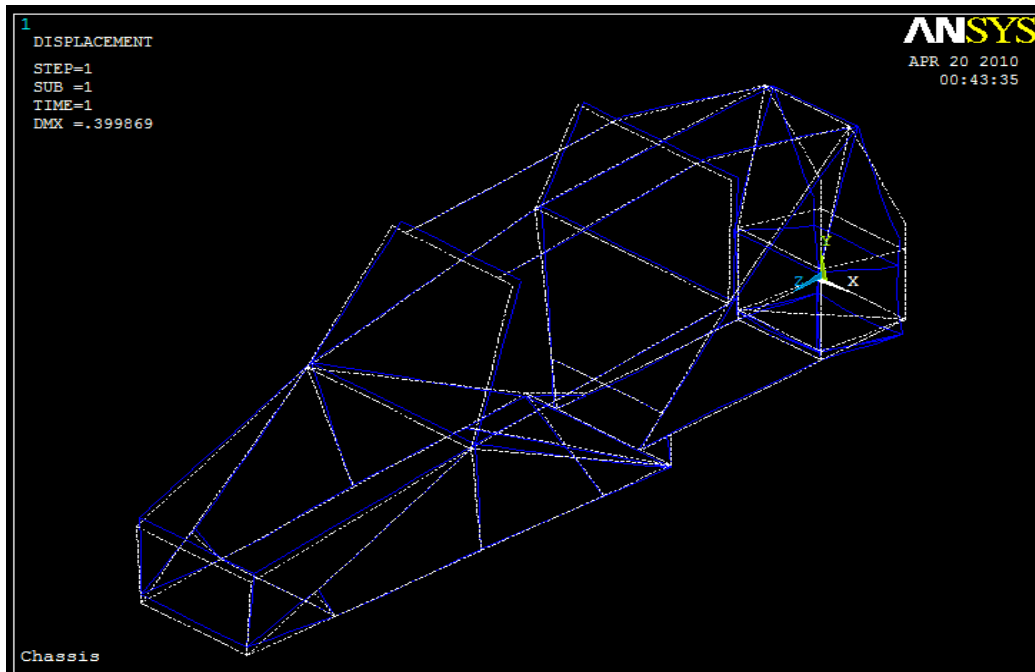


Fig. 7.21: Deformed shape of the chassis when it is accelerating with $d = 1.5g$

Here, it can be seen which part of the chassis suffers the most.

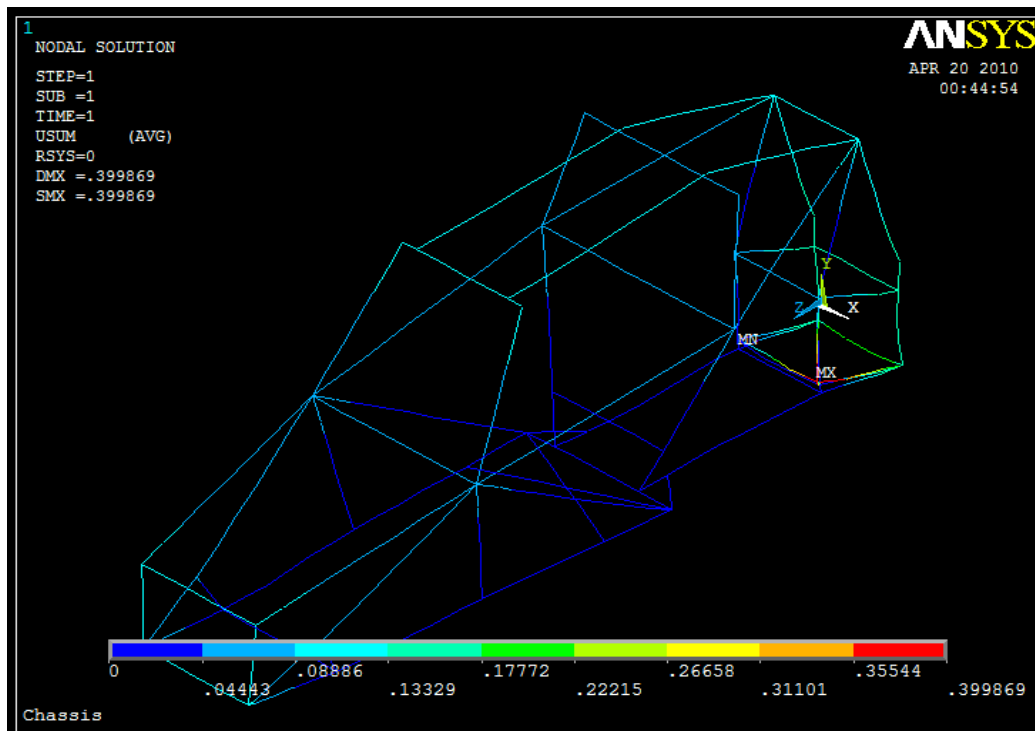


Fig. 7.22: Stress distribution when the kart brakes

7.5.3. Cornering:

1. Inertia of the pilot on the seat belt brackets:

It is assumed that the effort is applied on the seat brackets and not on the seat belt brackets. It is a left turn, and then the distribution of loads is as follows:

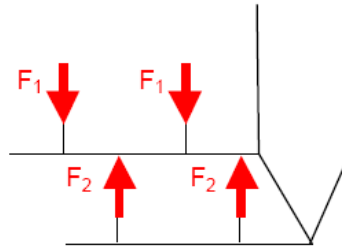


Fig. 7.23: Inertia of the pilot on the seat belt brackets

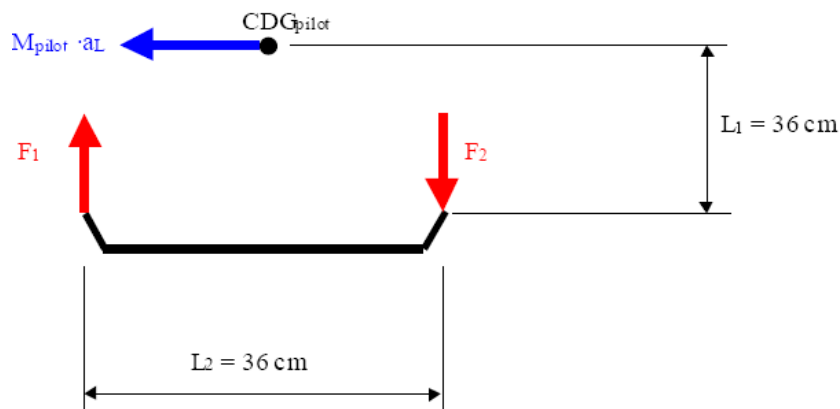


Fig. 7.24: Other view of the inertia of the pilot on the seat belt brackets

$$F_1 = F_2$$

$$M_{\text{pilot}} \cdot a_L \cdot L_1 = F_1 \cdot L_2 / 2 - F_2 \cdot L_2 / 2 \quad \rightarrow \quad F_1 = F_2 = M_{\text{pilot}} \cdot a_L / 2 = 75 \cdot 9.8 / 2 = 367.5 \text{ N}$$

The 95% of the cases, it is chosen as the mass of the pilot a value of 75 Kg, therefore, in this example, the mass will be 75 Kg. And, as it was said before, the acceleration will be $a_L = 1 \cdot g \text{ m/s}^2$ (lateral acceleration).

2. Inertia of the engine on the engine brackets

Taking a balance similar than in the previous scenario, in this case we have the following reactions:

$$\mathbf{R}_1 = \mathbf{M}_{\text{engine}} \cdot \mathbf{a}_L / 2, \mathbf{R}_2 = \mathbf{0}, \mathbf{R}_3 = \mathbf{M}_{\text{engine}} \cdot \mathbf{a}_L / 2$$

$$\mathbf{R}_1 = \mathbf{M}_{\text{engine}} \cdot \mathbf{a}_L / 2 = 95 \cdot 9.8 / 2 = 465.5 \text{ N}$$

$$\mathbf{R}_3 = \mathbf{M}_{\text{engine}} \cdot \mathbf{a}_L / 2 = 95 \cdot 9.8 / 2 = 465.5 \text{ N}$$

The direction of the loads is the reverse of the lateral acceleration.

3. Inertia of the chassis itself

The method of calculation is similar than in the case of maximum acceleration.

$$\mathbf{q} = \rho \cdot \mathbf{A} \cdot \mathbf{a}_L [\text{kg} / \text{cm}] = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (1 \text{ g}) = 18.36 \text{ Kg} / \text{m}$$

Being:

A: Section area; **a_L**: lateral acceleration; **g** = 9.8 m/s²

The direction of the force is the same than the kart's lateral acceleration.

4. Weight of the chassis itself

The method of calculation is similar than in the case of maximum acceleration.

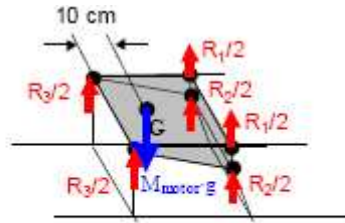
$$\mathbf{q} = \rho \cdot \mathbf{A} \cdot \mathbf{g} [\text{kg} / \text{cm}] = (7.85 \cdot 10^{-3}) \cdot [(4.0^2 - 3.6^2) \cdot \pi/4] \cdot (\text{g}) = 18.37 \text{ Kg} / \text{m}$$

Being:

A: Section area; **g** = 9.8 m/s². The direction is always towards the ground.

5. Weight of the engine

The method of calculation is similar than in the case of maximum acceleration.



$$L_3 = 10 \text{ cm}, L_1 = 20 \text{ cm}$$

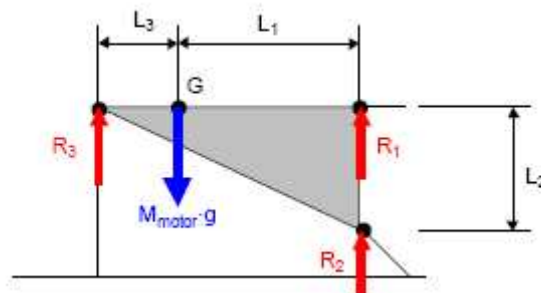


Fig. 7.25: Load balance of the weight of the pilot

Is then:

$$R_1 = 0, R_2 = 31.66 \text{ kg}, R_3 = 63.33 \text{ kg}$$

Note that the direction of the force on the chassis is downward, not toward above. In the figures below the engine balance has been represented.

6. Weight of the pilot

The method of calculation is similar than in the case of maximum acceleration.

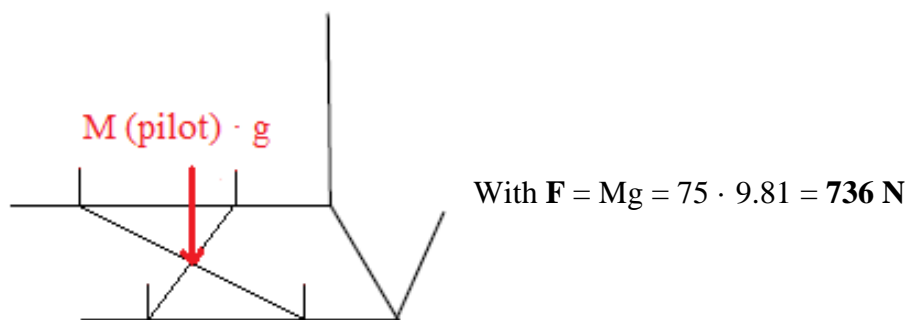


Fig. 7.26: Weight of the pilot

In the following figure is shown how would be the load system applied to the chassis when it is cornering:

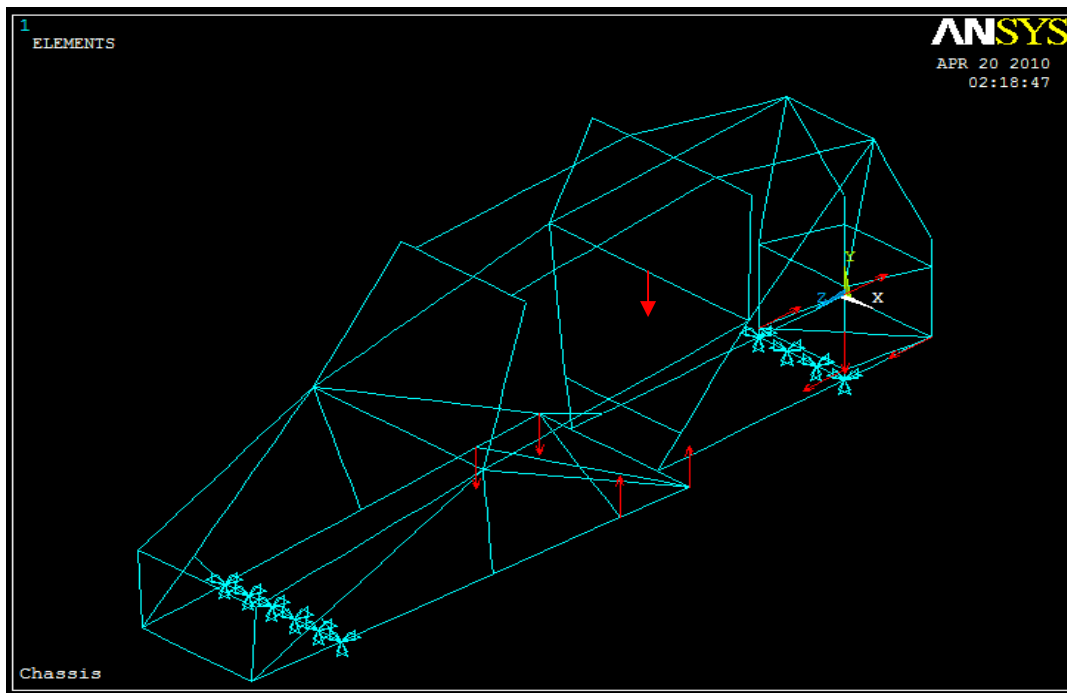


Fig.7.27: Load balance when the kart is cornering

And the deformed shape in an excessive way is like it is shown below:

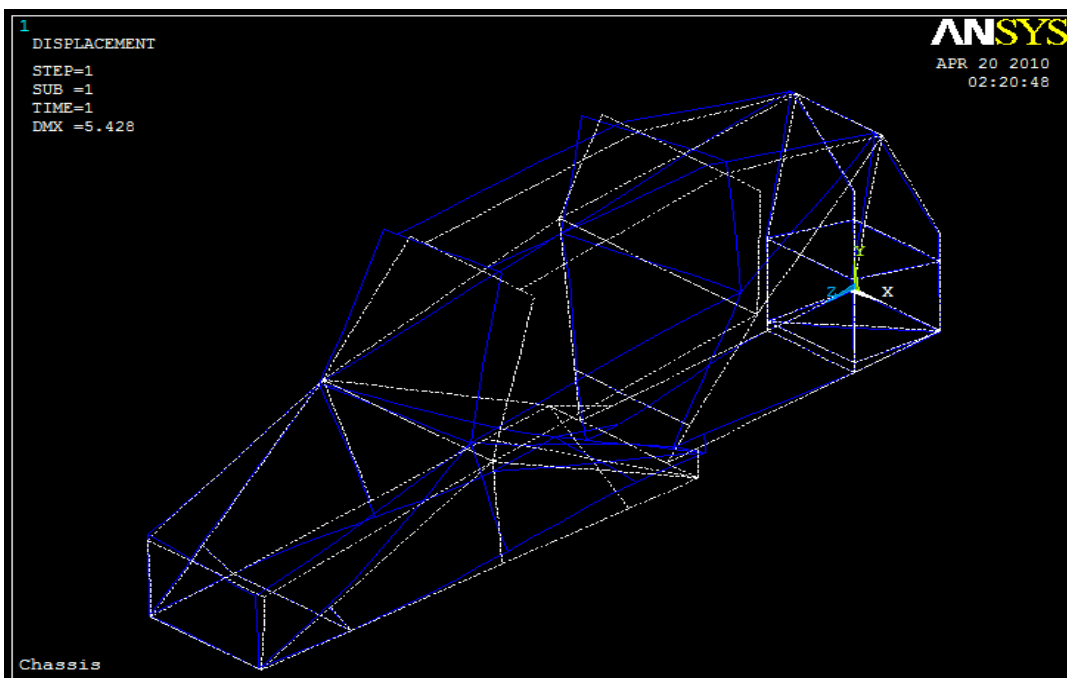


Fig. 7.28: Deformed shape when the kart is cornering

Here, it can be seen which part of the chassis suffers the most.

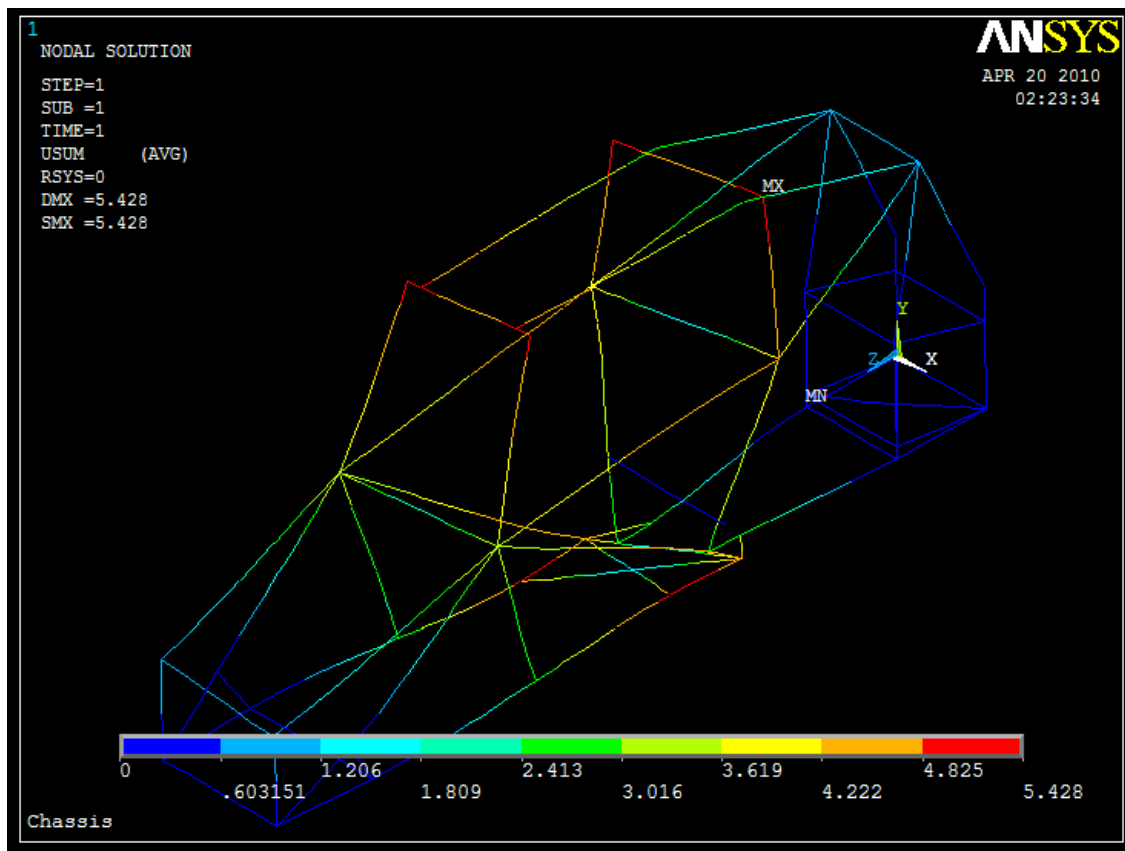


Fig. 7.29: Stress distribution when the kart is cornering

7.6. TORSION STIFNESS

Before testing that the chassis supports the loads hypothesis defined, it should be verified if the chassis is not much deformable compared in torsion.

First of all, the rear part of the chassis is fixed like it is shown in the following figure:

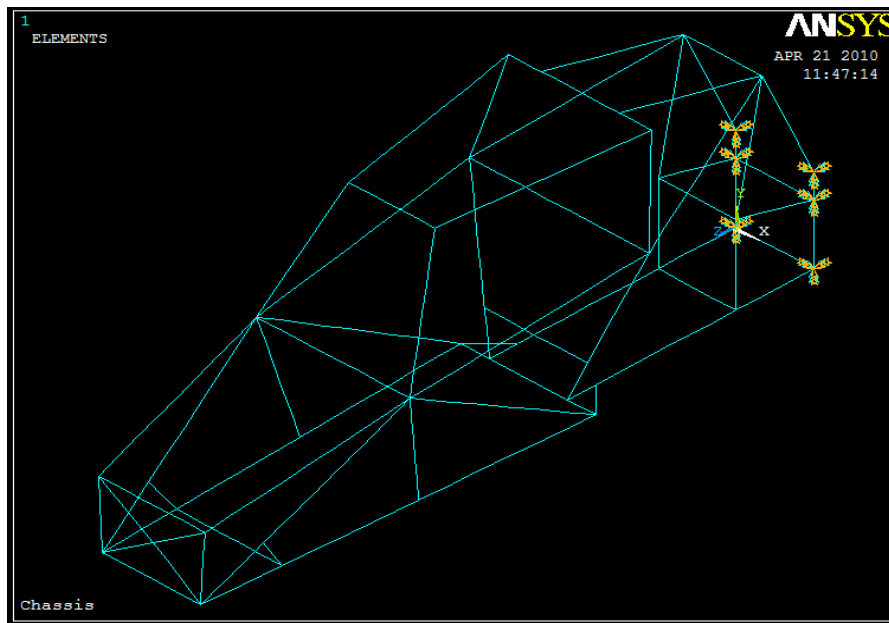


Fig. 7.30: Rear part fixed

A torque of, for example, 10 Kg·m is applied in the front part of the chassis:

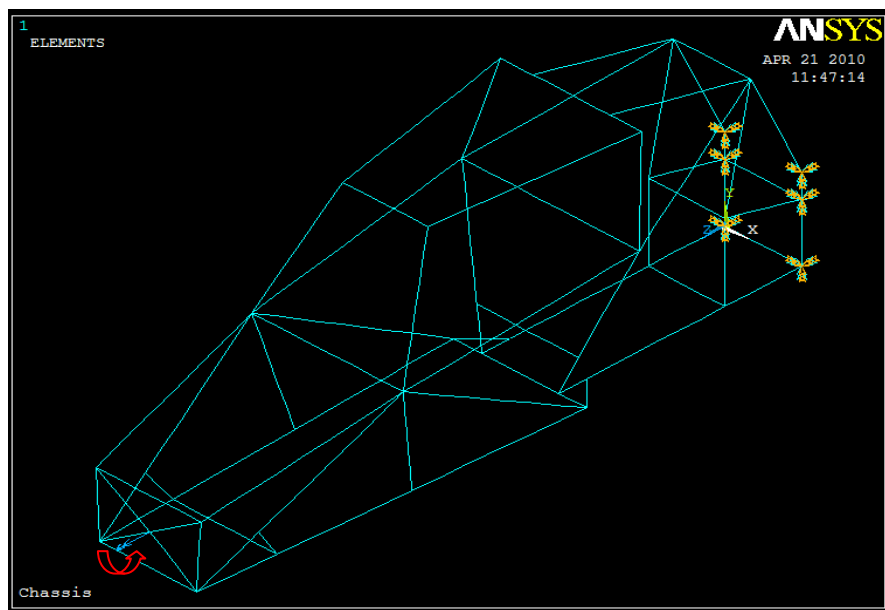


Fig. 7.31: Torque applied on the front

The angular deformation in that key point is measured. It can be seen the deformed shape in an excessive way:

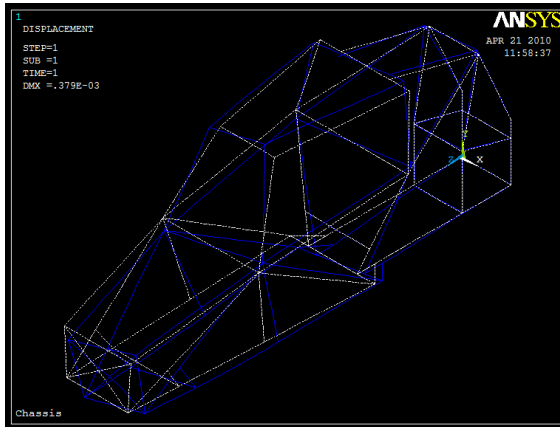


Fig. 7.32: Deformed shape

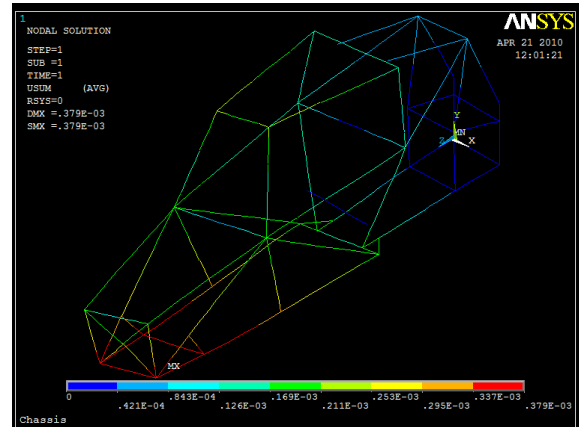


Fig. 7.33: Maximum stress

Now, the torsion stiffness can be calculated with the following equation:

$$K_{\text{torsion}} = \frac{M_t}{\text{gyration } (^\circ)}$$

In the point where was applied the torque, the gyration is: **$0.37807 \cdot 10^{-3}$ rad**

(Node 171; $0.94023E-03$ (x), $-0.76437E-03$ (y), $-0.52747E-04$ (z); **$9.7738E-04$**)*

$$9.7738 \cdot 10^{-4} \text{ rad} = \mathbf{0.056^\circ};$$

Therefore, **$K_{\text{torsion}} = 10 / 0.056 = 178 \text{ Kg} \cdot \text{m} / ^\circ$**

For a Kart Cross, the minimum value of the torsion stiffness is **$150 \text{ Kg} \cdot \text{m} / ^\circ$** (minimum recommended), so, the value calculated in this chassis is **valid**.

* (results obtained by ANSYS)

7.7. CREEP LIMIT CHECKING

To perform the checking of the creep limit in the chassis will be needed the obtaining of the stress distribution in each case, or, which are the bars that are withstanding more stress. This Von Misses stress is compared with the maximum stress that can withstand material ($f_y = 355/1.1 = 322.73 \text{ MPa}$) and, thereby, it is obtained the use of each section in the form of percentage. Then it is analyzed the results of each scenario:

$$\sigma_{VM} = \sqrt{\left(\frac{N}{A} + \frac{M \cdot \frac{D}{2}}{I}\right)^2 + 3\left(\frac{T \cdot \frac{D}{2}}{I_p}\right)^2}$$

7.7.1. Maximum acceleration:

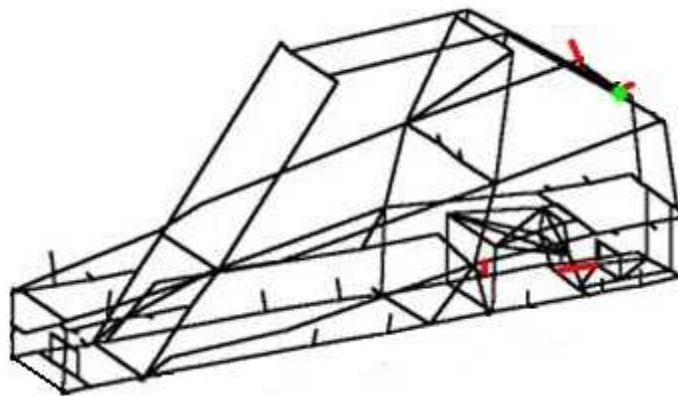


Fig. 7.34: Checking of the creep limit in acceleration

In maximum acceleration, the bars that suffer the most are the red ones. The critical section is the green point. There are 4 bars which work in a capability higher than 20%. The critical section works at 22.75% of its potential.

7.7.2. Maximum deceleration:

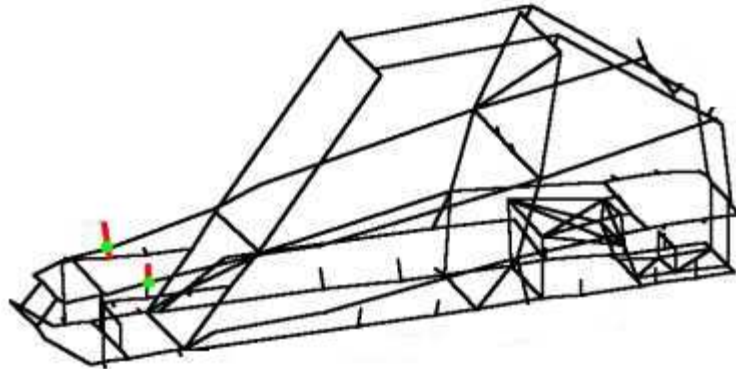


Fig. 7.35: Checking of the creep limit in deceleration

There are 3 bars which work in a capability higher than 20%. The critical section works at 24.34% of its potential.

7.7.3. Cornering:

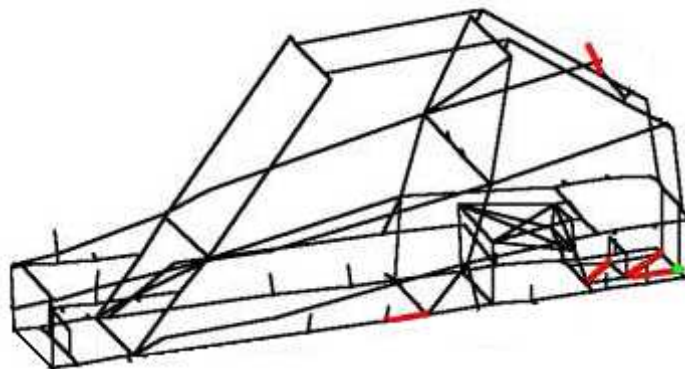


Fig. 7.36: Checking of the creep limit in cornering

There are 6 bars which work in a capability higher than 20%. The critical section works at 35.19% of its potential.

As a general conclusion, from the study of bars for the three scenarios, it has been found that the working sections are far below of the creep limit. As it has already been mentioned several times throughout the project, that is because, the chassis is designed to be rigid and not only to hold the maximum stress.

7.8. CHECKING OF THE STABILITY OF THE BARS

The bars that must be checked are those that are subject to compression. The sections are Class 3 and the verification to be made is:

$$\frac{N_{sd}}{\chi \cdot A} + K_y \cdot \frac{M_{y,sd}}{W_{el,y}} + K_z \cdot \frac{M_{z,sd}}{W_{el,z}} \leq f_y$$

It will be taken in advance the fact that the bars work with tensions far below its capacity, to simplify the calculation. It will be taken the maximum buckling coefficient, but actually this is not the case. It is taken the following values:

$$\chi = 0.0672; K_y = 1.5; K_z = 1.5$$

The result we seek is the relationship, given in %, between the maximum stress of each section of the compressed bars and yield stress of the material. These coefficients implicitly indicate what section of each bar is being applied the buckling verification formula. In this case it has been applied this formula in every section of each bar. This means that our calculation is not entirely exhaustive. It has the advantage that if all sections withstand, the bars will withstand if the calculation is made more exhaustive. Thereby, it must be taken into account that the results shown below are not results that are numerically correct. The outcomes must be regarded as qualitative. The formula will be applied in the critical section. So:

7.8.1. Maximum acceleration:

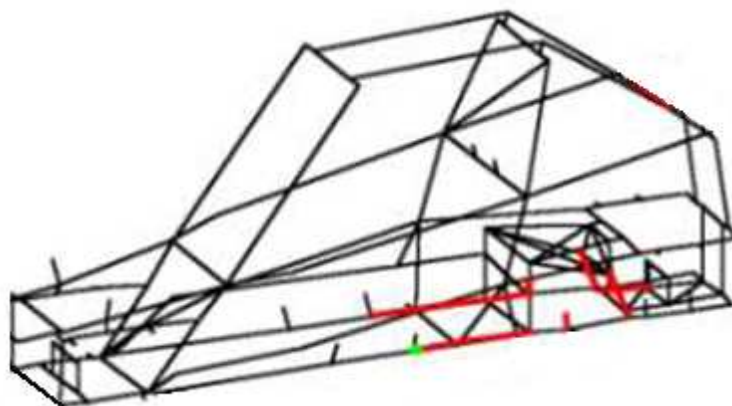


Fig. 7.37: Highest danger of buckling in acceleration

In maximum acceleration, the bars that suffer the most are the red ones. The critical section is the green point. There are 11 bars which work in a capability higher than 30%. The critical section works at 42% of its potential.

7.8.2. Maximum deceleration:

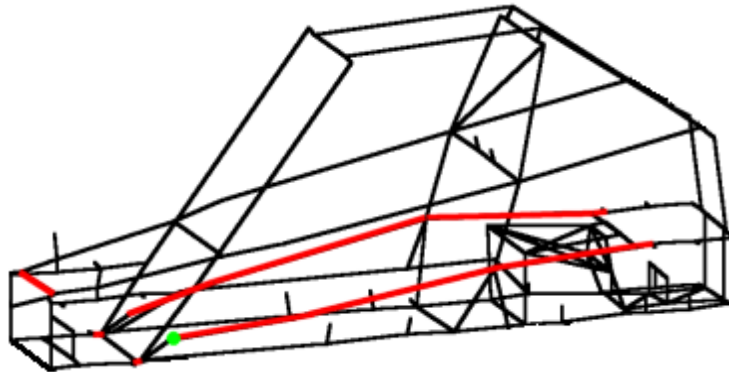


Fig. 7.38: Highest danger of buckling when braking

There are 11 bars which work in a capability higher than 30%. The critical section works at 56% of its potential.

7.8.3. Cornering:

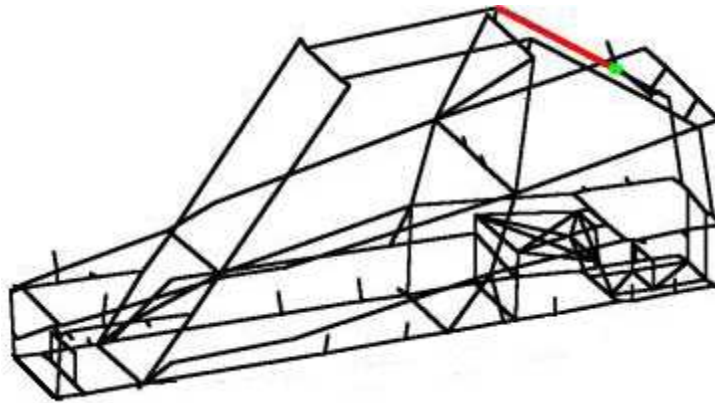


Fig. 7.39: Highest danger of buckling when cornering

There are 7 bars which work in a capability higher than 30%. The critical section works at 37.66% of its potential.

As a final conclusion of the 3 hypothesis, it can be said that the structure is far away from buckling even if it is supposed the worst cases.

8. CONCLUSIONS AND FUTURE LINES OF WORK

The first and most important conclusion is that there is no method completely automatable to design and calculate a chassis. It is needed a creative mind to make decisions regarding changes in geometry that it can give up to get the final design.

As can be seen in this report, designing this chassis, following this method, is long and tedious. Depending on the initial design, must be done a high number of iterations to achieve a satisfactory result. It is therefore very important to follow the design criteria defined in the report from the beginning.

It is also needed the realization of some previous sketches before the full design with the computer software.

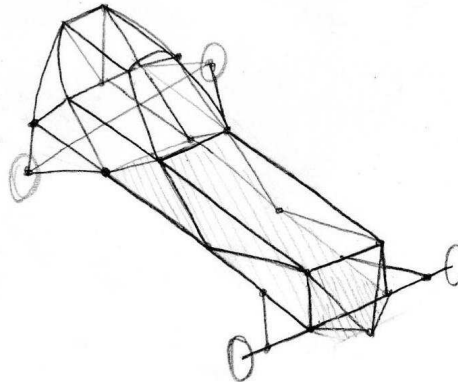


Fig. 8.1: First sketch of the chassis

As has been seen in the various sections of this calculation, chassis has torsion rigidity sufficient for its purpose and all of the bars withstand the stresses that may occur while driving. The chassis could be lighter, reducing some material until the decrease of the torsion stiffness to a value of $150 \text{ kg}\cdot\text{m}^0$, which is the recommended minimum stiffness for Kart cross.

The last characteristic mentioned in the sections of aims and objectives that the chassis should have, is the cost. The first choice of the material was carbon fiber (most expensive); this material would be the best choice if our main objective would have been the weight of the chassis, but, the Kart Cross rules (Appendix A) says that the material should be metallic and the percentage of carbon must be lower than 0.22 %. Therefore, the steel S355 is the best option to design the chassis. In fact, that material is the recommended by the Kart Cross competition.

As it was said before, when the loads were applied in the chassis, the values of the acceleration in the 3 cases (maximum acceleration, maximum deceleration and cornering), were taken of a complete collection of data from several karts in a Kart Cross circuit. The values were raised in order to done the calculation safer.

	Measured in the circuit	Used in the analysis
Maximum acceleration	2.15G	2.5G
Maximum deceleration	-1.35G	-1.5G
Cornering	0.9G	1G

Table 8.1: Values taken in the calculation

It is also noted that the position in which the man drives is not very comfortable. He should have the arms and legs too bent, which will be a handicap for the ability to drive.

As a final conclusion, it must be said that this project is a method which can lead to tubular chassis engineering, although not the ideal method. The ideal is not feasible now because CAD technology has not reached the point where it can be drawn the tubular structures with the interested flexibility.

9. FUTURE LINES OF WORK

The chassis has been designed taking into account only the considered static loads. When driving, it is obvious that, in the vehicle, appear vibrations due to the efforts from the field have different distribution. This will make that the chassis can fail to fatigue.

The design of the kart has been done taking into account only the weight of the pilot, the weight of the engine, and every loads caused by the weight of the chassis itself. A complete would be the one which takes into account the suspensions, transmission, steering, etc... Moreover, a kart is exposed to collisions that are more common than we would like to, and usually, they have serious consequences. According to the motorsport magazines, the chassis should absorb these efforts. To absorb these efforts, the chassis should be deformed. This requires that certain areas of the chassis should have relatively small stiffness. At this point, it is reached a contradiction with what has been seen in this project, so, a balance between rigidity and flexibility on the different areas of the chassis should be done. Another point that should have been studied is the determination of the applicable load assumptions due to aerodynamic forces.

As within all procedures there are some steps that require more time than others. This applies to the verification of the efficiency of joints. The joints have to be checked one by one, so it's worth keeping in mind that when designing the chassis number of knots should be as small as possible.

It is an iterative method, and depending on how close is the initial design and the final solution, sooner or later will be finished. It is advisable to make a good initial estimate. But there are no tricks to it; the more experience the better projecting the initial design.

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Eurocode 3

APPENDIX A: KART CROSS TECHNICAL REGULATION

APPENDIX A: KART CROSS TECHNICAL REGULATION

A. - For the year 2010, the latest engine version admitted will be MODEL 2009.

B. - The contestant must be in possession of the original shop manual motor manufacturer.

Art. 1. Engine.

The engine must be strictly standard (of origin), its modification is not allowed and it will obey the following conditions:

Art. 1.1. The lightweight, grinding, machining, polishing, balanced or any kind of physical treatment, chemical or any mechanical engine components is forbidden. It is also forbidden the adding or removing of material to any engine part.

It is only allowed the following:

Art. 1.2. It is allowed the elimination of the anti-pollution devices of the engine.

Art. 1.3. The engine must have a launch system with the means available on board the vehicle. It is expressly forbidden the use of external auxiliary batteries.

Art. 1.4. The exhaust is free, provided it meets the following:

The exhaust outlet shall be at the rear of the vehicle and be fitted to a maximum of 80 cm. and a minimum of 10cm. above the floor. It should be avoided that the exhaust gases impair the driver of the vehicle behind. None of the exhaust elements will overcome the perimeter of the chassis frame or body viewed from above.

It is mandatory that the exhaust meets the noise limit established in Article 8.1.2 of this Regulation.

Art. 1.5. Radiators water shall not overcome beyond the perimeter of the chassis or the body and shall be located behind the driver's compartment, without being in contact.

Art. 1.6. It is authorized the installation of an oil radiator if the engine does not have it or substitute another when applicable.

It must be respected all rules of this radiators and pipes regulation.

Art. 1.7. The vehicle must be equipped with a battery firmly secured and protected. If is close to the carrier must be covered with insulation and protection watertight.

Art. 1.8. The Cutting systems ON (Cut-off and / or similar) to change automatic running are allowed.

Art. 1.9. It is allowed the installation of an electric fuel pump to replace the original of the vehicle while respecting all the rules. The fuel pressure regulator original can be replaced by a manually adjustable.

Art. 1.10. It is allowed external spraying water on the radiators, where the sole aim of that spraying is the cooling. The device should not be placed outside the perimeter of the chassis, it will not exceed 2 liters and it must obey the Art.10.1 of this regulation.

Art. 1.11. In injection engines must be taken into account:

Art. 1.11.1. That all motor peripherals (sensors, alternator, starter, intake housing, trumpets, etc) must be strictly standard (of origin) and no modification is allowed, every element must work properly and is not allowed their deactivation.

Art 1.11.2. It can be only removed the wires, sensors and actuators that do not submit information to the switchboard, and have no original electrical function (headlights, turn signals, horn, switch, kickstand sensor, etc.). It can be replaced the original boot switch-on by other, and the key contact by the circuit breaker.

Art. 1.11.3. The exterior of the unit and all connectors should be standard, maintaining their original function and being interchangeable with the original switchboard.

Art. 1.12. The carburetor engines must take into account:

Art. 1.12.1. It should be kept the initial fuel system of the engine, being free only their settings (gum, needles and springs). The air intake is free choice.

Art. 1.12.2. The ignition system, switchboard, plugs, wiring, etc..., Is free choice.

Art. 1.12.3. The alternator can be removed.

Art. 2. Transmission.

Art. 2.1. The gearbox and clutch will be the initial ones of the engine without modification. It is allowed the change of the clutch drive system, mechanical by hydraulic or other vice versa.

Art. 2.2. The final drive ratio (pinion-crown) is free choice. It is recommended to mount an effective chain guard.

Art. 2.3. The use of differentials is forbidden.

Art. 3. Chassis and bodywork.

Art. 3.1. All vehicles built after 01/01/2010, must have their structure designed to allow attachment of the harness belts according to the specifications required for the use of the

HANS by the pilot. In addition, all vehicles built after 01/01/2003 and all vehicles equipped with fuel injection engine must have the approval of the RFEA. All chassis must have a plate welded on a visible place of it, clearly identified, which reflected the following:

- Manufacturer.
- Date of manufacture.
- Serial number.
- Number or reference package.

The contestant must be in possession of the necessary documentation attesting to the approval by the RFEA.

Art. 3.2. The maximum dimensions of vehicles, including bodywork, will be the following:

- Length: 2.600mm.
- Width: 1.600mm.
- Height: 1.400mm.

Art. 3.3. The chassis of the vehicle consists of a tubular structure which supports the mechanical elements and provides the necessary protection to the pilot in case of rollover or accident. The chassis must meet the following specifications:

- It will be made in carbon steel pipe, unalloyed, cold drawn, seamless, with a maximum carbon content of 0.22%. The minimum diameter of the tube is 30mm. and minimum thickness of 2mm.
- There should be a roll bar and a central striker joined at the top. The diameter of these arcs will be 40mm. and minimum thickness of 2mm.
- The line drawn between the top of the central arches and front must overcome, in at least 5cm. the pilot's helmet.
- The distance between the front end of the frame and the pedals will be 10cm.
- It should be installed two longitudinal straps attached to the central arch and moving back as much as possible.
- The floor or ground of the vehicle should be rigid from the front and at least until the vertical line passing through the central arch; it has to be aluminum with a maximum thickness of 4mm, or steel sheet with a maximum thickness of 2 mm.
- No part of the vehicle must present sharp edges or sharp edges.

- It is not allowed in any reinforcements, shields and other items, whose sole purpose is not the ones described in the preceding paragraphs.

- The chassis should be deformable in order to absorb the possible collisions.

Art. 3.4. The bodywork must be rigid, hard and opaque material, and it shall be securely fastened to the chassis. It must obey the following specifications:

- It is not allowed any sharp edge. The angles and corners should be rounded with a radius of 15mm. or greater.

- In the front and sides must be installed a bodywork that provides protection against stones. The height of this bodywork shall be a minimum of 42cm. measured relative to the plane passing through the driver's seat bracket.

- Viewed from above all of the mechanical components of the vehicle, required for propulsion, (engine, transmission, etc...) must be covered by the bodywork.

- The panels used shall be no thicker than 10mm.

- The Installation of a hard top and opaque on the pilot is required.

Art. 4. Side impact protection.

It will consist of a steel tube structure with a minimum size of 30mm. x 2mm. attached to the chassis on both sides of the vehicle, at the level of the axis of the wheels, and occupying at least 60% of the battle. These structures will be extended to a maximum of up to a straight line between the outer surfaces of the wheels.

Art. 5. Weight.

At no time during the test, the minimum weight of the vehicle may be less than 310 kg without driver, without fuel, without water in the sprayer and in running order, the use of drags is prohibited.

Art. 6. Suspensions.

Art. 6.1. The axles are suspended, not being allowed the assembly of rigid caps between the axles and chassis anywhere.

Art. 6.2. The number and type of shock absorbers and springs are free choice. It is not allowed the hydraulically connection with each other shock absorbers.

Art. 7. Steering.

Art. 7.1. The steering system is free choice, acting only on the front wheels. Systems by chains, cables or hydraulics are prohibited.

Art. 7.2. It is necessary to use a system extract of the steering wheel of the SPA type.

Art. 8. Fuel tanks, oil and cooling water.

Art. 8.1. All deposits must be located behind the pilot and isolated from the driver's compartment through panels, so that, in the case of leakage or rupture of the shell, the liquid cannot pass this compartment. This also applies to fuel tanks, with respect to compartment engine and exhaust system. The fuel filler neck must be tight and must not overcome the bodywork. The cooling water tank must have a stopper equipped with an overpressure valve.

Art. 8.2. The fuel tank has to be metallic and must be located behind the seat. It must be mounted in a position sufficiently protected and firmly anchored to the vehicle. It cannot be in the driver's compartment and must be separated from it by a fire resistant panel. The fuel tank must be installed at least 30cm. from the cylinder head and the exhaust system, unless it is separated from these by a waterproof and fireproof panel.

Art. 8.3. The gas pipes must be adequately protected against the fire.

Art. 8.4. The maximum capacity of the fuel tank is 10 liters.

Art. 9. Wheels and tires.

Art. 9.1. The maximum diameter of the rim is 10 "and maximum width is 8".

Art. 9.2. It is authorized knobby tires manufactured specifically for these vehicles.

Art. 9.3. It can be installed in the rear wheel hub caps the same material as the skirt, attached to the wheel tightly and secure with screws.

APPENDIX B: CALCULATION WITH ANSYS

APPENDIX B: CALCULATION WITH ANSYS

INTRODUCTION

ANSYS 12 delivers innovative, dramatic simulation technology advances in every major physics discipline, along with improvements in computing speed and enhancements to enabling technologies such as geometry handling, meshing and post-processing. These advancements alone represent a major step forward on the path forward in Simulation Driven Product Development. But ANSYS has reached even further by delivering all this technology in an innovative simulation framework, its latest-generation ANSYS Workbench.

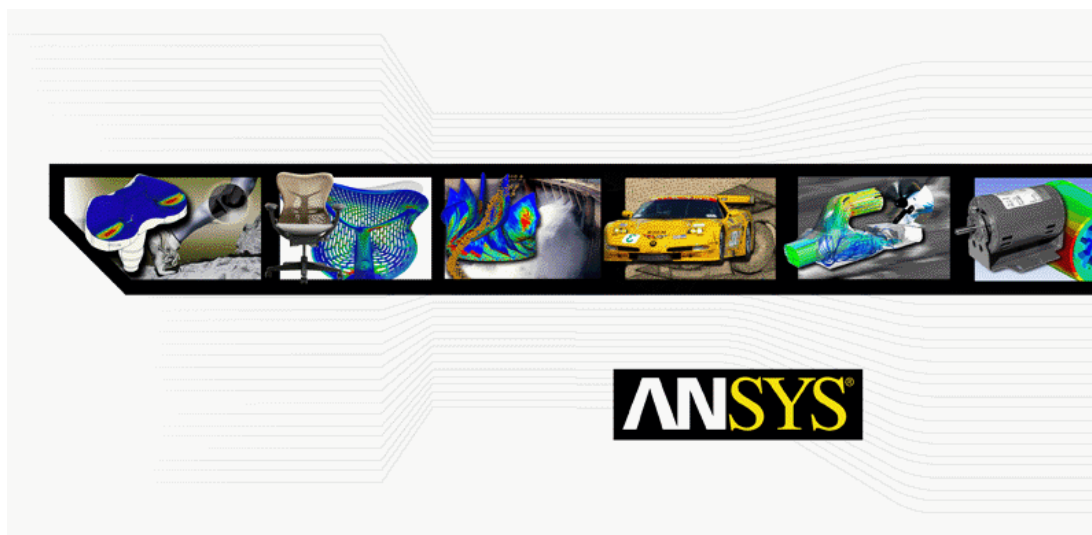


Fig. B.1: Ansys front page.

HOW DOES IT WORK?

In this project, the design has been done with the 3 principal parts of the software:

- Preprocessor
- Solve
- General Postproc

The first part to begin the design is the preprocessor. Here is where the drawing is done.

In this case, the first thing done was the determination of every keypoint of the chassis.

Those points are:

1	0.00	0.00	0.00
2	0.00	0.00	30.0
3	30.0	0.00	0.00
4	30.0	0.00	30.0
5	0.00	25.0	0.00
6	0.00	33.0	30.0
7	30.0	25.0	0.00
8	30.0	33.0	30.0
9	0.00	35.0	0.00
10	30.0	35.0	0.00
11	0.00	80.0	20.0
12	30.0	80.0	20.0
13	0.00	0.00	95.0
14	30.0	0.00	95.0
15	-19.5	57.0	83.0
16	49.5	57.0	83.0
17	-12.5	94.0	75.0
18	42.5	94.0	75.0
19	-11.0	0.00	95.0
20	41.0	0.00	95.0
21	0.00	94.0	75.0
22	30.0	94.0	75.0
23	0.00	0.00	84.0
24	30.0	0.00	84.0
25	-7.86	0.00	160.
26	37.9	0.00	160.
27	-5.45	0.00	210.
28	35.4	0.00	210.
29	-4.00	0.00	240.
30	34.0	0.00	240.
31	-14.5	45.2	170.
32	44.5	45.2	170.

33	-5.45	27.0	240.
34	35.4	27.0	240.
35	37.1	13.6	219.
36	-1.55	89.3	148.
37	31.6	89.3	148.
38	-7.15	13.6	219.
39	-4.87	14.2	92.0
40	-19.5	57.0	-83.0
41	49.5	57.0	-83.0
42	-14.6	42.8	86.0
43	-14.6	42.8	86.0
44	34.9	14.2	92.0
50	15.0	39.1	105.

Table B.1: list of keypoints

With these keypoints (measures in cm) can be drawn the straight lines that make the chassis up:

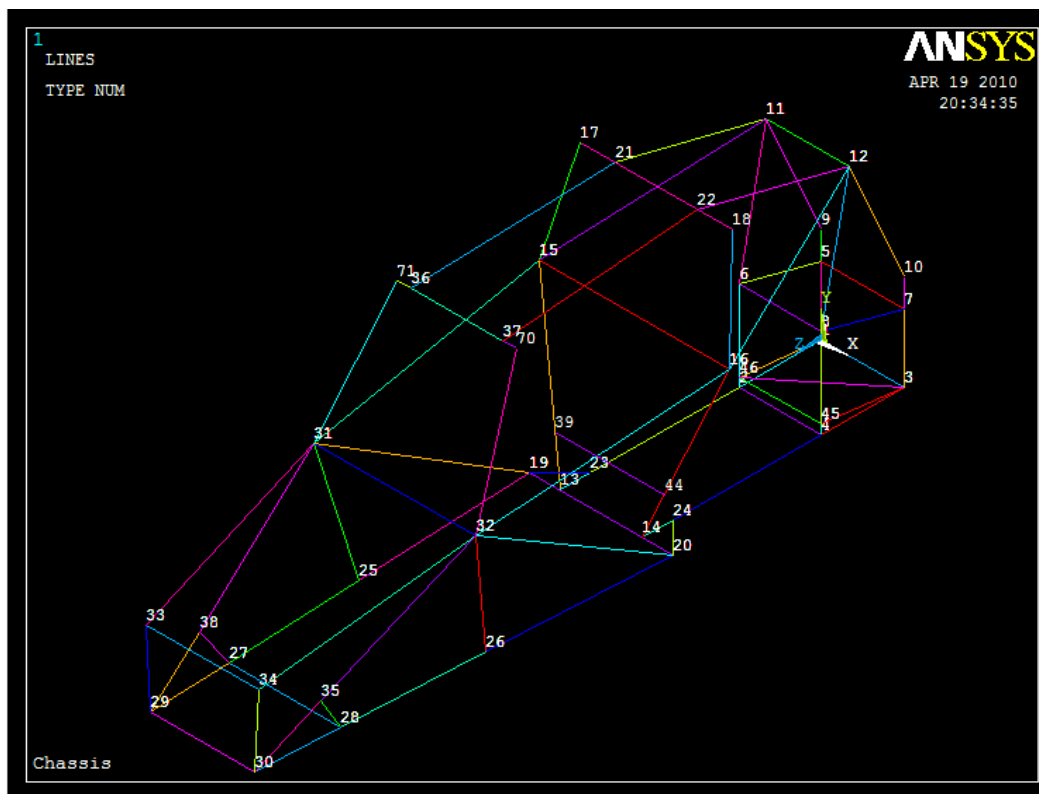


Fig. B.2: Design of the chassis in ANSYS

And their 3 different views are:

- Elevation:

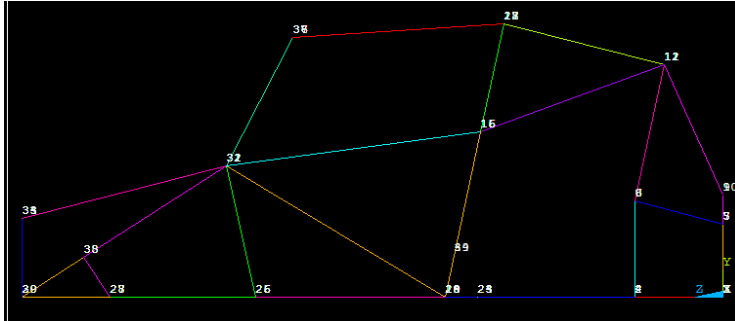


Fig. B.3: Elevation view

- Profile:

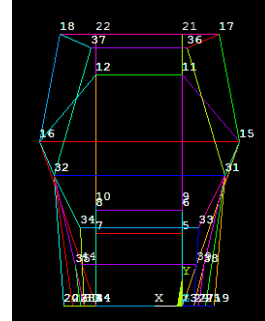


Fig. B.4: Profile view

- Floor plan:

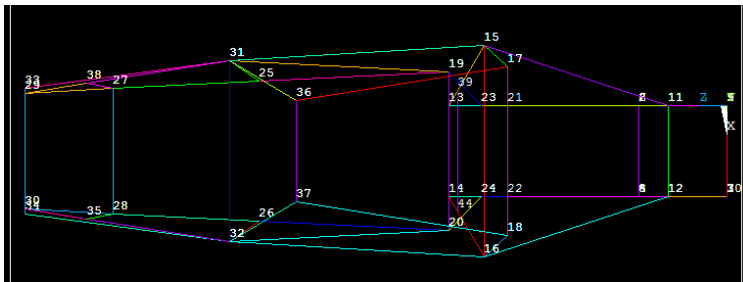


Fig. B.5: Floor plan view

The dimensions of the chassis are in cm:

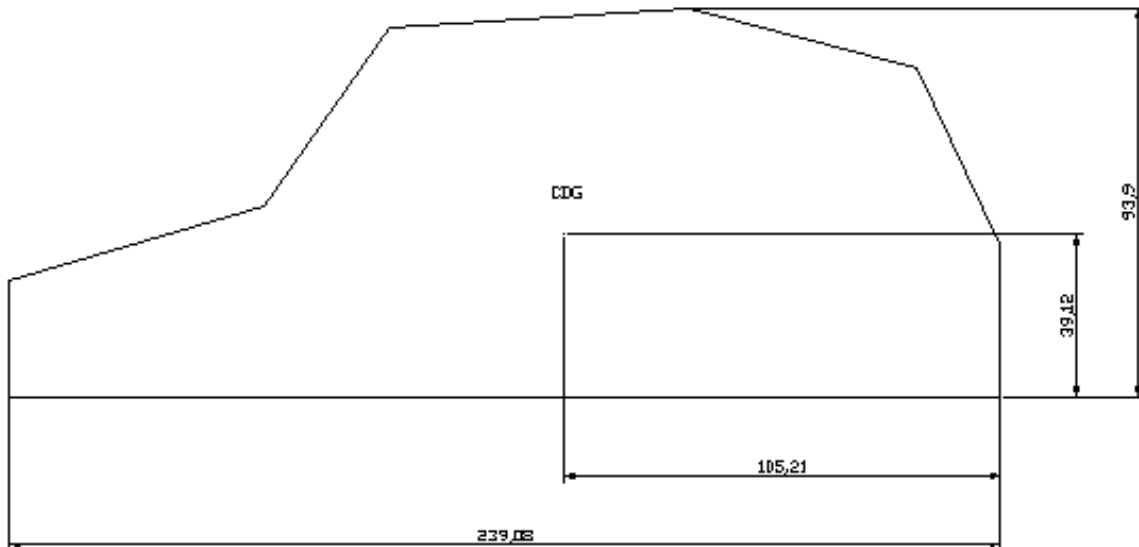


Fig. B.6: Center of gravity and dimensions

The next step is the realization of the meshing. In ANSYS, before doing the meshing of the structure, it must be selected the size of each elements of the mesh. In this case, the size was 5 cm. It must be chosen the type of material, its properties, and the diameter of the tube.

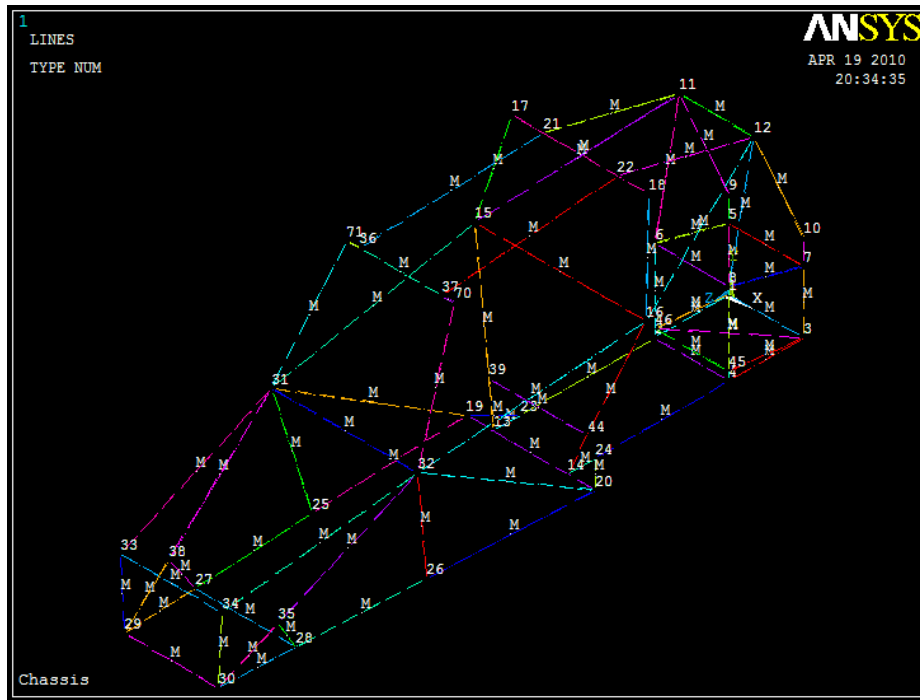


Fig. B.7: Size control of the mesh.

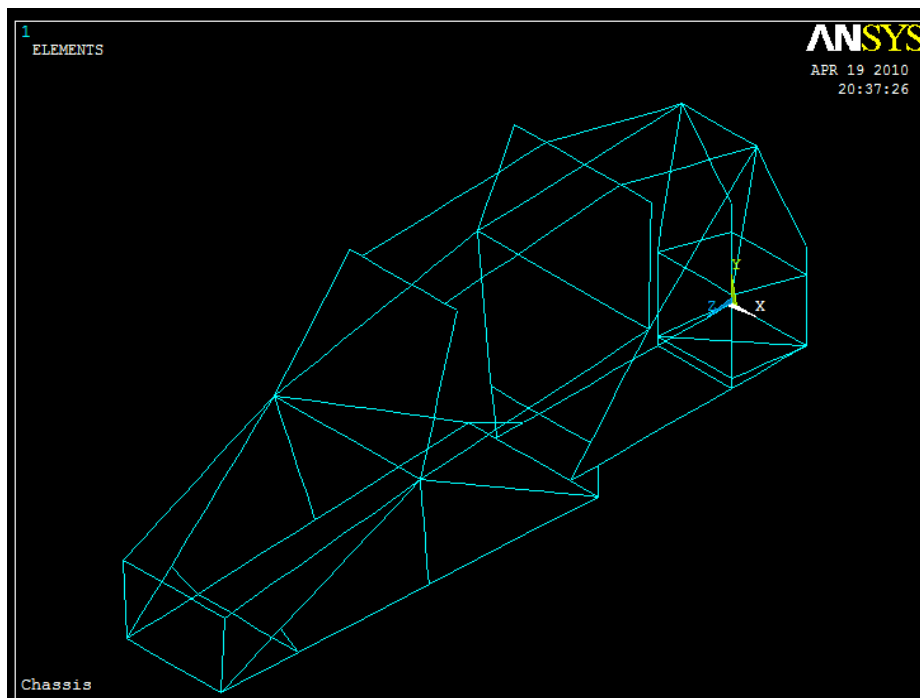


Fig. B.8: Mesh completed

Once the structure is mesh, it is the moment of the second part of the design: the 'solve' part.

In this part, ANSYS is the responsible of the application of every load in the key points, lines, etc...

It will be different in each different case of application:

- Maximum acceleration:

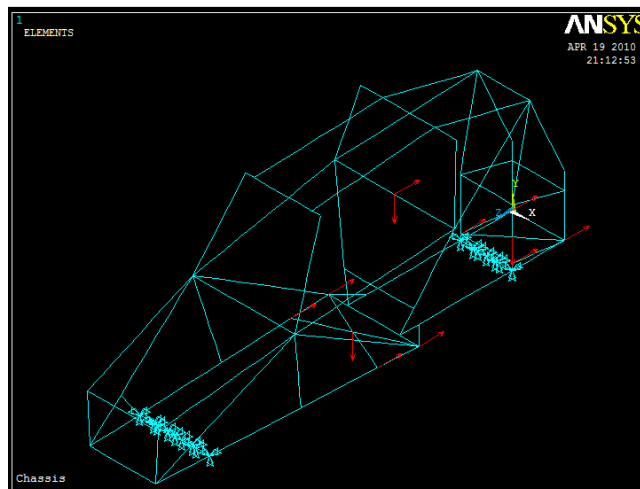


Fig. B.9: Load balance in maximum acceleration

- Maximum deceleration:

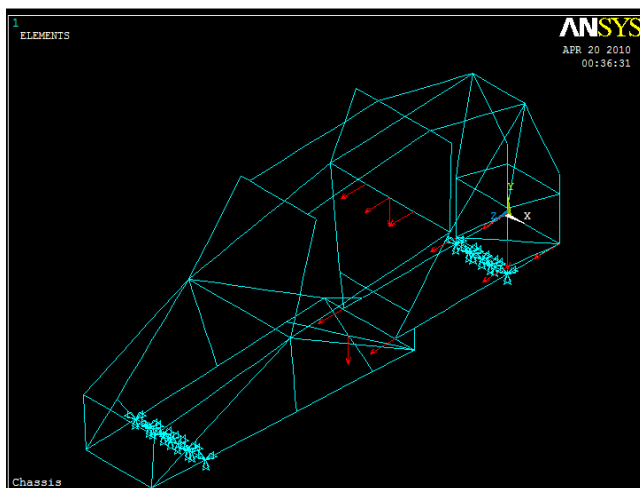


Fig. B.10: load balance in maximum deceleration

- Cornering:

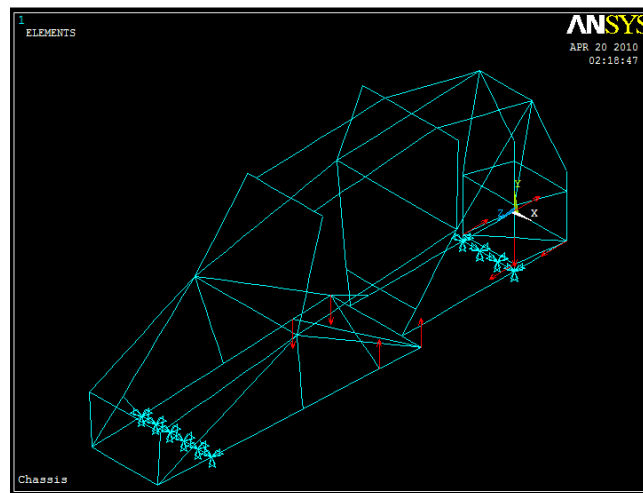


Fig. B.11: Load balance in cornering

And, finally, the last part in the design is showing the results, the ‘General PostProc’. They can be given as a list of results, or as a plot results (in the report, in the section of calculus, are given as a plot results. See section 7).

ANSYS can show different solutions:

- Nodal solution
- Stress solution (main stresses, Von Mises stress, etc...)
- Deformed shape
- ...

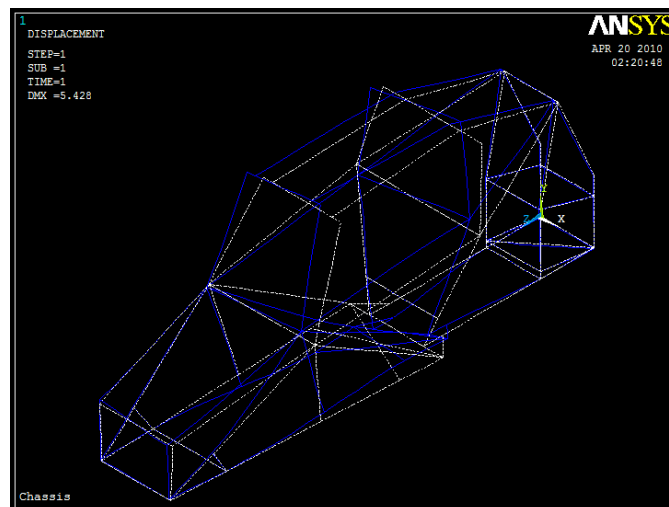
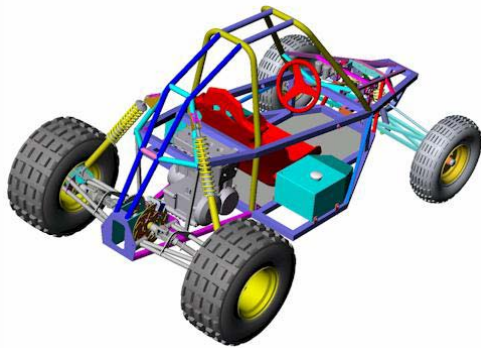


Fig. B.12: Deformed shape when cornering

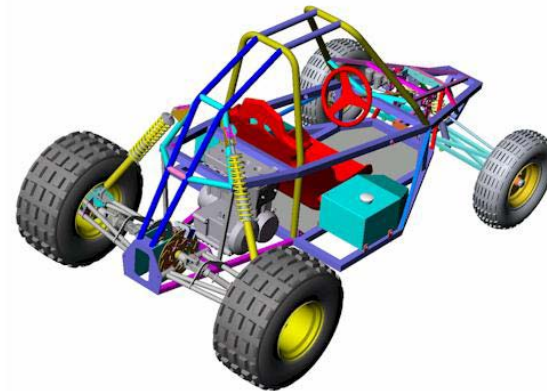
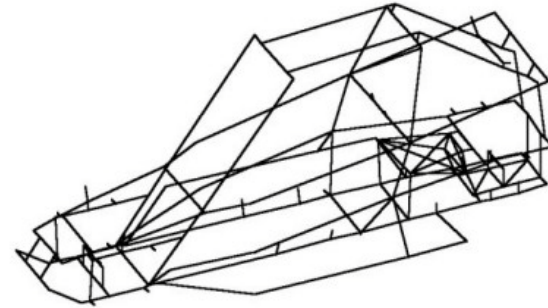
Design of a tubular chassis for Kartcross



Mikel Villafanca

Introducción

- ▶ What is?
- ▶ Functions:
 - Supports the kart
 - Rigid connection
 - Necessary stiffness
 - Pilot protection



Aims & objectives

▶ **Aim of the project:**

- Design the best solution of a kart's chassis with ANSYS.

▶ **Objectives:**

- Static calculation
- Dynamic calculation
- Creep limit
- Torsion
- Buckling

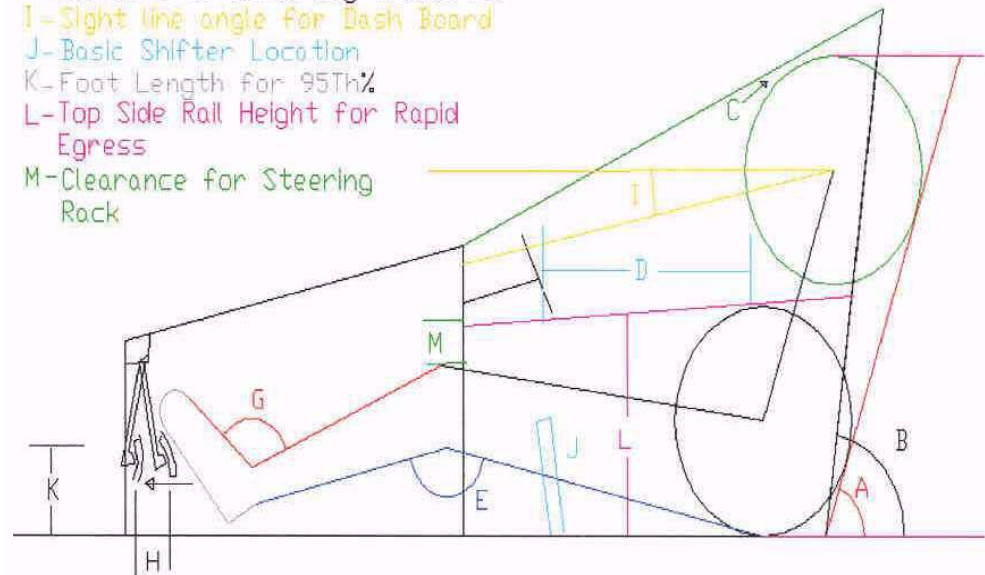


Design criteria

- ▶ Stiffness
- ▶ Weight
- ▶ Space
- ▶ Cost
- ▶ Applied loads

- A- Tilt angle of Seat
- B- Tilt angle of Roll Hoop
- C- 2" Minimum Clearance
- D- Chest to Steering Wheel Distance
- E- Leg angle
- F- Tail bone to Head Height for 95th%
- G- Ankle angle
- H- Distance of Ankle/Leg Travel for
- I- Sight line angle for Dash Board
- J- Basic Shifter Location
- K- Foot Length for 95Th%
- L- Top Side Rail Height for Rapid Egress
- M- Clearance for Steering Rack

Ergonomics
Layout



Materials

- ▶ Different possible choices
 - Carbon fiber
 - Carbon alloy
 - **Steel S355 (CHS : Circular Section Tubes)**
- ▶ Steel properties:
 - $f_y = 335 \text{ N/mm}^2 \text{ (MPa)}$
 - $G = 81,000 \text{ N/mm}^2 \text{ (MPa)}$
 - $\rho = 7850 \text{ Kg/m}^3$
 - $E = 210,000 \text{ N/mm}^2 \text{ (MPa)}$
 - $\nu = 0.33$
 - $\alpha = 12 \times 10^{-6} / \text{C}$

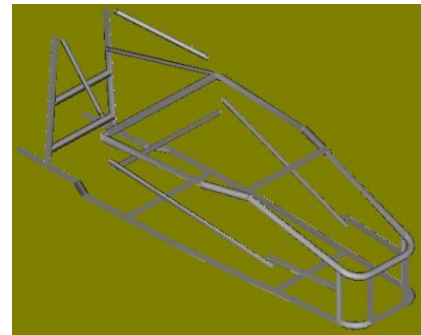
Chassis parameters

▶ General dimensions:

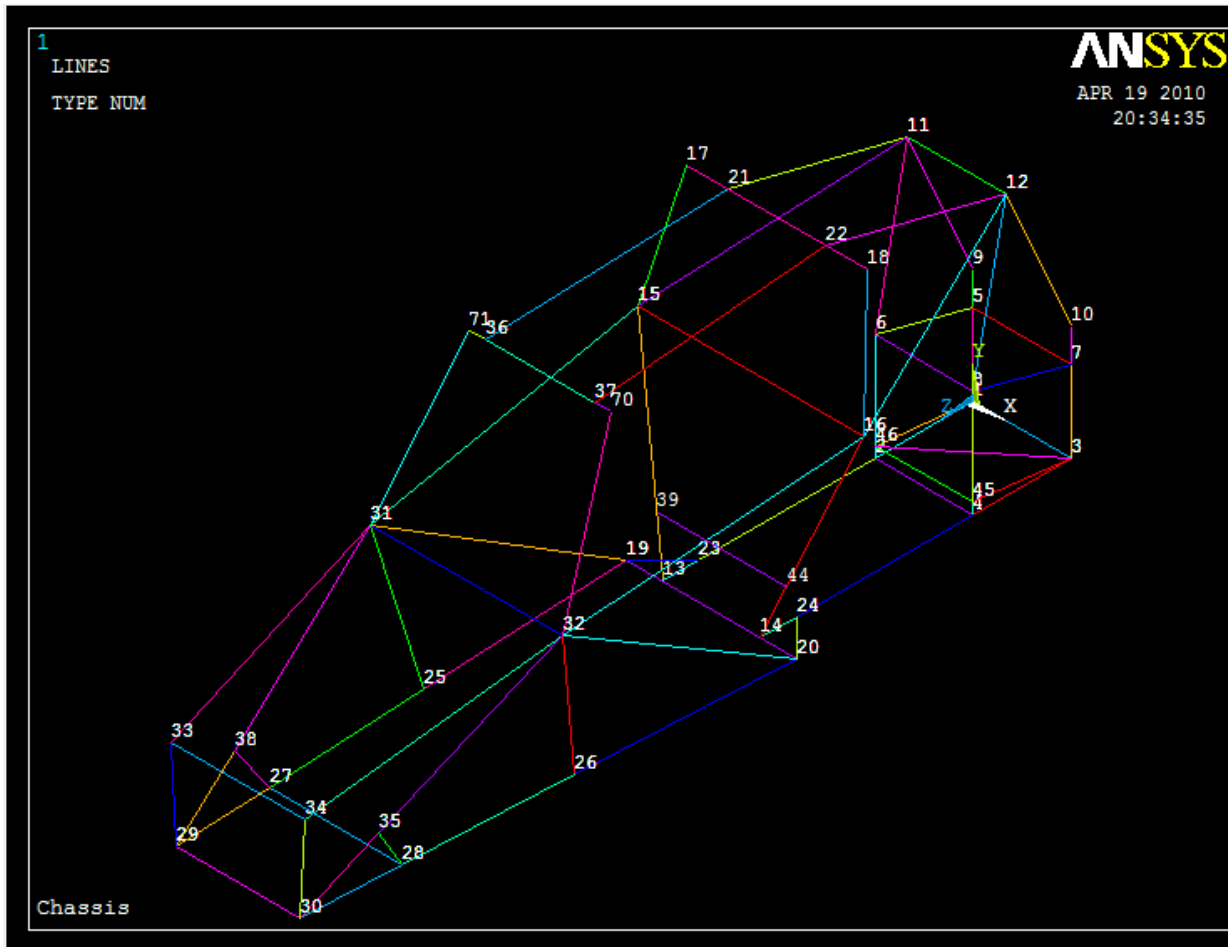
- Length = 2600 mm High = 1400 mm
- Width = 1600 mm Weight = 310 Kg
- Wheelbase = 1850 mm

▶ Dimensions of the tubes:

- 40x2 mm central and front arches
- 30x2 mm rest

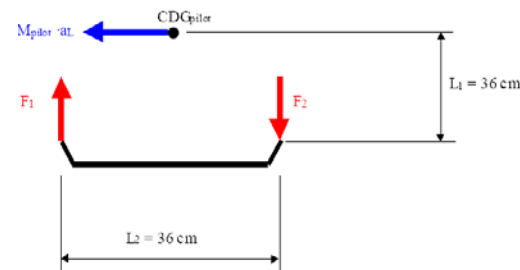
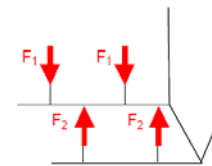
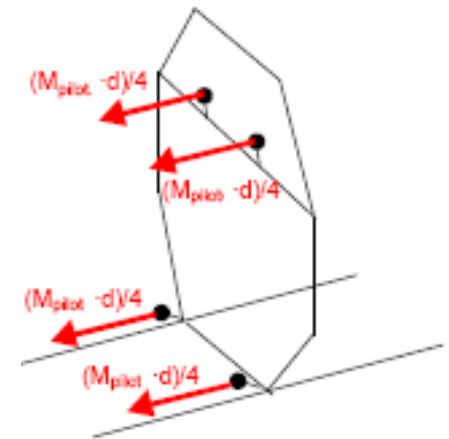
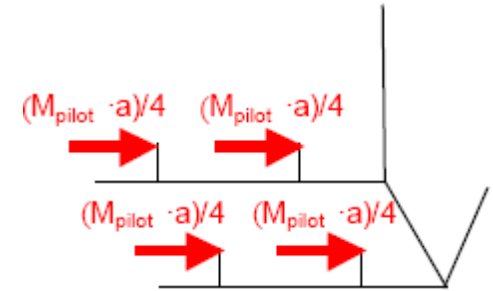


Final design



Load hypothesis

- ▶ Inertia of the pilot on the seat's brackets
- ▶ Inertia of the engine on the engine brackets
- ▶ Inertia of the chassis itself
- ▶ Weight of the chassis itself
- ▶ Weight of the engine
- ▶ Weight of the pilot



Results

▶ Creep test in the bars:

- 23% (max acceleration)
- 24% (braking)
- 35% (cornering)

$$\sigma_{VM} = \sqrt{\left(\frac{N}{A} + \frac{M \cdot \frac{D}{2}}{I}\right)^2 + 3\left(\frac{T \cdot \frac{D}{2}}{I_p}\right)^2} \leq f_y = 322.73 \text{ MPa (100\%)}$$

▶ Checking of the stability of the bars (buckling)

- 42%
- 56%
- 37%

$$\frac{N_{sd}}{\chi \cdot A} + K_y \cdot \frac{M_{y,sd}}{W_{el,y}} + K_z \cdot \frac{M_{z,sd}}{W_{el,z}} \leq f_y \text{ (100\%)}$$

▶ Torsion analysis

- Gyration = 0.056°
- Torque (M_t) = 10kg·m

$$K_{torsion} = \frac{M_t}{\text{gyration (}^\circ\text{)}} = 178 \text{ Kg} \cdot \text{m/}^\circ \geq 150 \text{ Kg} \cdot \text{m/}^\circ$$

Conclusions

- ▶ Objectives achieved
- ▶ The results are correct
- ▶ Removing some material, could be got a lighter chassis.

Thank you