POLITECNICO DI TORINO

Master of Science in Automotive Engineering

Panda 4WD - Hybrid: design and analysis of a new suspension system for a hybrid vehicle for rally competitions

Passione motoristica italiana dal 1986

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1. INTRODUCTION

This thesis work is a part of a bigger project which involves a team of students from the Politecnico di Torino coordinated by professor Stefano Carabelli. All of them work for the same target: to build a functional off-road hybrid vehicle having a Fiat Panda 1°generation as a base to start with.

1.1. Objective

The target of the whole project is to participate to the Panda Raid, an amateur rally competition in which only some kind of old school vehicles are allowed. The idea proposed by professor Carabelli is to participate with a hybrid vehicle to introduce a new concept at the competition.

The objective of this thesis is to design a new rear suspension system for the Panda vehicle to be able to implement all the changes required for the hybridization process, which consists on the installation of an electric motor in the rear part of the car to move the rear wheels.

All the work done for the project, and in particular for this thesis, was possible thanks to the company Italtecnica S.R.L. with which the polytechnic works on many projects. The advices and the kindness of the owner Mr. Mario Cavagnero, and of his son Carlo Cavagnero, were fundamental for the development of the ideas, as well as for the technical support.

1.2. Work steps

The first step is the choice of the starting platform to work on, among the two possibilities available: 2WD and 4WD. The choice will fall on the one which can better accommodate the rear electric motor together with the kinematic chain to the wheels. The study will be structured as a comparative trade-off in which all the possible layouts will be shown, pointing out all the pros and cons of each one.

The following step is the creation of the new suspension components using the CAD software Solidworks. The criterion used in this phase is to maintain unchanged most of the remarkable points of the suspension, to preserve the very good comfort and stability of the original platform and to keep as many original components as possible, for a cheaper solution.

The presence of the electric motor, and in particular of the drive shafts, will require a substantial modification of the attachment of the suspension bridge (the principal component of the suspension system) to the wheel hub. The introduction of a structural linking flange between the bridge and the hub will become necessary.

With these constraints and the experience of Italtecnica the final layout will be soon decided, also considering the manufacturability and the cost of the new parts, as well as the available space for the components.

Once the final layout is decided, an elasto-kinematic model will be created using the software Adams View, with two purposes. First, since the ground clearance has to be increased by few centimeters a check of the suspension geometry is needed. Second, a determination of the loads on the new components under the most critical conditions will be fundamental for the design. In particular, a model of the rear suspension system will be created starting from the kinematic data and specifying the relative motion between the components. It will be inserted also a model of the frame of the vehicle with its own center of mass and polar moments of inertia, including also concentrated masses simulating the real parts of the car. Then the correct material will be added to each component. Once created the model of the whole car, a specifically designed CAD model of a typical off-road path will be inserted, giving the inputs for the movement of the vehicle system. The loads on the suspension critical components will be given as outputs by the software.

The next step will be the structural analysis of the new components on the basis of the loads previously found, using FEM analysis software. In particular, the study will be concentrated on the new and most stressed piece: the flange connecting the suspension tube to the wheel hub. In this case the inputs will be the constraints on the flange at the highest loads coming from the previous analysis.

As last step, after the final shape of the flange will be decided, the 2D drawings will be created for the manufacturing process. All the phases are summed up in Figure 1.

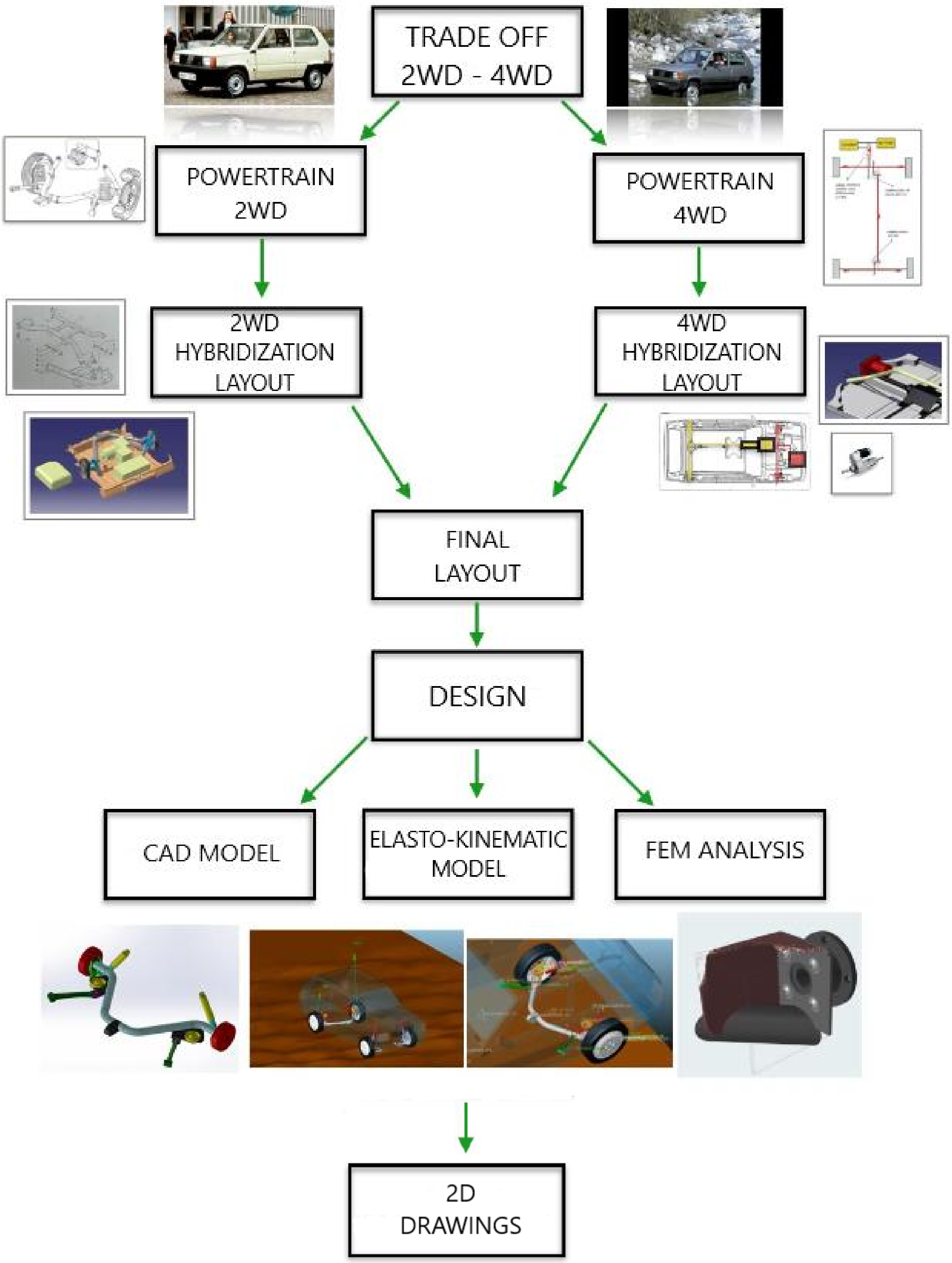


Figure 1: Work phases

1.3. Panda Raid competition

Figure 2: Panda Raid [1]

Panda Raid is an amateur competition which takes place every year in March. There are more than one hundred teams challenging each other during this long run aboard an old school Fiat Panda.

The race is divided into seven long stages through Morocco and the stress level on the crews and on the vehicles are very high, testing the capabilities of the teams to manage the difficulties throughout the journey. To have an idea of how difficult it is, just know that to win it is sufficient to arrive to the end and the speed is not important.

Moreover, in a world more and more technologically advanced, where it is impossible to get lost thanks to GPS and Internet, Panda Raid has decided to ban all the electronical devices. A compass, a Roadbook given by the organization and the intuitions and organization of the teams are the only things allowed to complete the race. This is how the first expeditions started and the authentic concept of rally-raid.

Each stage of the race is different from the others for difficulties, length and for ground characteristics. The route is mostly placed in the south of Morocco and is composed almost entirely of dirt road and sandy areas in the desert.

Figure 3:Panda Raid [2]

Since the runs are very long and difficult and since the equipment allowed is basic, Panda Raid provides a professional crew made of emergency doctors, logistic staff, classification experts and many others to guarantee a high level of safety for all the participants.

On the other hand, to keep a certain level of adventurous sensations the staff will intervene just in case of emergency and no additional help will be provided if not needed. To further improve safety, each vehicle is equipped with a satellite device that gives the real time position of the car to the organization. Adventure is ensured as shown in Figure 2: Panda Raid [1] and Figure 3.

The race starts in the city of Madrid and it develops through more than 3000 km and it finishes in the city of Marrakech. Following the concept of “Rally Cross Country”, the Morocco state is crossed from east to west and from north to south in every direction. The only things known for sure are the starting point and the finishing point of each stage, it is up to the teams to find a way in the middle.

The vehicle admitted to the competition are the Fiat Panda 2WD or 4WD, the Seat Panda and the Seat Marbella. All of them must have been produced before 2004 because they are of very simple construction and very easy to repair during the race (Figure 4).

Figure 4: Repairings during the race

It is known that using an internal combustion engine to drive a car, carbon dioxide is emitted in the atmosphere. Panda Raid really cares about the environmental aspect and for this reason the CO_2 emissions of the whole race will be estimated, basing on the fuel consumed. Then, through the GoodPlanet foundation, an amount of money will be donated to environmental associations to compensate for the emissions.

For example, on past editions, Panda Raid contributed to the construction of a bioclimatic school for the children of Ait Ahmed, as well as to the promotion of beekeeping and of biodiversity in the forest of Mesguina.

The final ranking of the race is based on a scoring system that takes into account the navigation of the teams through the routes and the mean speed inside some sectors of the stages.

2. TRADE-OFF ANALYSIS BETWEEN 2WD AND 4WD STARTING PLATFORMS

The aim of the following chapter is to identify the optimal modification to be implemented to the Panda 1°gen. transmission to obtain the hybridization of the vehicle. In particular, the gasoline engine will be kept to move the front wheels while the electric motor will give power to the rear axle to obtain a parallel hybrid system.

2.1. 2WD and 4WD preliminary considerations

The objects of the comparative trade-off are the Panda 2WD for road use and the Panda cross 4WD with manual insertion of the four-wheel-drive mode, which is widely used in off-road terrains.

For this analysis the workshop manuals of the Panda equipped with the 1108 fire mpi engine have been used as reference books. This engine is the same for both the platforms but the transmission to the wheels is much different so they will be analyzed in detail in the next reports.

Next, a list of the hybridization strategies will be presented for both the versions, following the measurements taken to evaluate the available spaces for the new components: electric motor, gearbox, drive shafts and battery pack.

Then also many other factors related to the complexity, manufacturability, cost and assembly simplicity will be considered in the last phase of the trade-off. Considering all these aspects the best platform will be chosen as starting point for the hybridization.

The second part of this dissertation will explain the design process of the chosen layout.

Starting from a 3D model created with the software Solidworks, a structural analysis of the components will be performed, to ensure the correct behavior during operation.

2.2. Transmission system of the 4WD Panda

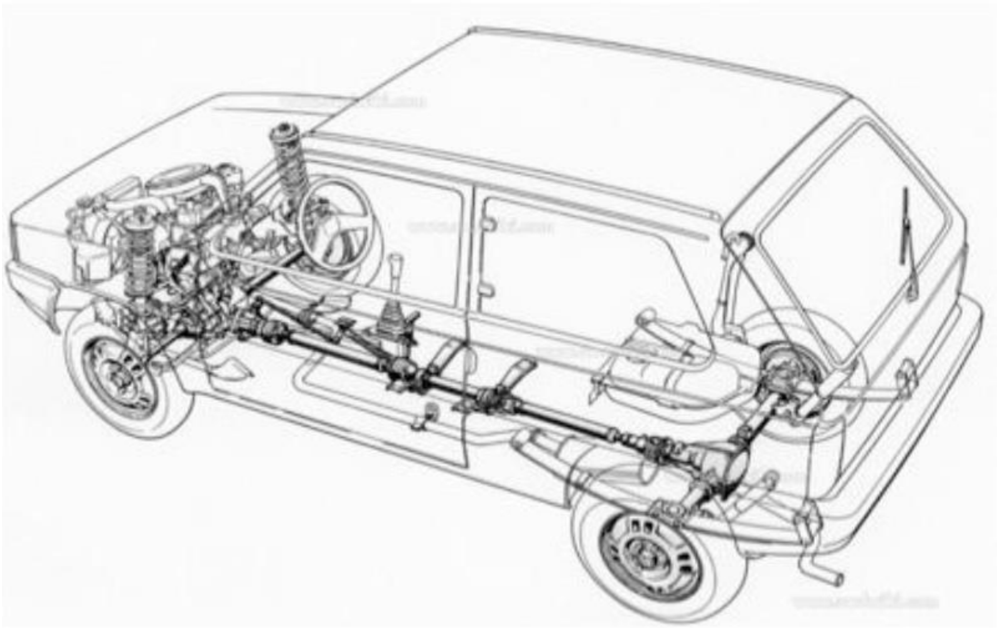


Figure 5: 4WD Panda transmission

The transmission system presented in Figure 5 allows the power flow from the engine to the front wheels during normal 2WD operation. By inserting the 4WD mode, through a lever, it is possible to transmit the power to all four wheels. The peculiarity is that there is no central differential so the torque can go to the axle which has more grip, assuring a very good off-road capability.

Below the two operation modes are shown in Figure 6:

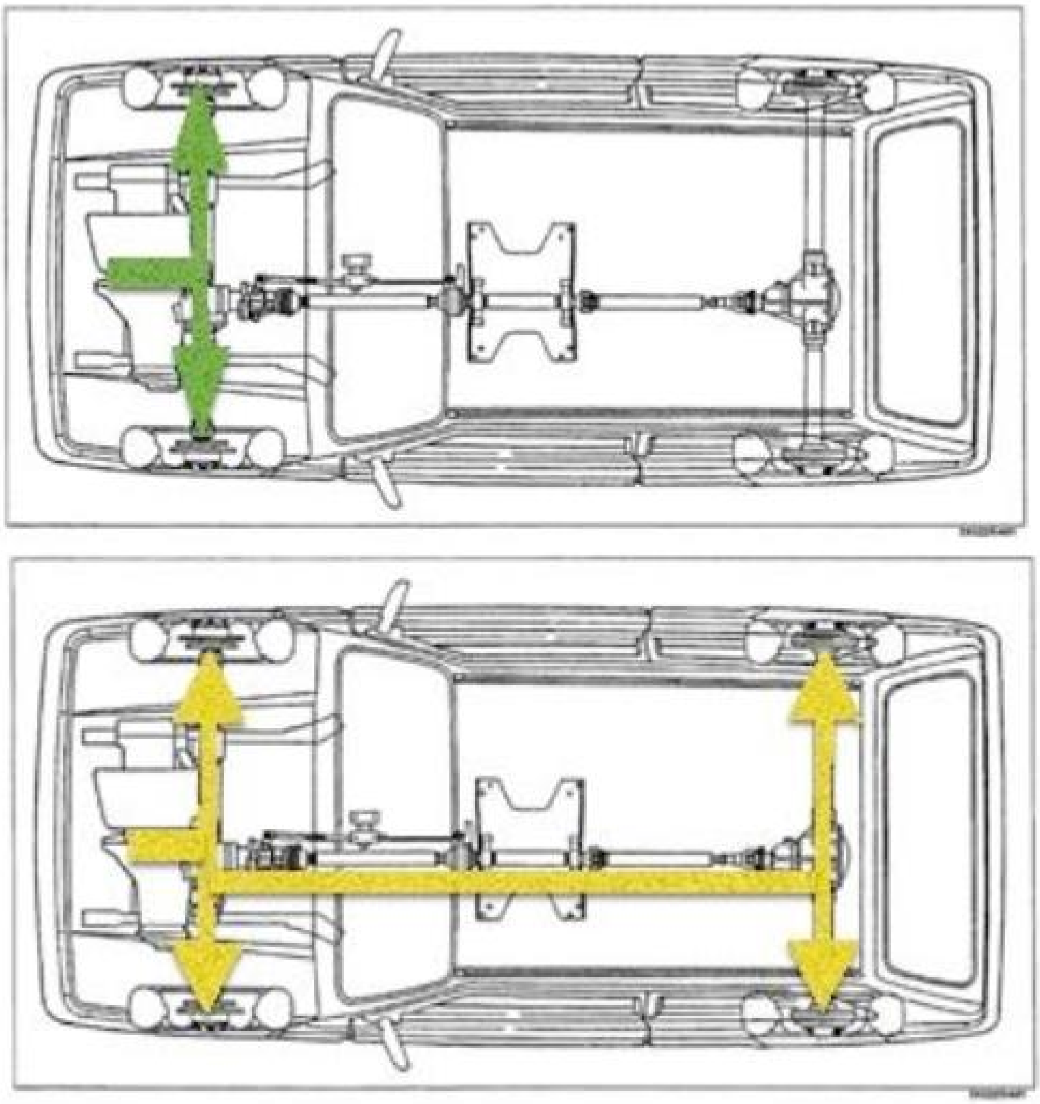


Figure 6: 2WD and 4WD operation modes

During 4WD operation mode the power flows through the following components (highlighted in Figure 7):

- the <u>five gears gearbox</u> (green color);
- the <u>front differential with power splitter</u> (red color);
- the <u>4WD activation system</u> (blue color);
- the <u>transmission shaft with 4 joints</u> (yellow color);
- the <u>rear differential</u> (grey color);

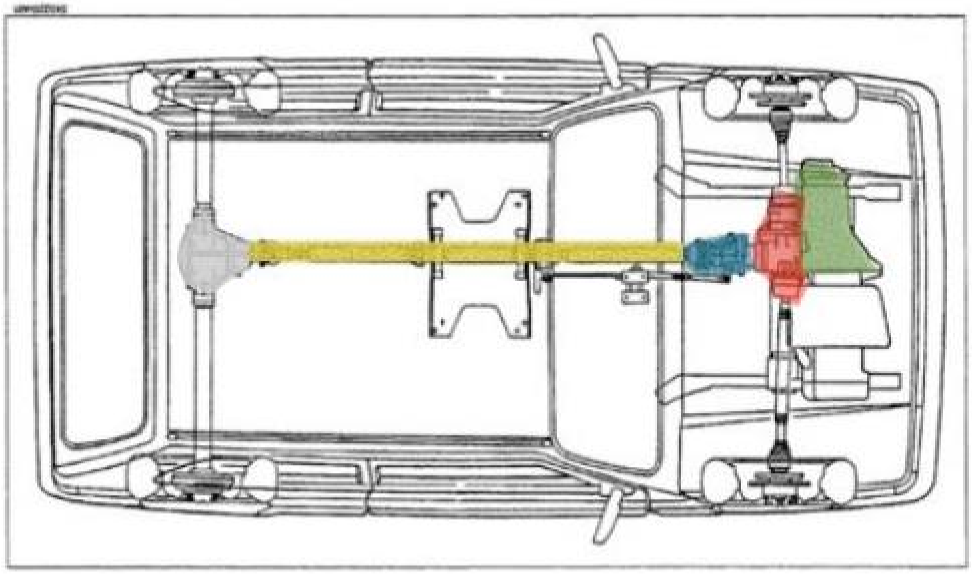


Figure 7: Components involved in power flow during 4WD operation

Five gears Gearbox

The gearbox is a 5-speed synchronized manual transmission with helical teeth gears apart from the reverse speed which has spur gears, as summed up in the following table:

Sincronizzatori	ad anello elastico (tipo Porsche)	3 4 5
	ad anello libero	1 2
Ingranaggi	a denti diritti	R
	a denti elicoidali	1 2 3 4 5

Table 1: Gears and synchronizers types for the different speeds

In the next table the gear ratio of each speed is shown:

		Panda 4x4 1108 mpi
Rapporto ingranaggi	1	3,909
	2	2,056
	3	1,272
	4	0,978
	5	0,731
	R	3,727

Table 2: Gear ratios

<u>Front differential with power splitter</u>

At the gearbox exit shaft there is the conical joint which moves the front differential. There is a gear attached to the differential gear which links the transmission shaft towards the rear axle transferring power to the rear wheels when needed, as shown in Table 3:

DIFFERENZIALE - RINVIO	1108 i.e.
Rapporto coppia cilindrica di riduzione	11/60 (5,455)
Rapporto coppia conica di rinvio	14/41 (2,929)

Table 3: Differential gear ratios

4WD activation system

Next to the cylindric gear of the differential there is a conical gear that moves a short longitudinal shaft; in practice the differential is linked to two final transmissions: a cylindrical one and a conical one. At the end of the longitudinal shaft there is a coupling sleeve that, when triggered by a user commanded lever, links the transmission shaft for the rear axle, activating the 4WD mode.

In the next figure it is possible to see the mechanism:

Figure 8: 4WD activation system

Transmission shaft with 4 joints

The transmission shaft is divided in three parts: the front one can move slightly to compensate for the movements of the engine and is linked at the ends with two sliding joints. The central part is fixed to the chassis and is linked to the third and final part through a universal joint. The third part of the transmission shaft must compensate for the relatively wide movement of the rear axle and so it is equipped with a sliding joint as the first piece. The transmission shaft is represented in Figure 9:

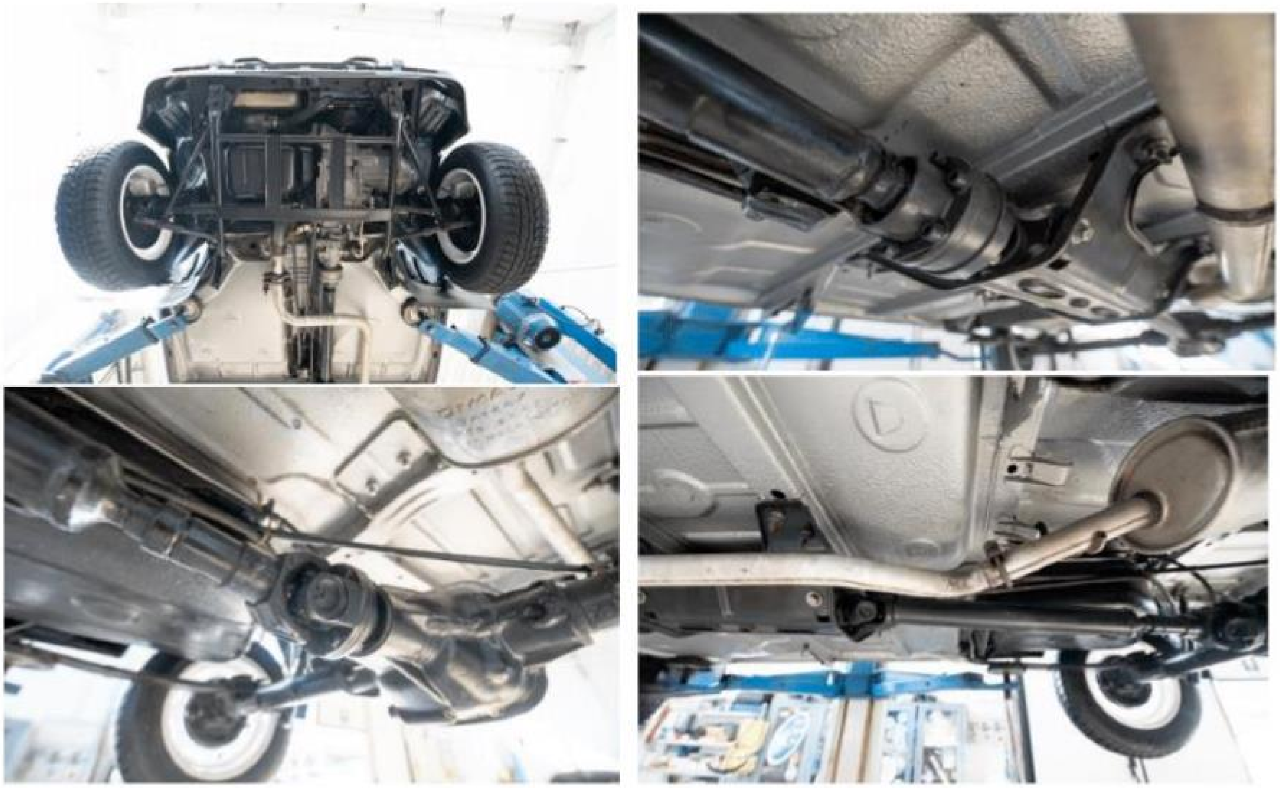


Figure 9: Transmission shaft

Suspensions

- **Panda 4WD rear suspensions:** rigid axle suspension with a tube linked to the vehicle frame with two longitudinal leaf springs and with double effect hydraulic dampers.
- **Front suspensions (the same in 2WD and 4WD vehicles):** MacPherson type independent wheels suspension with double effect dampers.

In Figure 10 both front and rear suspensions are shown:

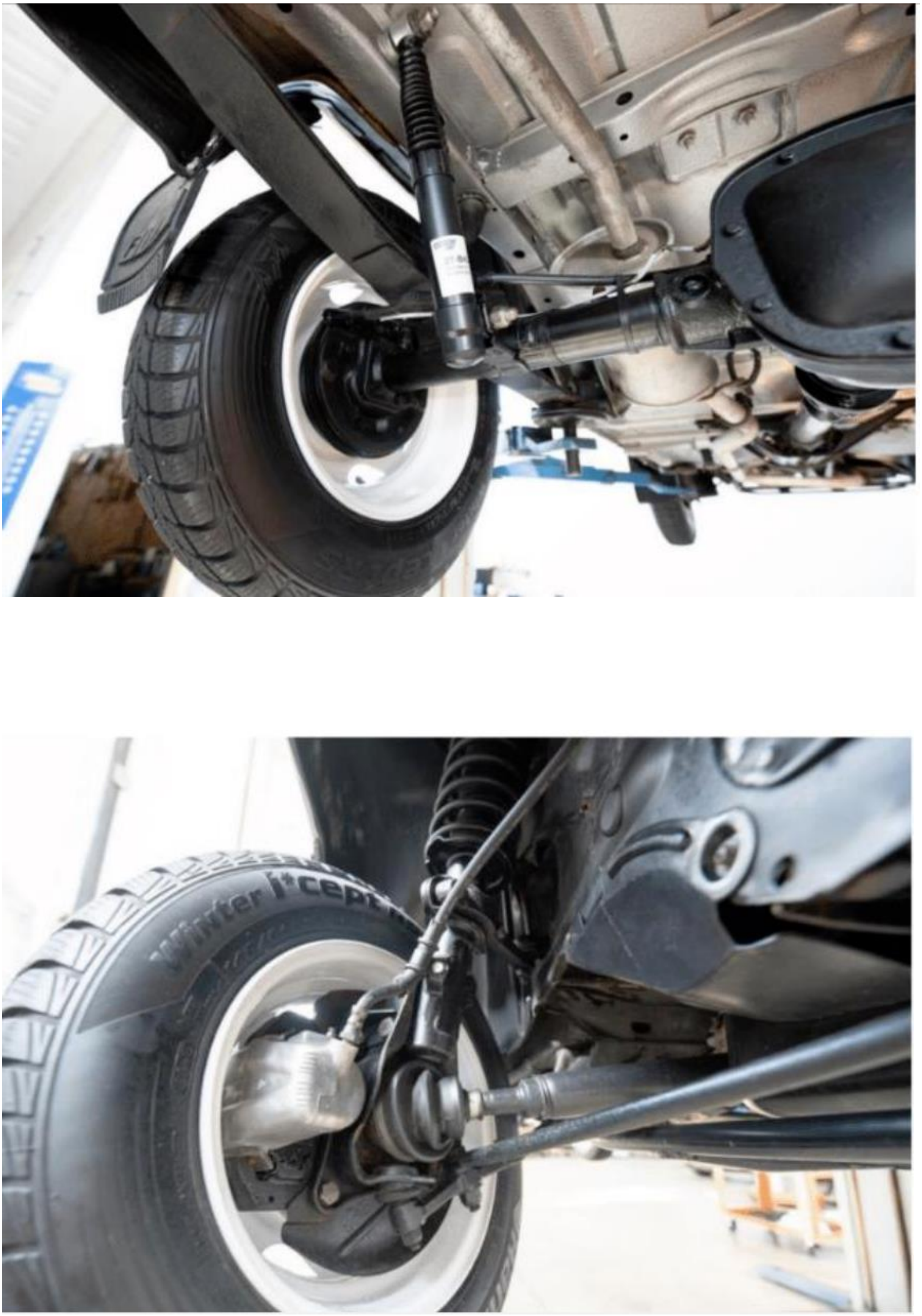


Figure 10: Rigid axle suspension (above) and MacPherson (below)

Summary of the transmission kinematics

Table 4: Total gear ratios

	Gear	Rapporti ingranaggi	Rapporto finale (motore-ruote)
Rapporto ingranaggi	1	3,909	21,324
	2	2,056	11,215
	3	1,272	6,939
	4	0,978	5,335
	5	0,731	3,988
	R	3,727	20,331

and Figure 11 resume the transmission ratios of all the gears and the respective final ratios (engine-wheels) to make the front-rear kinematic clearer.

Table 4: Total gear ratios

	Gear	Rapporti ingranaggi	Rapporto finale (motore-ruote)
Rapporto ingranaggi	1	3,909	21,324
	2	2,056	11,215
	3	1,272	6,939
	4	0,978	5,335
	5	0,731	3,988
	R	3,727	20,331

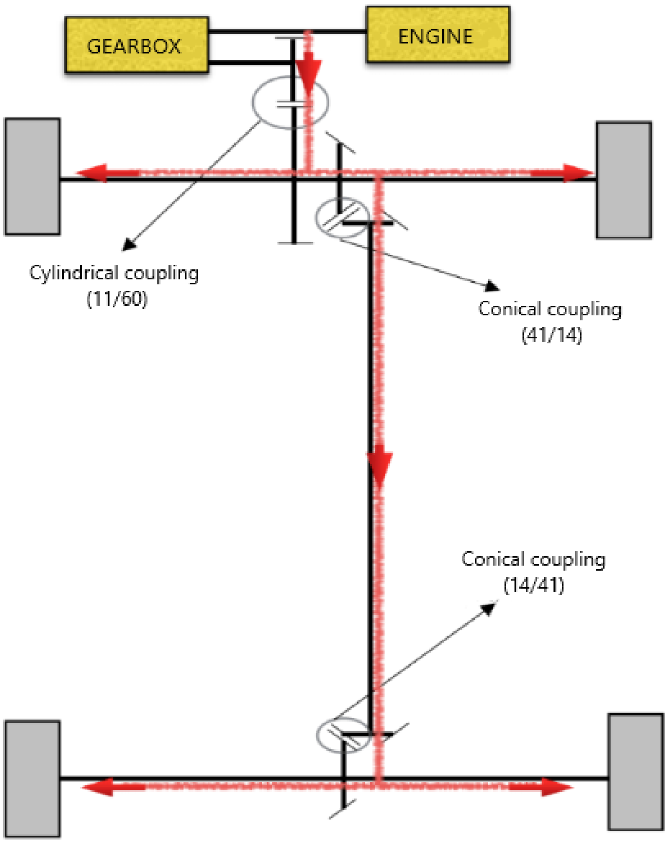


Figure 11: Transmission scheme

The final gear ratios have been obtained considering the coupling with the differential gear, with a final ratio of 5,455.

A section view of the differential group is shown in Figure 12:

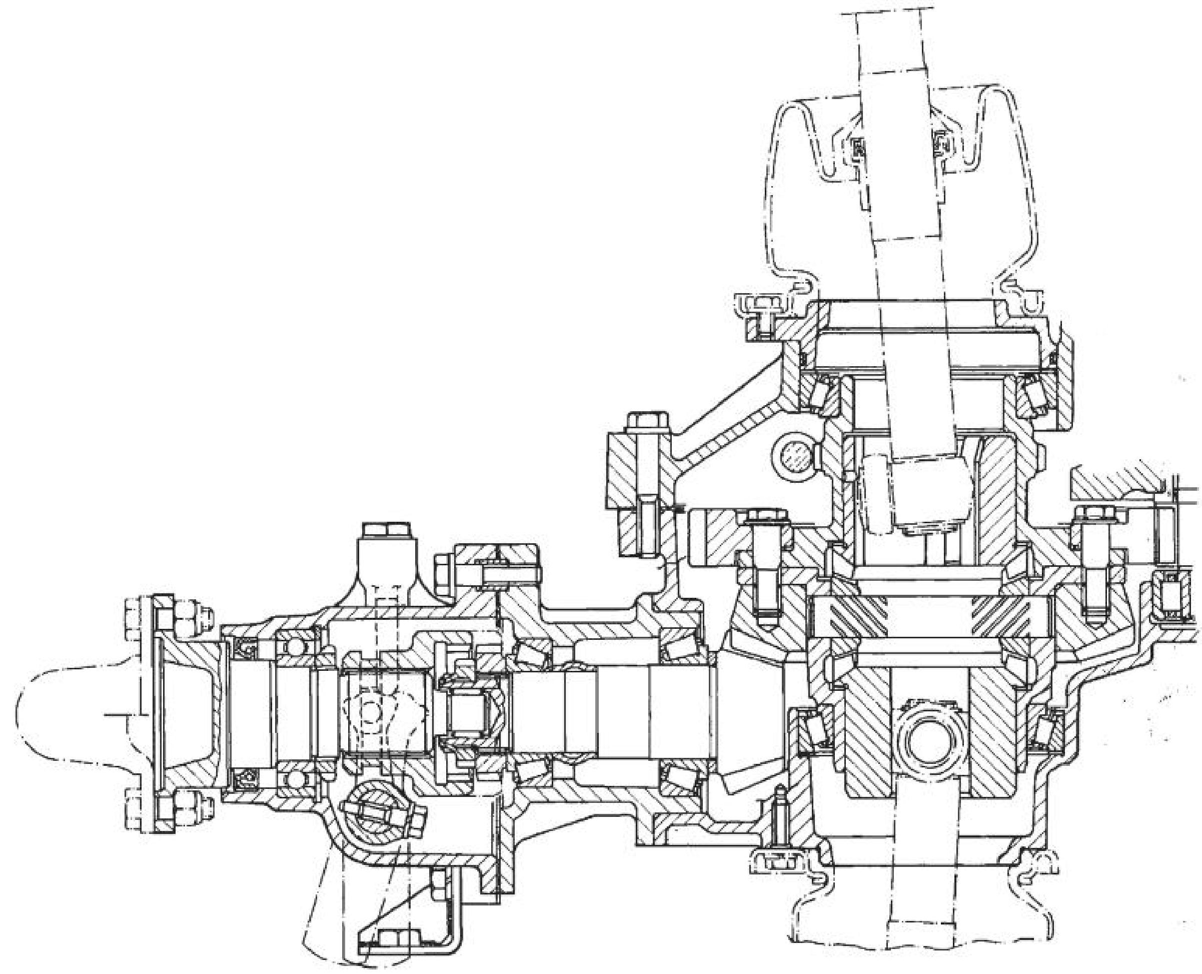


Figure 12: Section view of the differential group

<u>Bodywork</u>

From the workshop manual [1] many useful pieces of information were obtained about the measures of the frame of the Panda. These data were essential for the design phase of the new suspension layout with regards to the overall dimensions of the components of the hybrid powertrain.

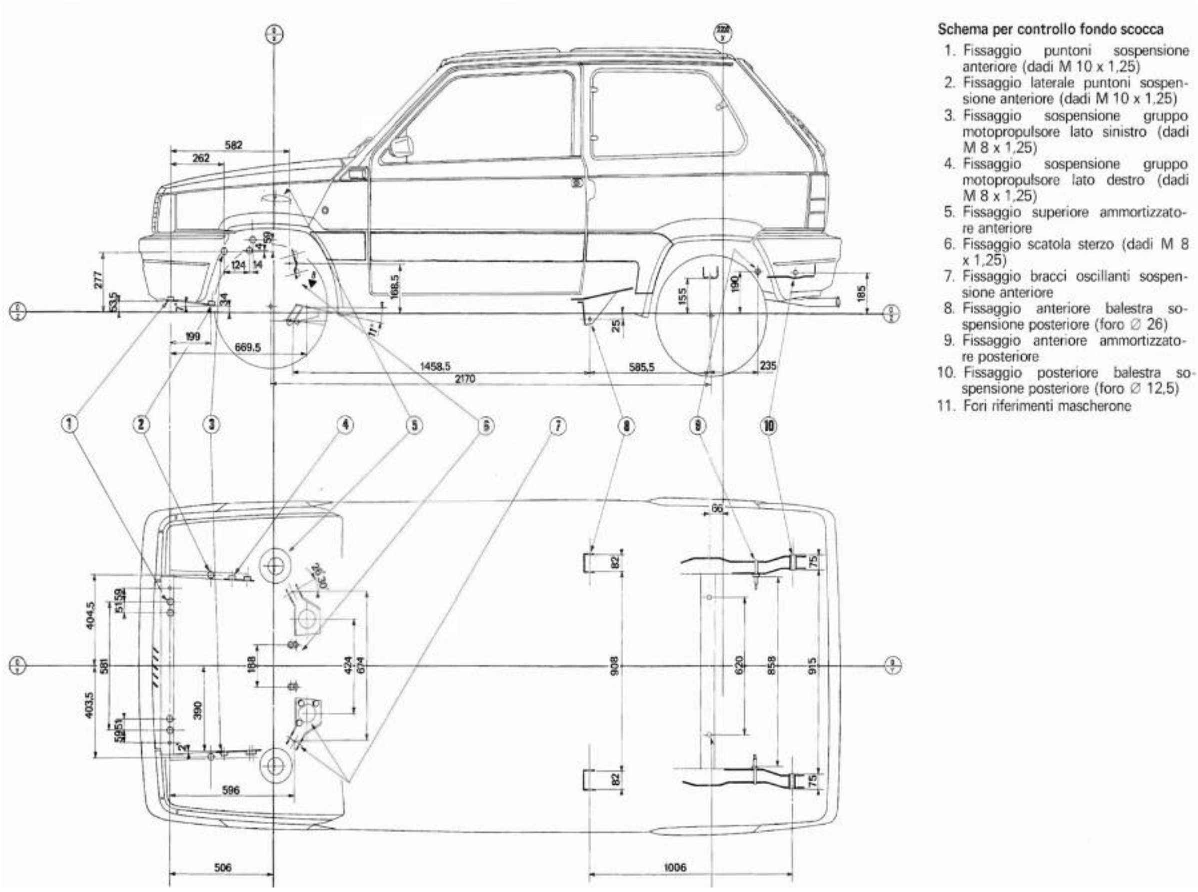


Figure 13: Bodywork dimensions Panda 4WD

2.3. Transmission system of the 2WD Panda

The Panda 2WD transmission system is much simpler than the 4WD one because the traction is just on the front axle. All the components transmitting power to the rear wheels are absent, including the levers for the actioning of the 4WD mode, the transmission shafts and the rear differential.

This leads to a different rear suspension system which now is constituted by a omega tube connecting the two wheels semi-independently. The wheel hubs are different too, due to the absence of the drive shafts and the presence of drum brakes.

Gearbox and differential

The Panda 2WD gearbox has the same mechanical components as the 4WD one, except for the gear ratios that are different with respect to the previous one. In Figure 14 the gearbox and the differential are shown: note that the conical gears giving power to the rear axle here are absent.

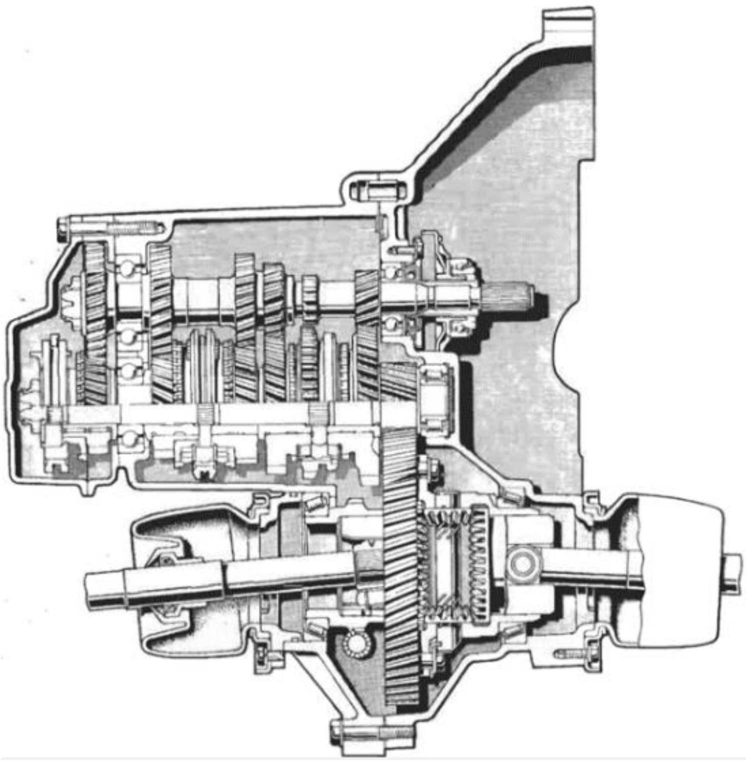


Figure 14: Panda 2WD gearbox and differential

From Table 5, Table 6 and Table 7 the differences between the 2WD and 4WD are clear:

CAMBIO DI VELOCITÀ

		Panda 1108 mpi	Panda 4x4 1108 mpi
		1 3 5 / 2 4 R	
	Tipo	C 526	
Sincronizzatori	ad anello elastico (tipo Porsche)	3 4 5	
	ad anello libero	1 2	
Ingranaggi	a denti diritti	R	
	a denti elicoidali	1 3 5 2 4	
Rapporto ingranaggi	1	3,909	3,909
	2	2,158	2,056
	3	1,345	1,272
	4	0,974	0,978
	5	0,766	0,731
	R	3,818	3,727

Table 5: Gear ratios comparison

DIFFERENZIALE - RINVIO

Rapporto	coppia cilindrica di riduzione	–	14/41 (2,929)
	coppia cilindrica di riduzione	16/57 (3,562)	11/60 (5,455)

Table 6: Differential ratios comparison

DIFFERENZIALE - RINVIO		Panda 1108 mpi	Panda 4 x 4 1108 mpi
Rapporto coppia cilindrica di riduzione		16/57 (3,562)	11/60 (5,455)
Rapporto sulle ruote	1	13,924	21,324
	2	7,868	11,215
	3	4,791	6,939
	4	3,469	5,335
	5	2,728	3,988
	R	13,600	20,331

Table 7: Gear ratios engine-wheels comparison

Suspensions

As previously mentioned the rear suspensions of the Panda 2WD the principal component is a Ω tube with central hinging to the frame. The other components attached to the tube are the longitudinal swinging arms, the helical springs and the double effect dampers.

Figure 15 illustrates the suspension geometry:

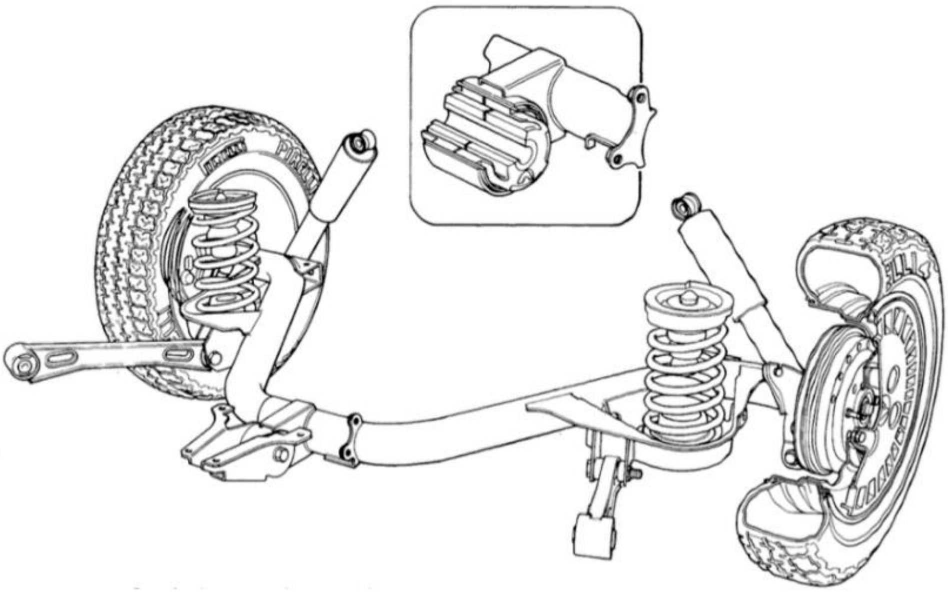


Figure 15: Rear suspension system Panda 2WD

2.4. Possible hybridization strategies for the Panda 4WD

- <u>Modification of the rear rigid axle with the installment of a transaxle</u>
- <u>Installment of an electric motor and gearbox along the transmission shaft</u>

2.4.1. Modification of the rear rigid axle with the installment of a transaxle

This solution consists of modifying the rear rigid axle to make room for the installation of a transaxle unit, which includes an electric motor, a gearbox and a differential group. The transaxle will be linked to the rigid axle, substituting the original differential. The transaxle group and the drive shafts to be used are shown in Figure 16 and Figure 17:

Figure 16: Transaxle group

Figure 17: Transaxle drive shafts

This system has some big disadvantages. The first one is that the transaxle would be a very high unsprung mass on the rear part of the car, changing drastically the vehicle dynamics. The second one is a manufacturability problem: it is very expensive and time consuming to find a way to accommodate the semi axis inside the rigid axle and to link the whole system to the chassis. Moreover, the original leaf spring suspensions should be changed leading to a complete redesign of the suspension system.

2.4.2. Installment of an electric motor and gearbox along the transmission shaft

This solution involves the insertion of the electric unit along the transmission shaft, which will be shortened depending on the installment position of the motor. Consequently, all the mechanical parts downstream of the motor, i.e. the rear differential and the rear rigid axle, would not be modified.

Since some original components transmitting power are kept, some considerations should be done for the dimensioning of the electric motor and gearbox, to avoid failures of the components due to high stresses.

The first step is to know the maximum power flowing through the components during the normal mechanical operation in 4WD mode.

Let's consider the worst case. Since there is no central differential, when the engine outputs the maximum power (40 kW), if the front axle has no traction, all the torque goes to the rear wheels, supposing that the rear axle has enough grip.

Considering all the gear ratios of the original gearbox and differential and taking into account the mechanical specifications of the engine, all the speed and torque ranges at which the components work were carried out.

In Figure 18 and Figure 19 the scheme of the new powertrain is shown, as well as the engine characteristic power and torque curves:

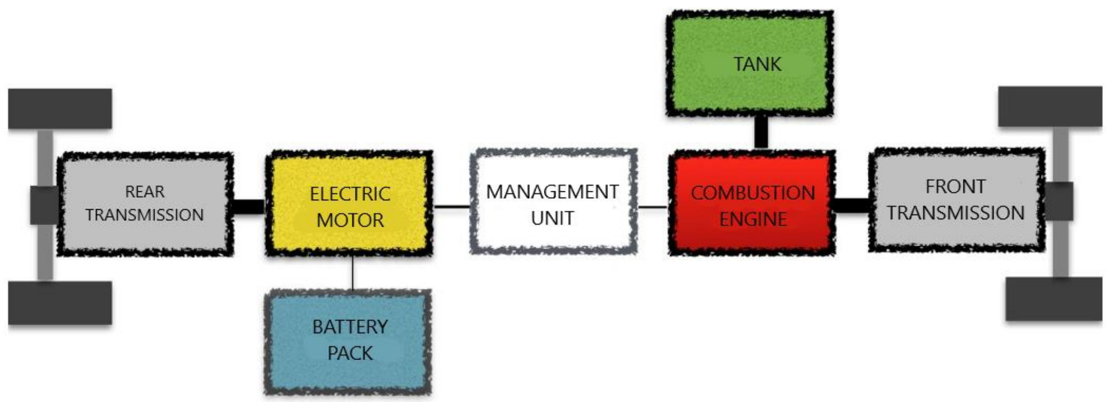


Figure 18: New powertrain scheme

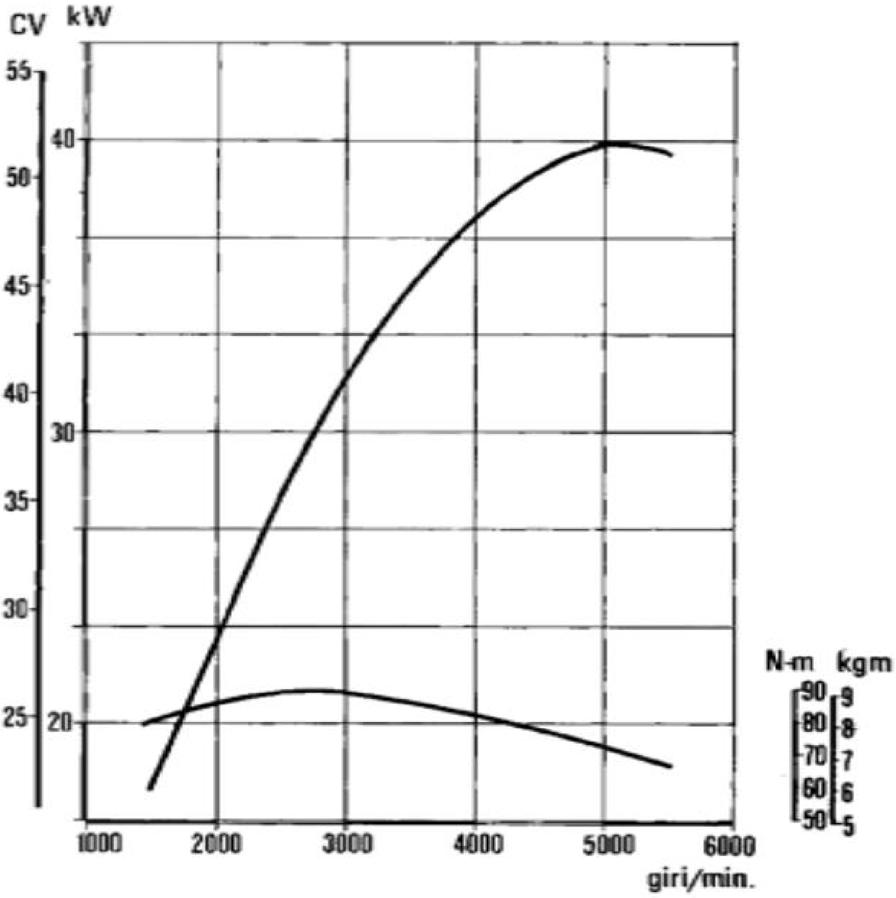


Figure 19: Engine power and torque curves

Power range

All the mechanical parts of the rear drivetrain including the joints, the shafts, the differential and the suspension system are developed to sustain a maximum peak power and torque under the most severe conditions. To not design from scratch a new powertrain and suspension system, it is important that the power generated by the electric motor does not overcome 40 kW, as mentioned previously.

Speed range

For the speed range it is possible to start from the maximum speed of the vehicle considering the maximum rotational speed of the wheels, or from the engine maximum rotational speed. In both cases the velocities at the shafts will be the same.

In the following table all the variables used for the next calculations are listed:

$\boldsymbol{\tau_V}$	Fifth gear transmission ratio
$\boldsymbol{\tau_{diff}}$	Differential cylindric gear transmission ratio
$\boldsymbol{\tau_r}$	Conical coupling transmission ratio
$\boldsymbol{n_1}$	Primary shaft maximum rotational speed
$\boldsymbol{n_2}$	Secondary shaft maximum rotational speed
$\boldsymbol{n_{diff}}$	Differential gear maximum rotational speed
$\boldsymbol{n_a}$	Transmission shaft maximum rotational speed

Table 8: Variables used in the kinematic study

$$\tau_V = 0{,}731 \quad (1)$$

$$n_1 = 5000\ rpm \quad (2)$$

$$n_2 = \frac{n_1}{\tau_V} = 6840\ rpm \quad (3)$$

$$\tau_{diff} = 5{,}455 \quad (4)$$

$$n_{diff} = \frac{n_2}{\tau_{diff}} = 1254\ rpm \quad (5)$$

$$\tau_r = 2{,}929 \quad (6)$$

$$n_a = n_{diff} \cdot \tau_r = 3673\ rpm = n_{max} \quad (7)$$

In Figure 20 a scheme of the gearbox and the differential is illustrated:

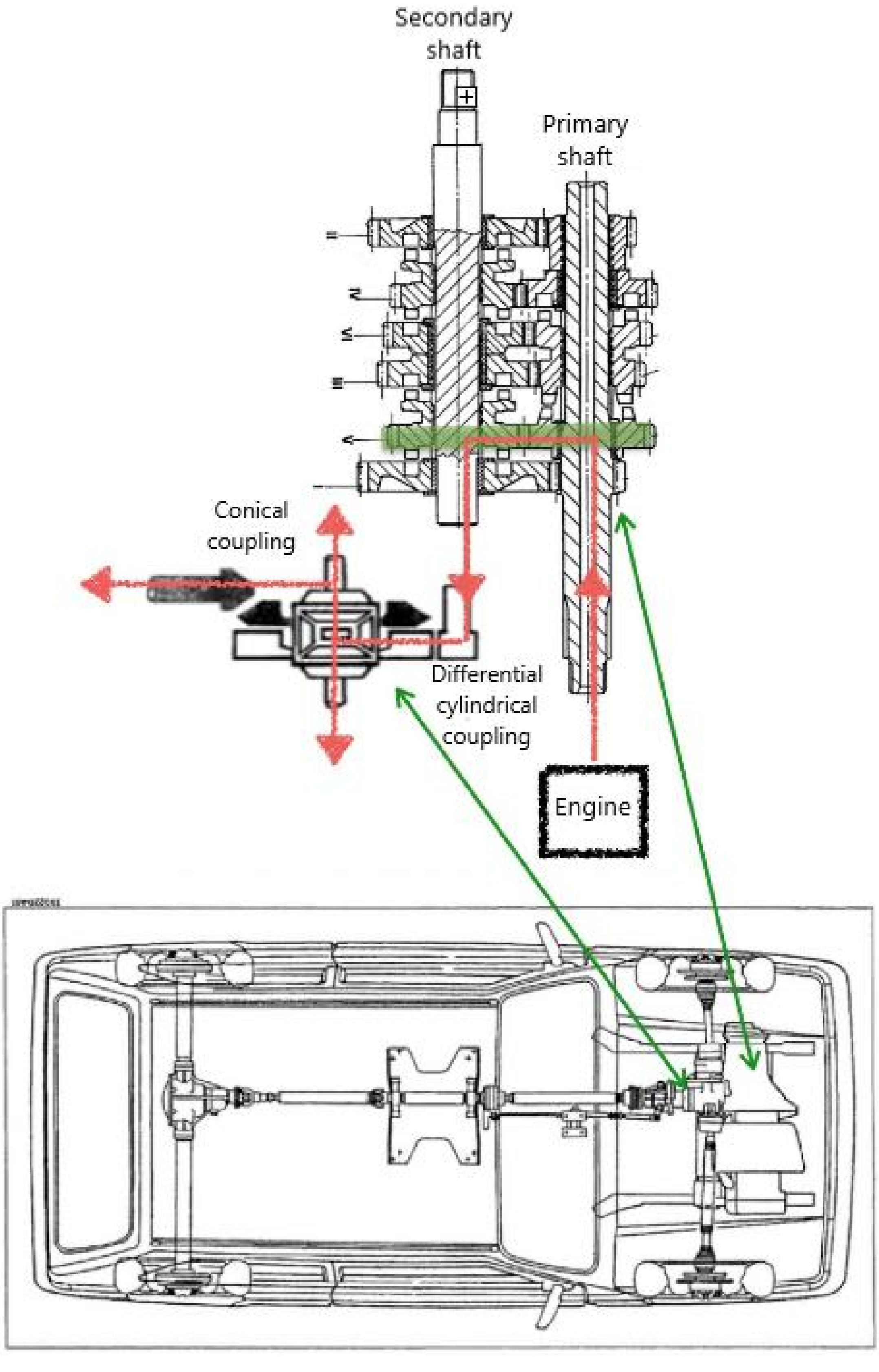


Figure 20: Gearbox-differential scheme with 5th gear engaged

Torque range

The starting point for the torque range is the mechanical characteristic of the engine where the maximum torque of the engine corresponding to 88 Nm is given. Then using the gear ratio of the first gear (the highest one) and of the differential, the maximum torque on the transmission shaft was easily computed. The following variables have been used:

$\boldsymbol{\tau_1}$	First gear transmission ratio
$\boldsymbol{\tau_{diff}}$	Differential cylindric gear transmission ratio
$\boldsymbol{\tau_r}$	Conical coupling transmission ratio
$\boldsymbol{C_1}$	Primary shaft maximum torque
$\boldsymbol{C_2}$	Secondary shaft maximum torque
$\boldsymbol{C_{diff}}$	Differential gear maximum torque
$\boldsymbol{C_r}$	Maximum torque on the conical coupling
$\boldsymbol{C_a}$	Transmission shaft maximum torque

Table 9: Variable used in the torque study

$$\tau_1 = 3{,}9 \tag{8}$$

$$C_1 = 88\ Nm \tag{9}$$

$$C_2 = C_1 \cdot \tau_1 = 342{,}2\ Nm \tag{10}$$

$$\tau_{diff} = 5{,}455 \tag{11}$$

$$C_{diff} = C_r = C_2 \cdot \tau_{diff} = 1873\ Nm \tag{12}$$

$$\tau_r = 2{,}929 \tag{13}$$

$$C_a = \frac{C_r}{\tau_r} = 640\ Nm = C_{max} \tag{14}$$

With this configuration the values of torque and speed needed on the transmission shaft require the use of a gearbox because the electric motor operates at much different speeds and torques with respect to the transmission shaft. For this reason the design of a single stage or even a double stage gearbox is mandatory.

In Figure 21 it is shown schematically the solution for this layout:

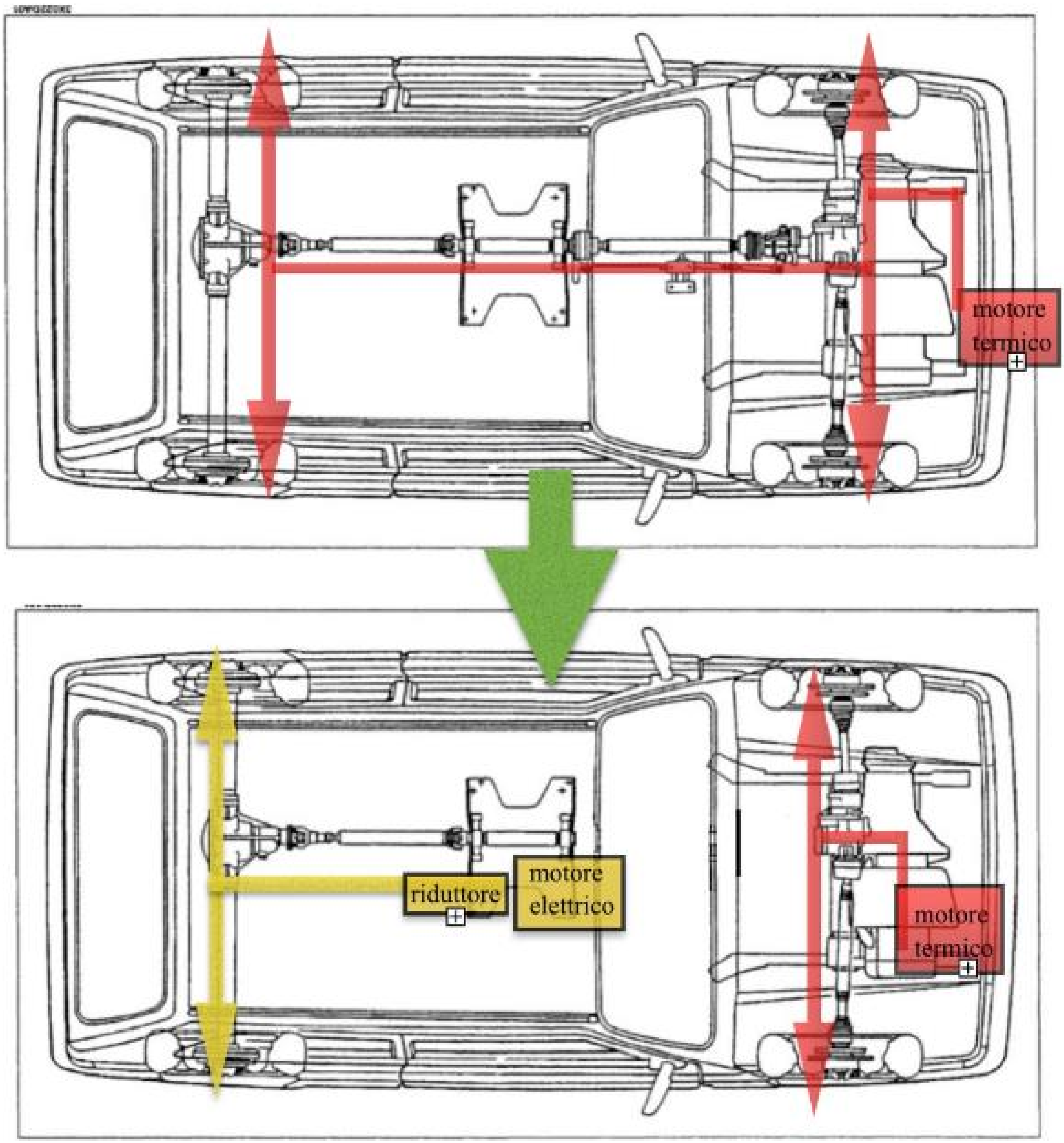


Figure 21: Panda 4WD hybridization scheme

The next step is the evaluation of the positioning of the electric motor with its gearbox along the transmission shaft. Different solutions have been considered:

1. Complete removal of the transmission shaft and consequent attachment of the electric motor to the rear differential. This would cause a substantial increase in the load of the rear suspensions, considering that the weight of the motor-gearbox and the battery pack is more than 100 kg. The leaf spring suspensions should be inevitably changed. Figure 22 represents the solution:

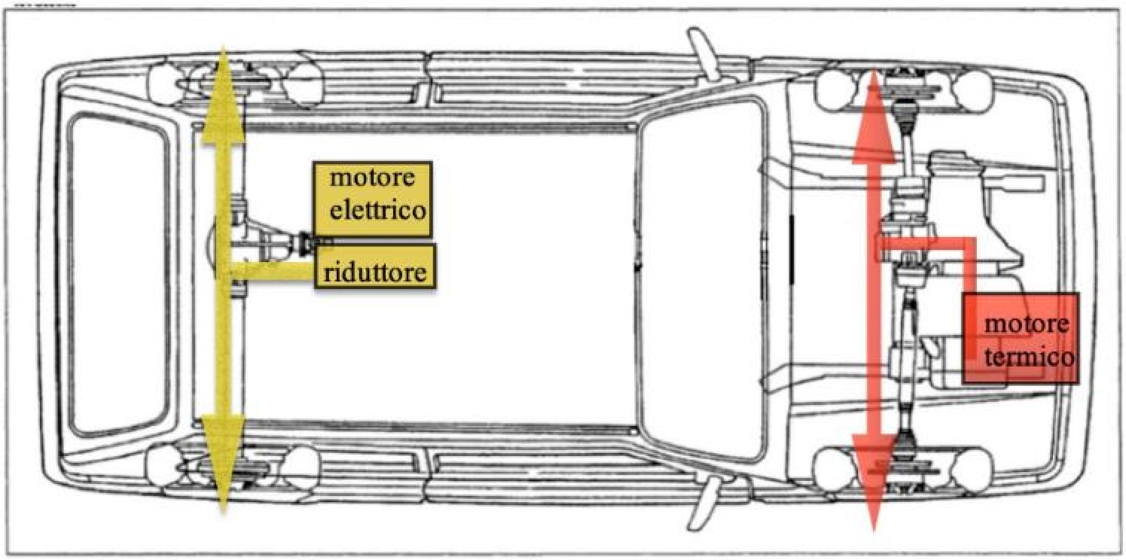


Figure 22: Electric motor attached to the differential

Regarding the vehicle weight and weight distribution other aspects should be considered:

- removal of the 4WD activation system;
- removal of the transmission shaft (about 20 kg)
- insertion of electric motor and gearbox (about 50 kg);
- insertion of battery pack, inverter and cables (about 50 kg);

In the following table the difference between 4WD and 4WD-H are shown:

	4WD	**4WD – H**
weight with 2 people [kg]	960	1010
weight distribution	55:45	50:50

Table 10: Weight difference between 4WD and 4WD-H with 1st layout

2. Another solution is the positioning of the motor instead of the central part of the transmission shaft keeping all the original joints (universal and sliding) as shown in Figure 23.

This configuration has one big disadvantage: the motor-gearbox unit will be positioned lower with respect to the ground due to packaging space and it will be much more exposed to the roughness of the terrain (rocks, dirt and gravel).

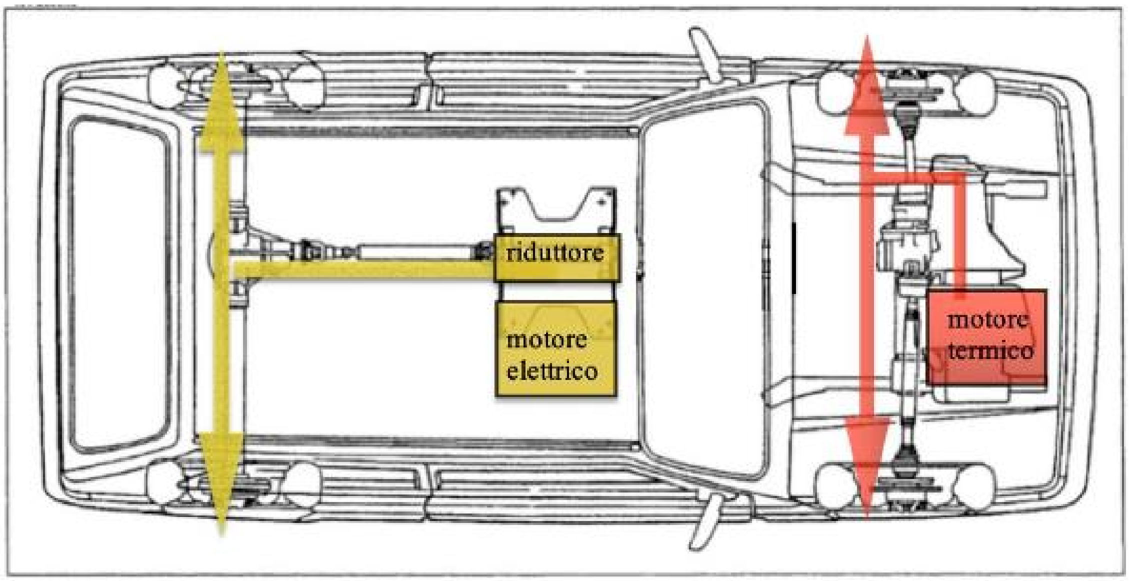


Figure 23: Electric motor attached to the central part of the transmission shaft

Regarding the vehicle weight and weight distribution this time there are some differences with respect to the previous case:

- removal of the 4WD activation system;
- removal of the transmission shaft (about 10 kg)
- insertion of electric motor and gearbox (about 50 kg);
- insertion of battery pack, inverter and cables (about 50 kg);

In the following table there are the differences between 4WD and 4WD-H for the second case:

	4WD	**4WD – H**
weight with 2 people [kg]	960	1020
weight distribution	55:45	52:48

Table 11: Weight difference between 4WD and 4WD-H with 2nd layout

3. Finally, the last layout for this solution is to put the motor instead of the first part of the transmission shaft, in a position not far from the combustion engine, as Figure 24 illustrates.

By doing so there is a lot of weight on the front axle, but it can be balanced by putting the battery pack towards the rear of the car, maintaining a similar weight distribution with respect to the original vehicle.

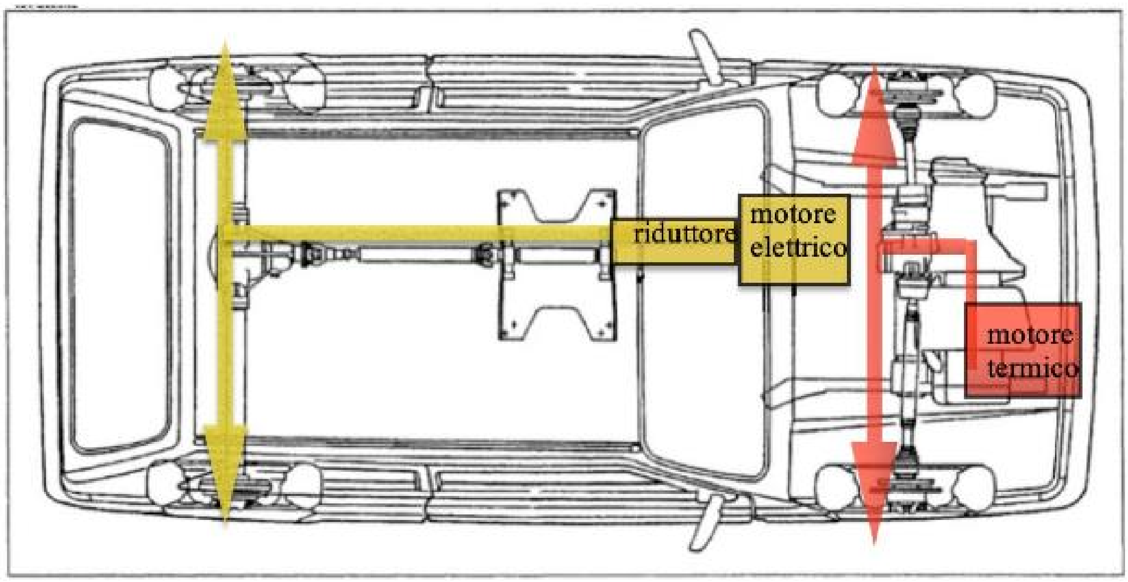


Figure 24: Electric motor on the first part of the transmission shaft

This time the vehicle weight is increased but the weight distribution is the same as the original one, as shown in Table 12: Weight difference between 4WD and 4WD-H with 3rd layout

- removal of the 4WD activation system;
- insertion of electric motor and gearbox (about 50 kg);
- insertion of battery pack, inverter and cables (about 50 kg);

	4WD	**4WD – H**
weight with 2 people [kg]	960	1030
weight distribution	55:45	55:45

Table 12: Weight difference between 4WD and 4WD-H with 3rd layout

Possible layouts

On the basis of the dimensions data of the motor-gearbox unit and of the available space on the vehicle, it was possible to define the layouts described above, which will be illustrated more clearly in the next pages.

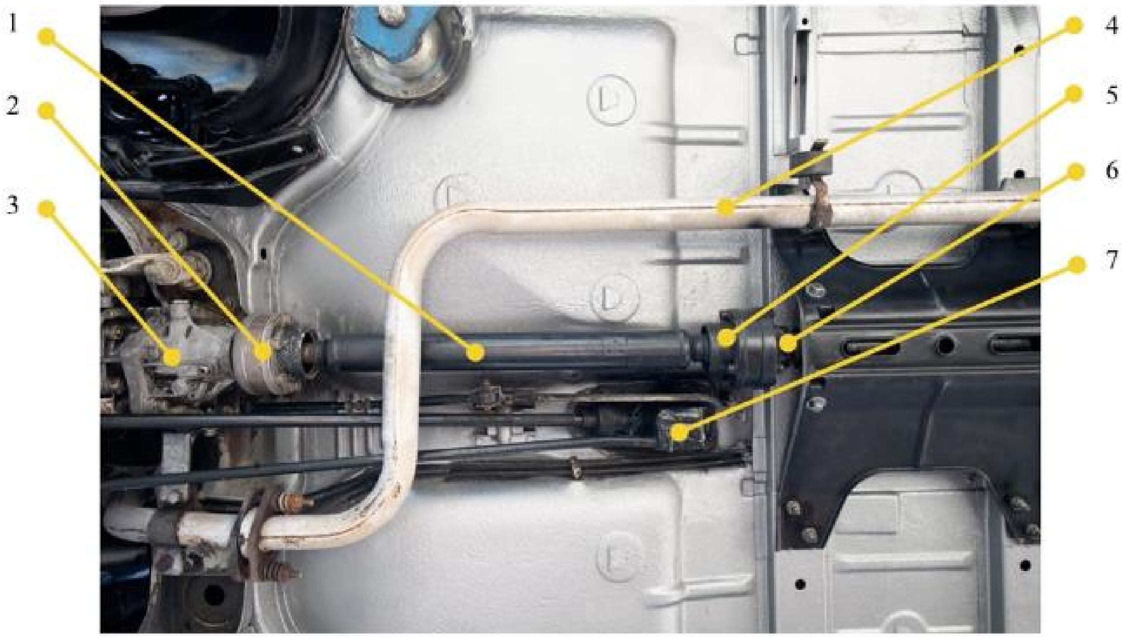


Figure 25: Components of the front part of the transmission

In Figure 25 the components of the transmission of the original vehicle are shown:

1) First part of the transmission shaft;
2) Front homokinetic sliding joint;
3) 4WD mode activation system;
4) Exhaust pipe;
5) Rear homokinetic sliding joint;
6) Starting point of the second part of the transmission shaft
7) Gearbox levers.

By using a single stage parallel shafts gearbox, the focus is on the simplicity of the system and on a low production cost. The problem is that a single stage reduction gearbox can achieve a gear ratio of 3 at maximum, to maintain a good contact between the meshing gears and have a high reliability. Consequently, the output torque and speed do not match the ones requested by the transmission shaft, unless the electric motor has a very high torque output, meaning it will be bigger and more expensive.

An alternative is to opt for a more complex double stage gearbox with a higher gear ratio. The reduction system will be more expensive and more difficult to design, but the motor could be much cheaper and small.

In the first solution the front part of the transmission shaft is removed resulting in:

- the advantage of having a larger space available between the exhaust pipe and the gear levers. Notice that the removal of the 4WD activation system, which is made by a levers system, make available more space for the motor-gearbox unit under the car.

- the disadvantage of being away from anchor points of the chassis such as the engine supports or the supports of the central transmission shaft. Then this solution will have to take into account the study of new anchor points for the motor-gearbox unit.

A 3D CAD representation of the layout is illustrated in Figure 26 in the following page. The components listed below are present:

- Vehicle frame (light-grey color);
- Exhaust pipe (white color);
- Motor-gearbox unit (red color);
- Transmission shaft with protective plate (black color);
- Gear levers (grey color).

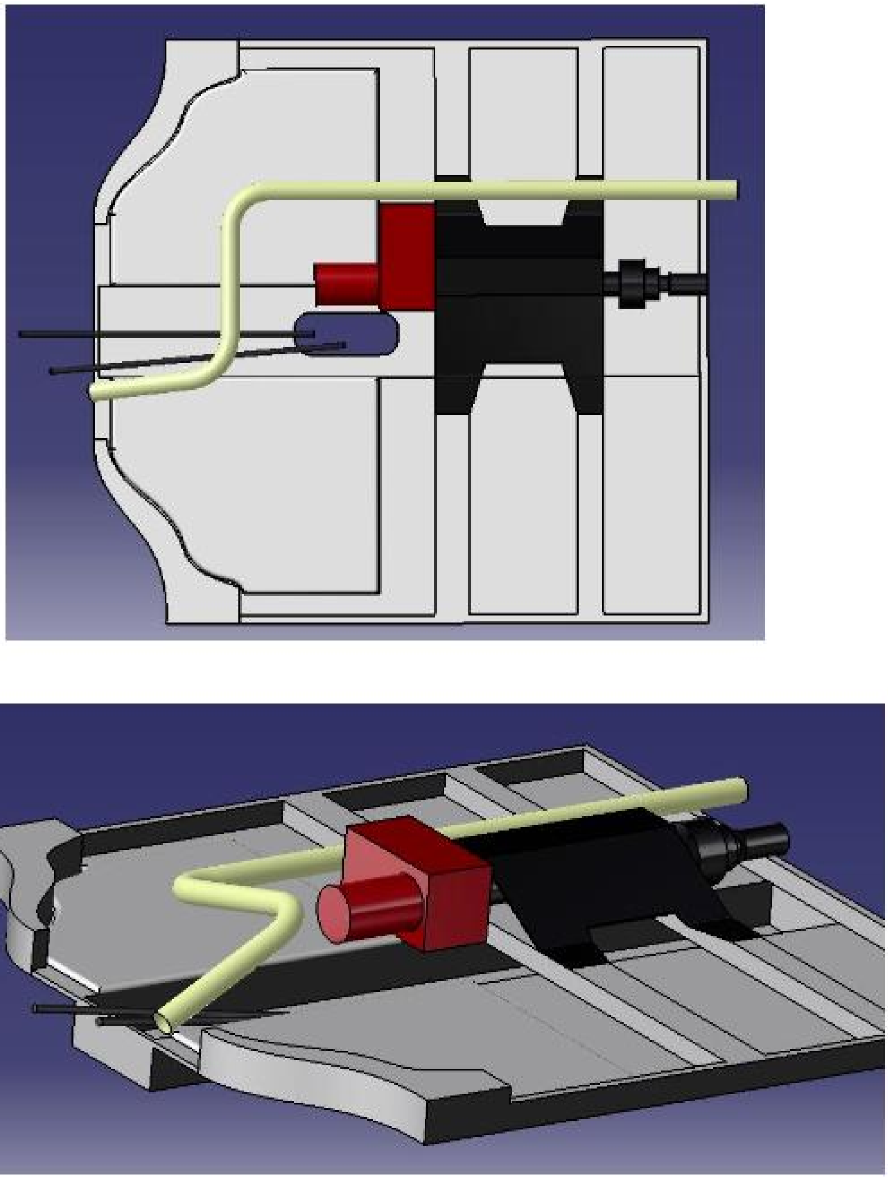


Figure 26: 3D model of solution 1

The first part of the shaft is removed so the output shaft of the reduction system will be machined to couple directly to the central part of the transmission shaft, which is grooved.

Instead, by keeping also the first part of the shaft (Figure 27) there will be:

- A reduced space to couple the reduction system output shaft to the shaft so the front sliding joint will be removed and a single stage gearbox is the only choice.

- The motor-gearbox unit is closer to the engine compartment having more anchor points available.

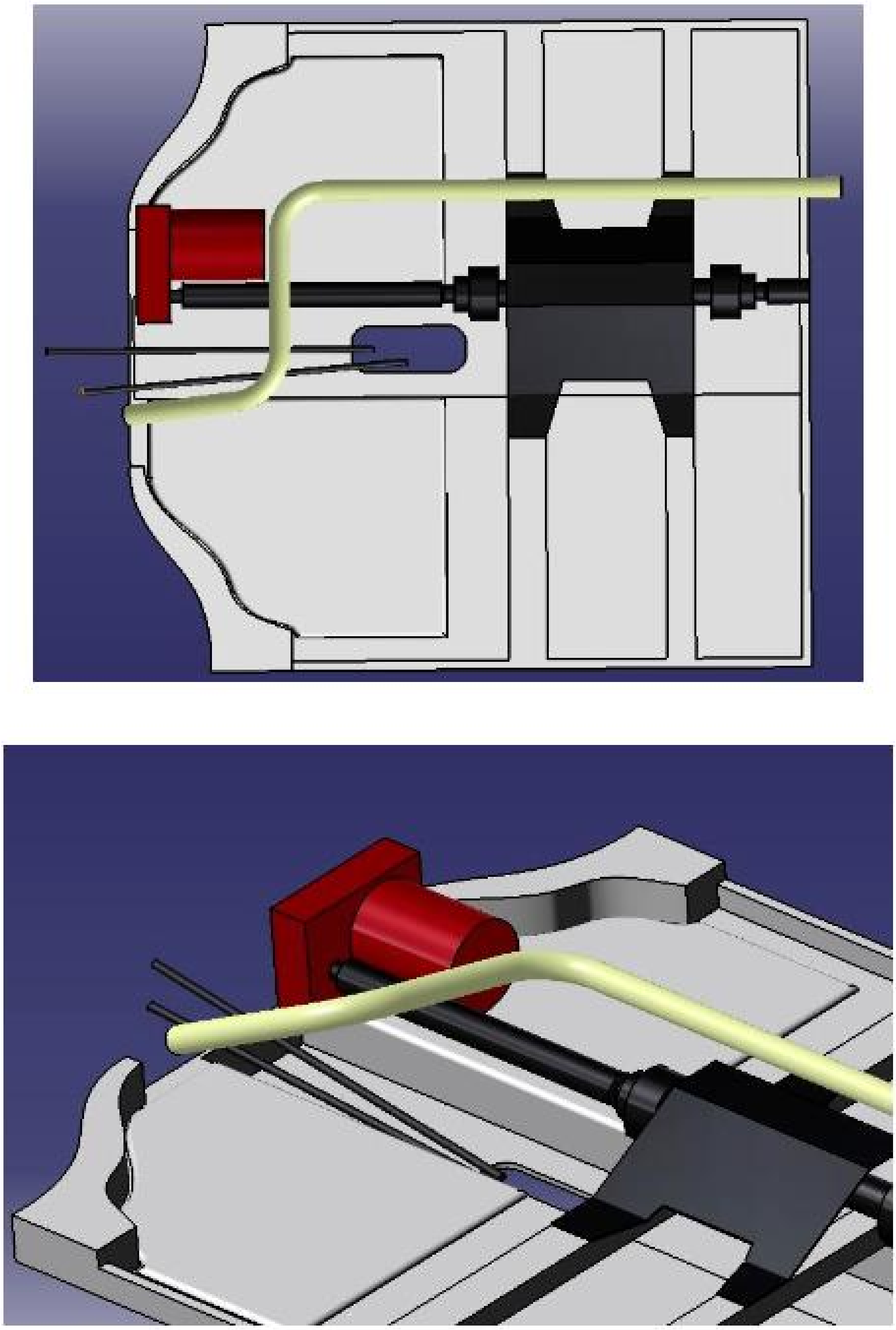


Figure 27: 3D model of solution 2

The front part of the shaft is kept and the output shaft of the gearbox is machined to have a perfect coupling without sliding joints.

2.5. Possible hybridization strategies for the Panda 2WD

The second possibility is to start from the 2WD version of the Panda to realize a 4WD hybrid vehicle.

As seen previously, in the 2WD version the kinematic chain transmitting power to the rear axle is missing and the rear suspensions are semi-independent with a Ω tube connecting the wheels.

As shown in Figure 15, the Ω suspension geometry leaves a big central space. This space is originally occupied by the tank. For the hybridization process the idea is to remove the rear seats of the car and to move the tank inside the cockpit, together with the battery pack. This is essential for the positioning of the transaxle unit without the modification of the suspension geometry (Figure 28).

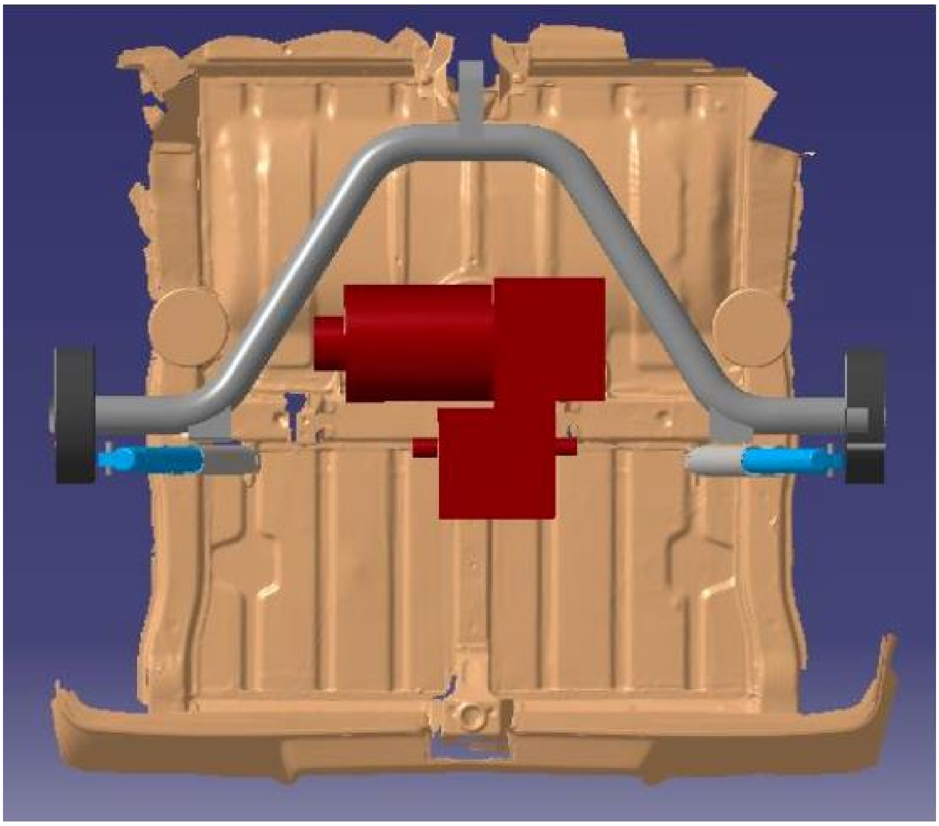


Figure 28: 3D model of the omega tube with the transaxle unit centrally positioned

One problem, which is evident in the 3D model, is that the original Ω tube interfere with the drive shaft passage towards the wheels. For this reason, a new shape of the tube should be designed with the introduction of new components such as new wheel hubs and a flange connecting the tube to the hubs, which by the way will be the object of study of the second part of this thesis.

2.5.1. Ω shape geometry modification

As previously said, the target of the modification of the geometry of the suspension tube is to avoid any interference between the tube itself and the drive shafts coming from the transaxle unit.

There is also a constraint to be respected to keep things as simple as possible: the most important points of the suspensions (the attachment to the frame which determine the geometry) should remain unchanged.

One exception is the position of the coil springs. In fact, since the height of the vehicle will be increased to improve its off-road characteristics, the position of the springs will be changed accordingly. Moreover, the shock absorbers are likely to be changed due to the different stroke required by the new arrangement, though their attachment points on the chassis will remain the same.

In Figure 29 and Figure 30 the new tube shape is sketched in yellow from different angles, while the red dots are the geometrical points to leave unchanged.

Figure 29: New suspension geometry (side view)

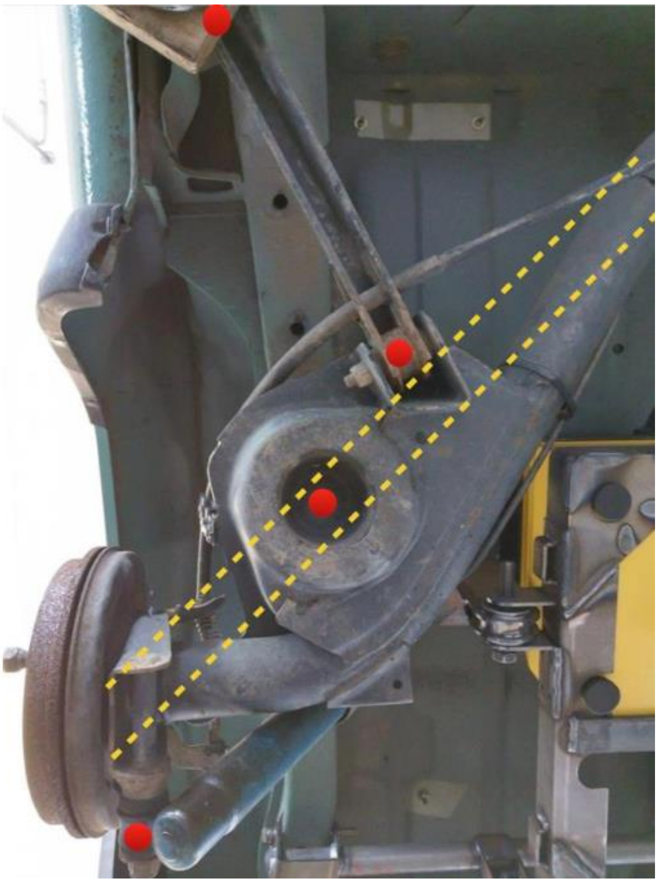


Figure 30: New suspension geometry (front and bottom views respectively)

Once measured the position of the geometrical points to keep fixed, with respect to a reference system placed at the central attachment of the tube to the chassis, the new geometry of the tube was sketched.

For the determination of the final shape, Italtecnica played a fundamental role giving advice about the manufacturability of the part.

In Figure 31 an upper view comparing the old and the new shape is proposed:

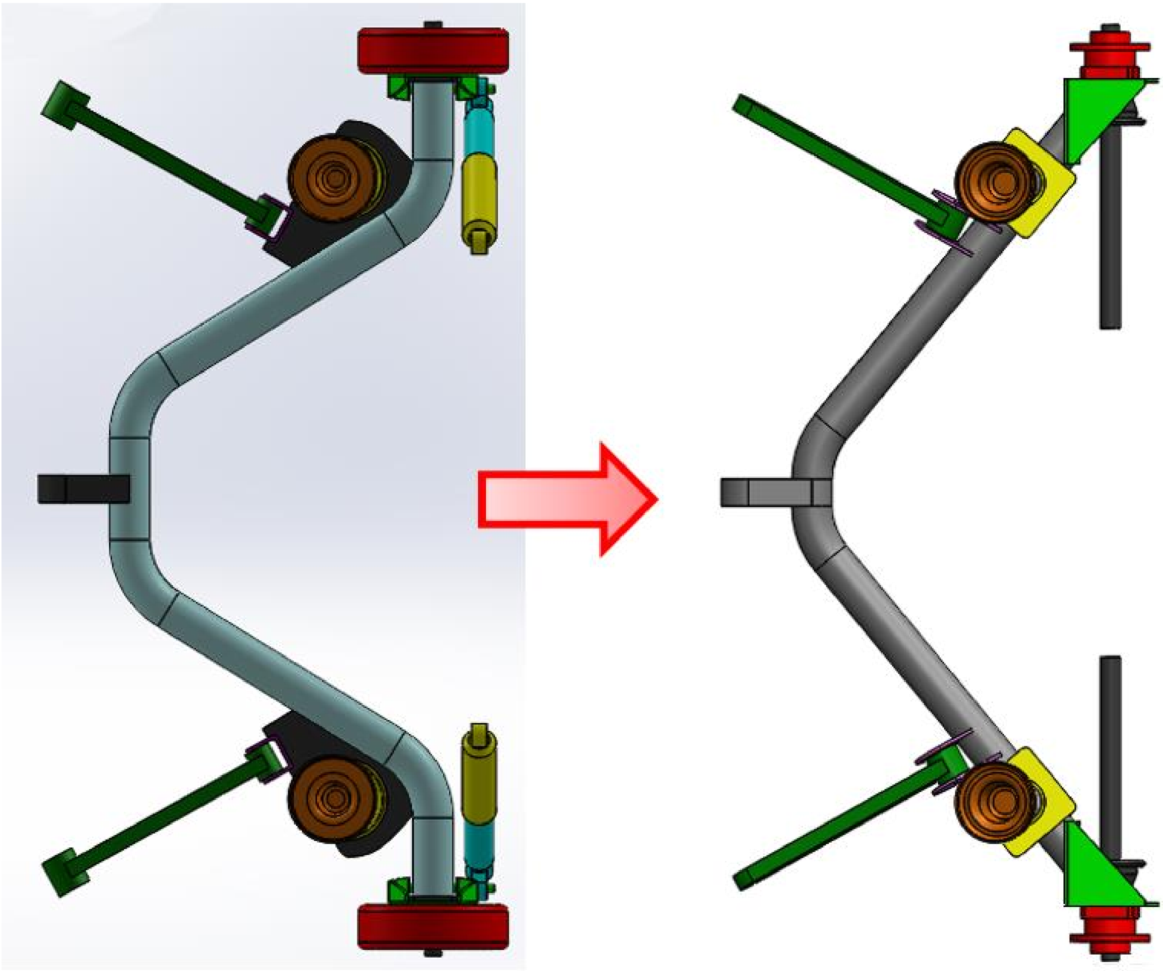


Figure 31: Old and new design (left and right respectively)

The main differences in the new design with respect the old one are:

- the tube has only one bend near the attachment point (less manufacturing cost);
- inclination of the tube for the passage of the drive shafts;
- introduction of a flange element to connect the tube to the new wheel hubs;
- no drum brakes (they will be substituted by disc brakes).

The idea for the wheel hubs is to use the same as the ones mounted on the front wheels (Figure 32) because they already contain the bearings and are designed to accommodate the drive shafts and to sustain the loads.

Figure 32: Front wheel hub

From the images it is clear that in addition to the flange, also new spring seats must be designed. This is a much simpler problem with respect to the design of the flange because the springs always work under compressive loads and, since the seats are attached to the upper part of the tube, the stresses are very low.

Another part to consider are the shock absorbers attachment points on the tube. A hole on the tube for the passage of the supporting pin will be required.

The most challenging work for this solution will be the design of the connecting flange, which were not present on the base vehicle and must sustain very high loads. Moreover, since it is an unsprung mass, it must be as light as possible, while complying with the available space.

2.6. Trade-off conclusions

After having presented all the possible modifications and after having discussed about all the hybridization layouts analyzing advantages and disadvantages of each one, the most adequate solution results to be the modification of the omega shape suspension system of the Panda 2WD vehicle.

The criteria considered for the choice are the following:

- convenience from the economical point of view;
- best arrangement for the components with the available space;
- construction and realization simplicity;
- resources and equipment available for the realization of the project.

3. DESIGN OF THE NEW REAR SUSPENSION SYSTEM

3.1. Creation of the 3D CAD model

3.1.1. Old suspension system

The starting point for the making of the CAD model of the new system was the suspension geometry of the base Panda, described in the previous chapter.

The first step was the measurement of the position of all the geometric points defining the constraints and the linkages of the old omega system. As previously mentioned, a target of the new design is to keep these points unchanged to save design time and money.

All the measures were taken thanks to the company, which has made available the necessary equipment to take precise and reliable measures.

The points under discussion are the following:

- Central attachment point of the Ω tube to the chassis;
- Attachment points of the springs on the chassis;
- Attachment points of the dampers to the chassis;
- Attachment points of the swinging arms to the tube;
- Attachment points of the swinging arms to the chassis.

Then there are the points modified due to change in the vehicle ground clearance:

- Attachment point of the springs on the tube;
- Attachment point of the dampers to the tube.

The reference system was placed in the central attachment point of the tube to the frame. X axis is parallel to the horizontal plane and it is directed on the longitudinal direction of the vehicle; Y axis is parallel to the horizontal plane as well but it is perpendicular to X; Z is oriented following the right hand rule.

Of these points, some are integral with the body while others are integral with the suspension tube, so they move relatively to the body.

As a consequence, before measuring the points, a scan of the vehicle underbody complete with the base Panda suspension was used as a reference point to build the new model. This scan (Figure 33) shows the full range of motion of the suspension, from the maximum compression configuration to the maximum extension with no load on the springs.

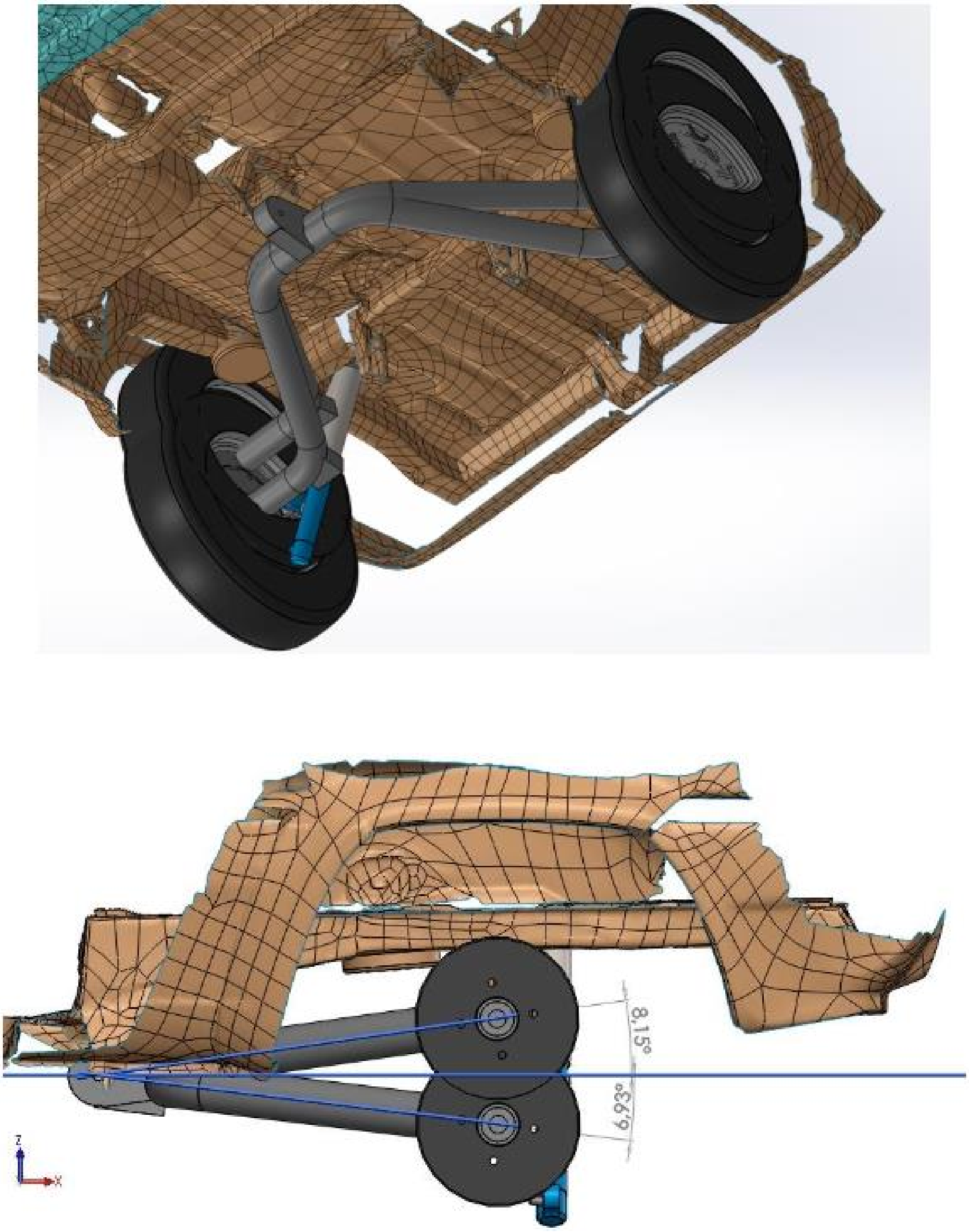


Figure 33: Scan of old suspension with range of motion

The horizontal plane crossing the suspensions in Figure 33 corresponds to the vehicle in running order, so it is the normal condition that will be considered for the new design. When the suspensions are fully loaded the angle formed by the line connecting the central point with the wheel axis and the horizontal plane is 8,15°. On the other hand, when the suspensions are fully unloaded the angle is 6,93°. The total range of motion of the suspension is then 15,08°.

Since the position of some points depend on the operating conditions of the suspension, a reference condition is needed. For sake of simplicity it was chosen the position of the normal running order of the car, with the tube perfectly horizontal.

In Figure 34 and Table 13 the points are shown once again, with the respective coordinates:

Figure 34: Remarkable points of the suspension

	x [mm]	y [mm]	z [mm]
Tube central attachment	0	0	0
Right swinging arm - tube	292.4	402	-42.9
Left swinging arm - tube	292.4	-402	-42.9
Right swinging arm – chassis	0	565	0
Left swinging arm - chassis	0	-565	0
Right spring seat - tube	409.8	458	-50.3
Left spring seat - tube	409.8	-458	-50.3
Right spring seat – chassis	393.3	456.6	142.3
Left spring seat - chassis	393.3	-456.6	142.3
Right damper - flange	607.6	588.5	-95
Left damper - flange	607.6	-588.5	-95
Right damper – chassis	607.6	363.2	204.2
Left damper - chassis	607.6	-363.2	204.2

Table 13: Coordinates of geometrical points. In red color are the ones that will change due to change in vehicle height

Building of the model – Once taken all the principal measures for the construction of the components, it was possible to realize the CAD model of the suspension system of the base Panda.

In Figure 35 it is possible to see the details of the spring seats supported by the flanges that will be modified in the new design.

Moreover, as previously said, the tube will not be attached to the drum brake anymore. In fact, the green flange connecting the tube to the drum, will be substituted by a structural flange of fundamental importance for the secure connection between the tube and the new hubs. The new flange will not support the damper anymore, but a hole will be drilled directly into the tube.

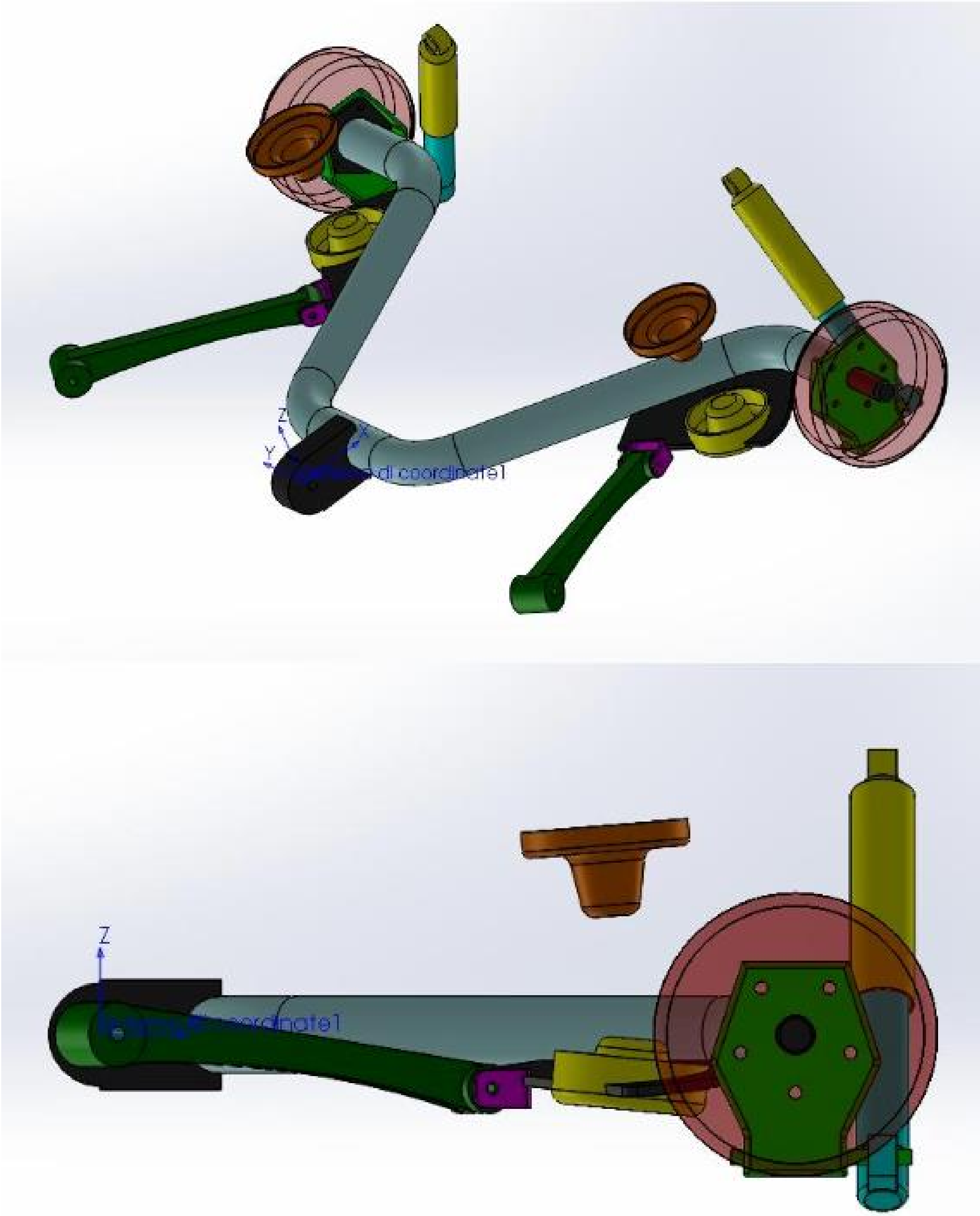


Figure 35: Old Panda suspension system

In Figure 36 and Figure 37 some detailed views are presented.

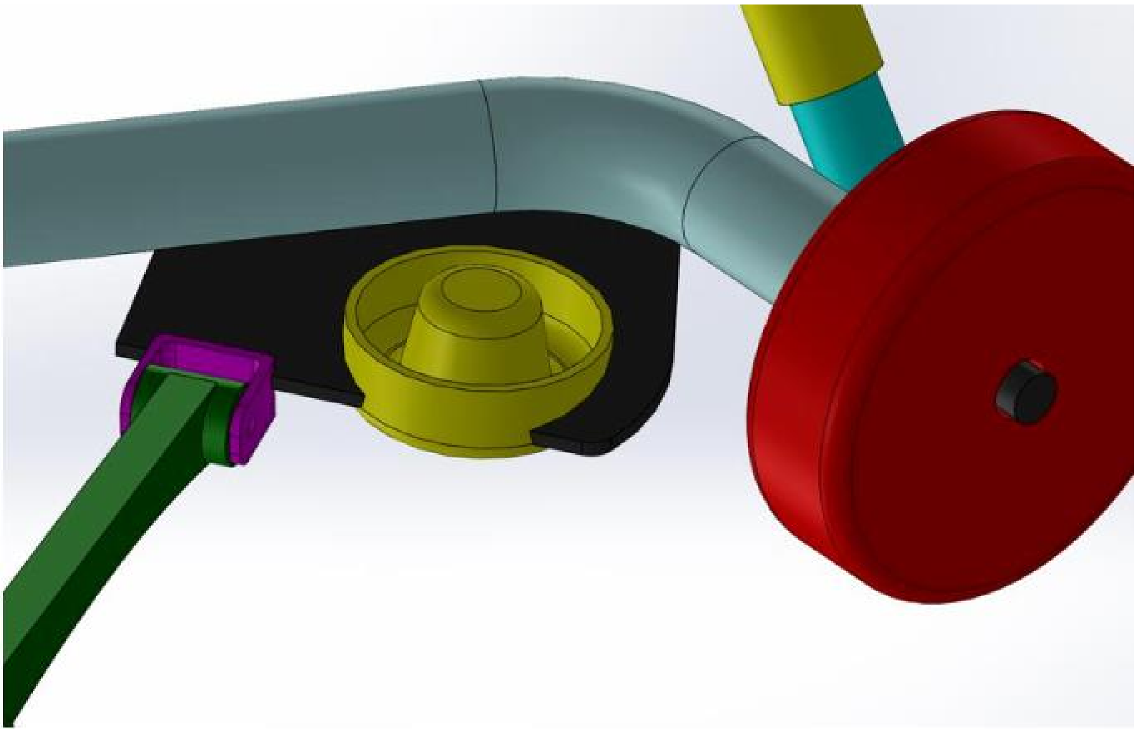


Figure 36: Spring seat and swinging arm attachment to the tube with a flange

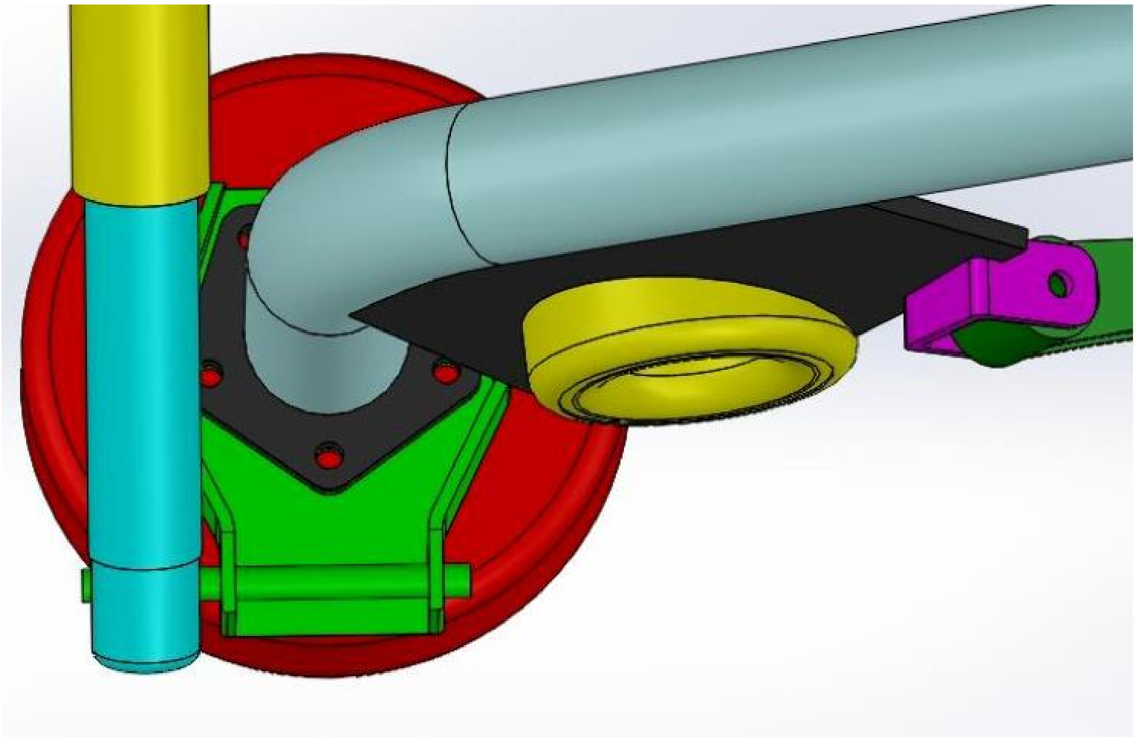


Figure 37: Detail of the flange connecting the tube to the drum and supporting the damper

3.1.2. New suspension system

In the CAD model of the new suspension system is illustrated. Summing up, the two constraints for the making of the model were:

- Find a new tube geometry that allows the passage of the drive shafts;
- Maintain the same geometry as before, changing only the points to accommodate the new vehicle trim.

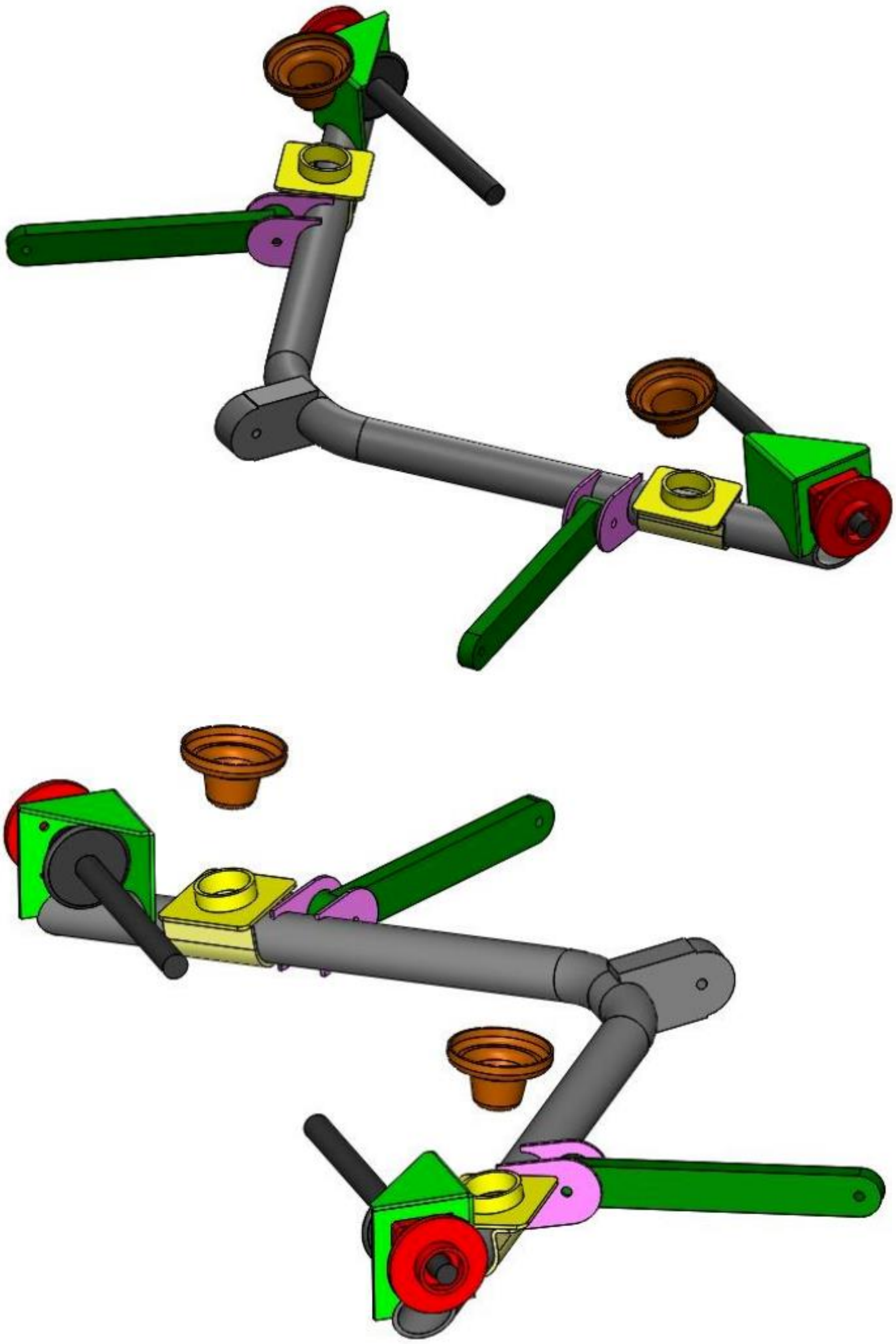


Figure 38: New suspension system

Tube – The geometry of the tube was modified so that it has just one bend near the attachment point, for manufacturing simplicity. Another difference, and the most crucial one, is that now the two ends of the tube do not finish on the center of the hub but are at a certain distance below, to allow the passage of the drive shafts. For this reason, the new flange (shown in green) was designed: as a structural connection between the tube and the hubs.

The dislocation of the tube is the result of a rotation of the tube itself of about 8°, as shown in Figure 39. As explained previously, the starting condition is the one of the vehicle in running order, where the wheels are horizontally aligned with the central attach point of the tube. The rotation concerns only the tube and all the other points are unchanged.

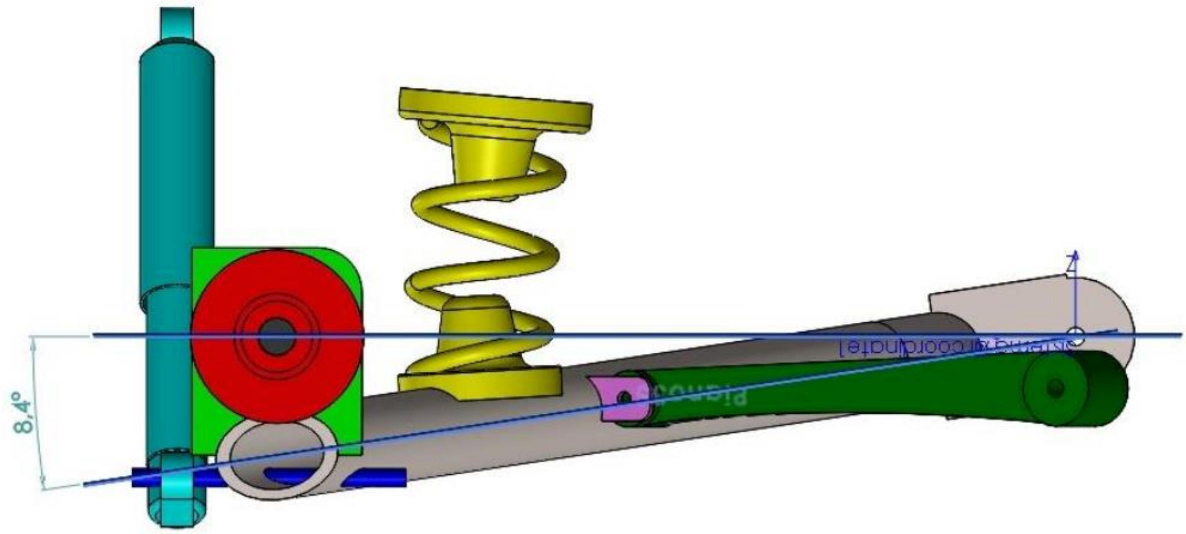


Figure 39: Inclination angle of the new tube design

Swinging arms – The horizontal tract of the tube near the attachment point is removed, anticipating the rotation towards the wheels and leaving a sufficient space for the attachment of the elastic bushings of the swinging arms. Without this arrangement, the attachment points of the bushing would have been inside the tube, leading to an expensive modification of the tube itself. As an alternative the arms would have been shortened changing the geometry of the suspension, complicating a lot the design. The tube then will not have a Ω shape anymore, but it will be V shaped.

Coil springs – With the new geometry the seats of the springs have been raised of about 30mm as not to interfere with the new tube. The seats now are in the best position possible being exactly on the top of the tube. This new arrangement has the following advantages:

- The spring has been raised of about 30 mm which corresponds to the quantity that the height of the vehicle has to be increased;
- Being on the top of the tube and working only under compressive loads means that the stresses on the component are very low and it is not necessary to study and dimension the part very accurately.

In Figure 40 a detail of the spring seat is shown. Notice the position on top of the tube. The beige color flange supporting the seat is actually oversized because not much reinforcement is needed to sustain the load.

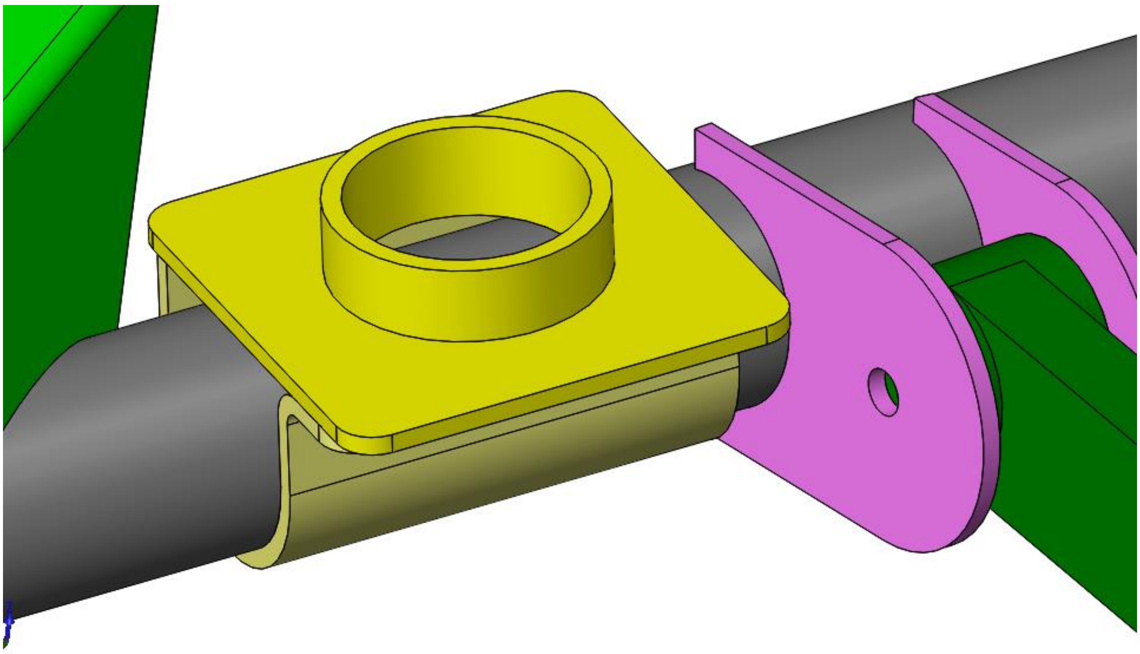


Figure 40: Spring seat detail

Shock absorbers – In the old layout the shock absorbers were supported by a pin attached to a flange linked to the tube and to the drum. In the new system the old flange has been removed and the tube has been lowered. The idea then is to drill a hole through the tube and insert the pin supporting the damper. This time though, the inclination of the pin will be different to guarantee an optimal behavior of the elastic bushing connecting the damper (Figure 41).

Moreover, due to the change in vehicle height, the shock absorbers will be replaced with longer ones, but maintaining the same dynamic characteristics as before.

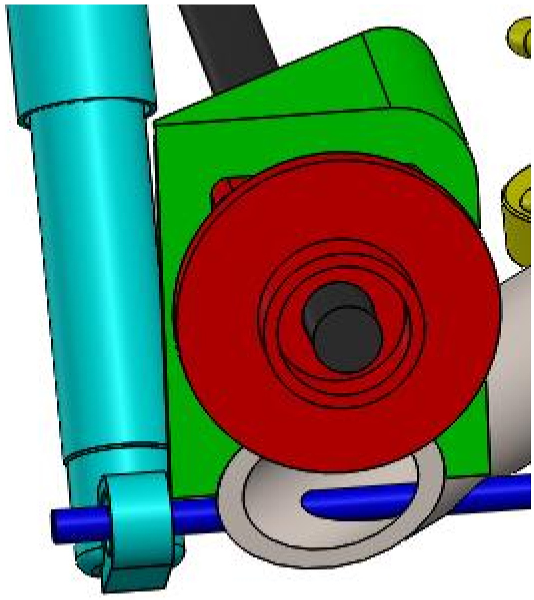


Figure 41: Detail of the pin connecting the damper to the tube

Wheel hub – In the base 2WD Panda, being the traction on the front wheels, the rear suspension system is not designed for traction purposes. In particular, as previously seen, the ends of the **Ω** tube finish directly at the center of the brake drums, that cannot be coupled with a drive shaft. For this reason, it was necessary to substitute the drum with a wheel hub capable of transmitting power. The choice was to use the same hubs mounted on the front wheels. Their advantage is that they have the bearings integrated so no design was required. In Figure 42 a comparison between old and new hubs is made.

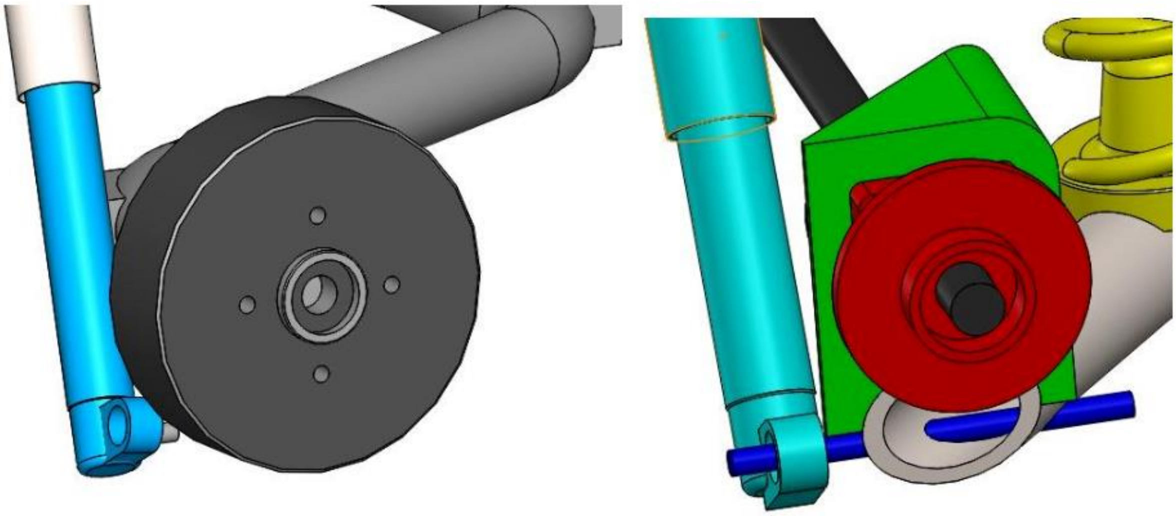


Figure 42: Old wheel hub (left) compared with new wheel hub (right)

Flange – The flange linking the tube and the wheel hub is the object of the remaining part of the work. It is the component which was designed from scratch and that has a fundamental role in the whole suspension system of the new vehicle. Many different shapes have been sketched trying to accommodate all the constraints, while simultaneously as many structural analyses have been carried out to evaluate the best geometry in terms of lightness and strength. In Figure 43 are shown the main geometries tested.

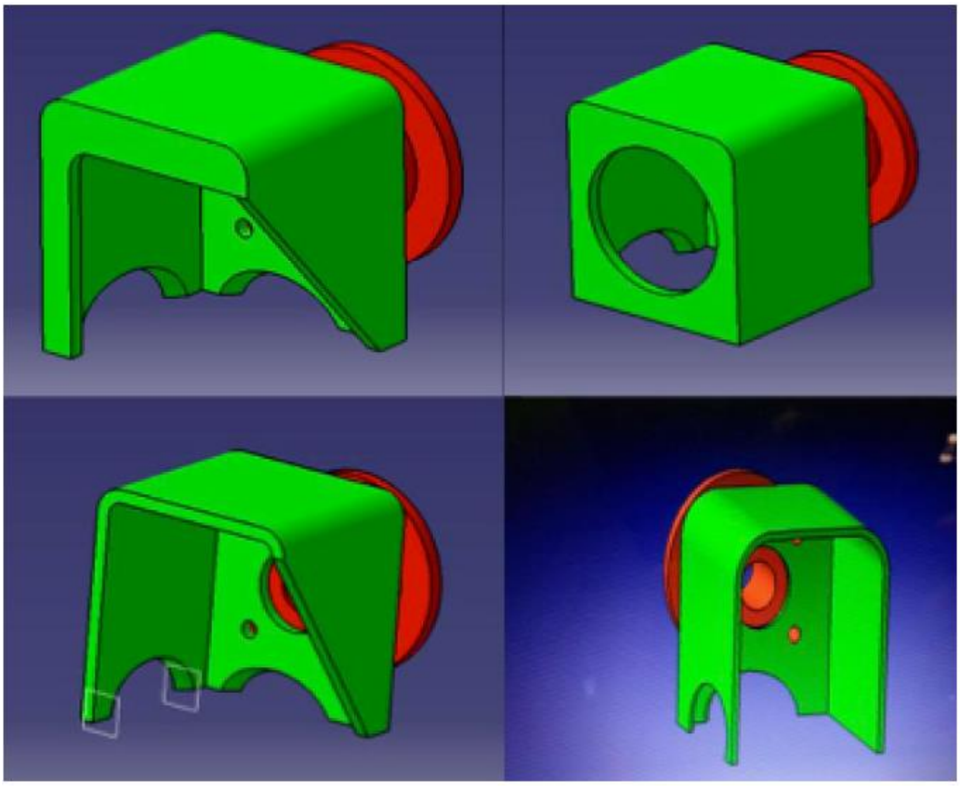


Figure 43: Different geometries for the flange

The design complexity of this component emerges from the following factors:

- Critical zone in terms of loads: the forces are applied to the hub at a certain distance from the flange so very high moments arise causing high stresses;
- Size problems: since the homokinetic joint has a large diameter and some clearance around it has to be considered, the flange must be quite big. On the other hand, there are limits on the size given by the wheel rim and by the stiffness of the flange itself.
- The mass of the flange should be kept as small as possible because there is already an increase of the mass due to the installment of the motor-gearbox on the rear axle.

For these reasons it was decided to set a topological optimization study to obtain an optimal geometry representing the best trade-off between stiffness, lightness and mechanical strength. This study is explained in detail in a dedicated chapter. As a conclusion of this chapter, in Figure 44 are shown the final results of the study.

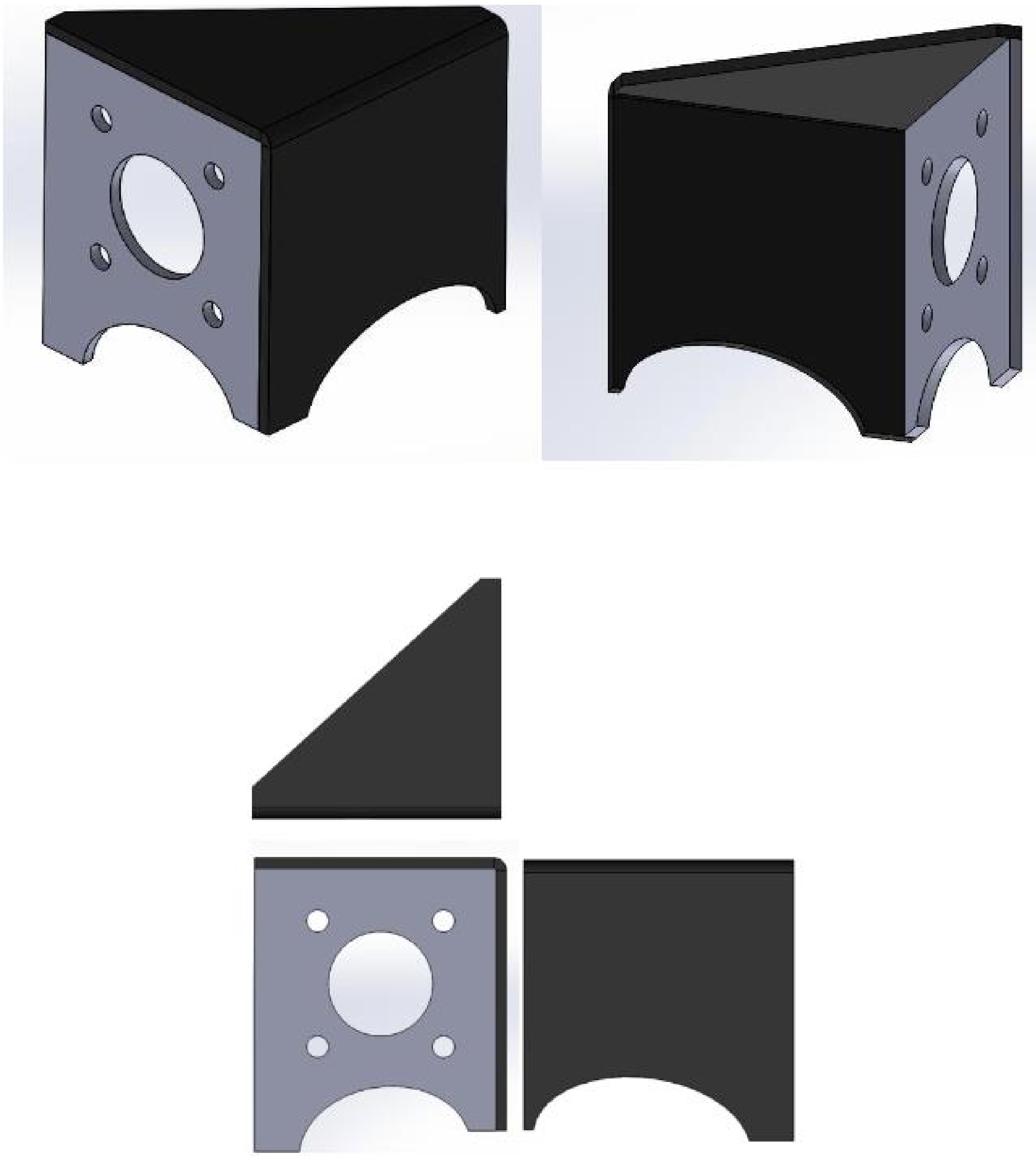


Figure 44: Isometric and orthogonal views of the final version of the flange

3.2. Multibody model for the determination of the loads

Once having defined the new suspension geometry, the shape of the flange connecting the tube to the wheel hub and once having defined the CAD models, a multibody model of the suspension system was created using the software Adams View. This model will be used to determine the loads affecting in particular the flange, under the most critical operating conditions.

The realization of this model is finalized to build the elasto-kinematic behavior of the suspension, modelling the linkages between the components and defining their relative motions. Finally, data regarding the characteristics of the springs and dampers will be inserted to obtain an accurate model.

The next step will be the building of an accurate model of the vehicle body, at which there will be attached the front and rear suspensions. Starting from the CAD of the frame, some masses representing the passengers, the engine, the motor-gearbox and the battery pack will be placed correctly inside the vehicle creating an accurate representation of the car in running order. After having inserted the masses inside the vehicle it will be possible to know some fundamental characteristics like the center of mass and the polar moments of inertia.

Once the complete model is ready, the last step before launching the simulations is to define the input to apply to the vehicle, to trace back to the loads.

This input is constituted by another CAD model imported into the software. It is the representation of a typical off-road profile made by two different parts. They simulate the most difficult conditions that the vehicle will encounter during the Panda Raid, testing the resistance of the components previously discussed.

At this point the post-processing phase starts. The obtained results will be analyzed so to validate the elasto-kinematic model. Next, this output data will be used as inputs for the structural analysis of the flange in the following chapter.

3.2.1. Creation of the model

The vehicle – As already anticipated, the model of the vehicle is a CAD sketch of the entire frame, including the four wheels, as shown in

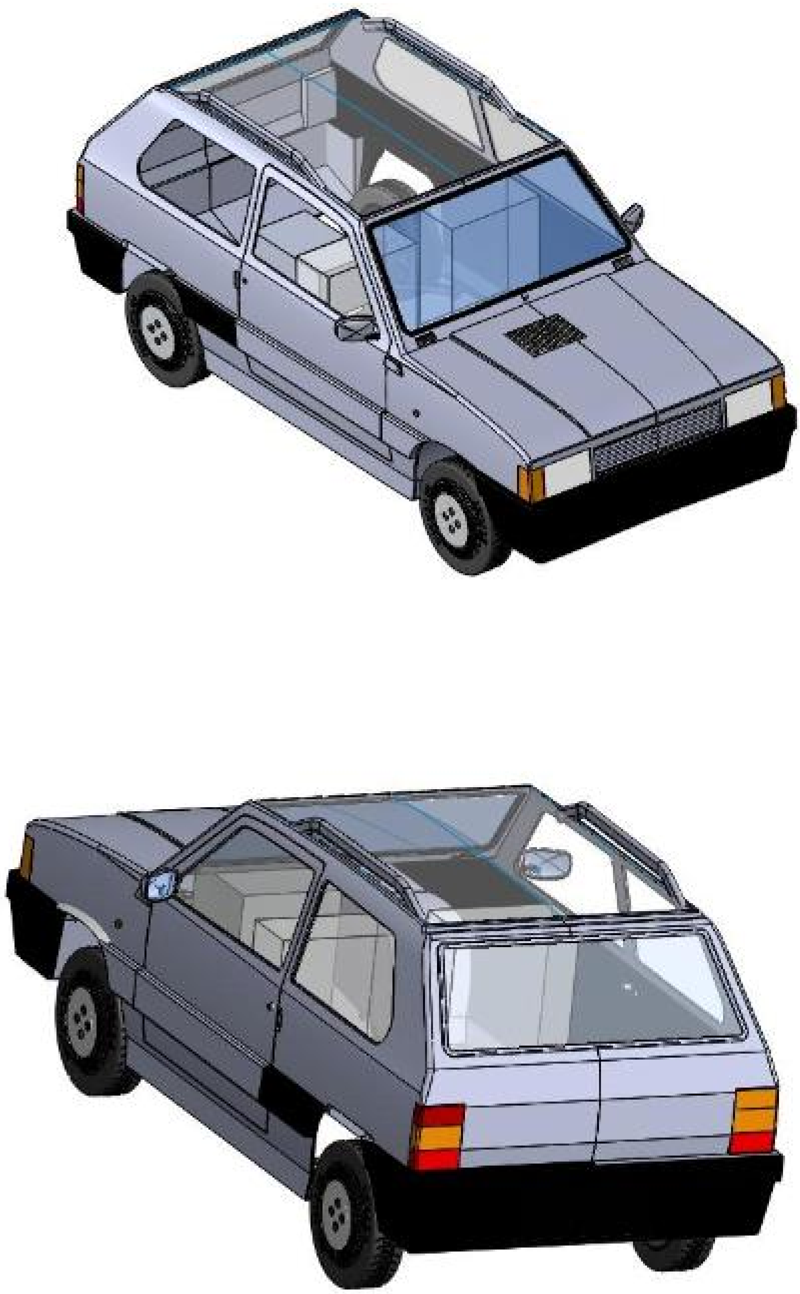


Figure 45: CAD model of the Panda

At this point, to obtain a realistic model, it was necessary to make an analysis of the weight distribution considering all the massive parts of the vehicle. In fact, it was not possible to establish a center of mass and the moments of inertia a priori.

All the massive parts added to the frame are listed below:

- Passenger compartment: two blocks representing the passengers, the seats, and the front accessories.
- Front part: one block representing the internal combustion engine with transmission and differential, the steering system and the drive shafts.
- Rear part: battery pack, tank, electric motor-gearbox unit, rear drive shafts and other structural parts.

It was used the workshop manual to take the information relative to the weight of the components [1].

For what the internal parts is concerned, they were considered evenly distributed in the whole frame for sake of simplicity. So, the weight of the body has been increased until the weight of the vehicle reached the correct value found in the manual.

Moreover, having designed the suspension system and having assigned the correct material to each component, it has been possible to define also the weight of the whole suspensions.

The center of the reference system was put at the attachment point of the tube to the chassis, with x axis directed along the longitudinal direction of the vehicle, y axis in transversal direction and z axis directed upwards.

In Table 14 all the added components are summarized, each one with its own weight and center of mass coordinates with respect to the reference system chosen.

	Peso [kg]	x [mm]	y [mm]	z [mm]
Right passenger	75	-460	350	450
Left passenger	75	-460	-350	450
Front part	250	-1680	0	280
Rear part	140	330	0	180
Body and interiors	350	-500	0	300
4 Wheels	64	540	0	0
Rear suspensions	42	400	0	0
Right MacPherson	18	-1630	610	0
Left MacPherson	18	-1630	-610	0

Table 14: Weights and center of masses of the added massive parts

After having inserted the data of each component and having positioned all of them inside the vehicle model, through the CAD software the essential data regarding the whole vehicle were determined. Of particular interest are the total weight of the car, the position of the center of mass of the car and its polar moments of inertia along the principal axis. In Table 15 this data are summed up while in Figure 46: Vehicle model with center of mass highlighted in redthe center of mass is represented with a red dot.

X_{CM} [mm]	Y_{CM} [mm]	Z_{CM} [mm]	I_{XX} [kg mm^2]	I_{YY} [kg mm^2]	I_{ZZ} [kg mm^2]	Weight [kg]	Weight distribution
-690	50	300	$2 \cdot 10^8$	$1,05 \cdot 10^9$	$1,15 \cdot 10^9$	1030	57:43

Table 15: Center of mass, weight, moments of inertia and weight distribution of the whole vehicle model

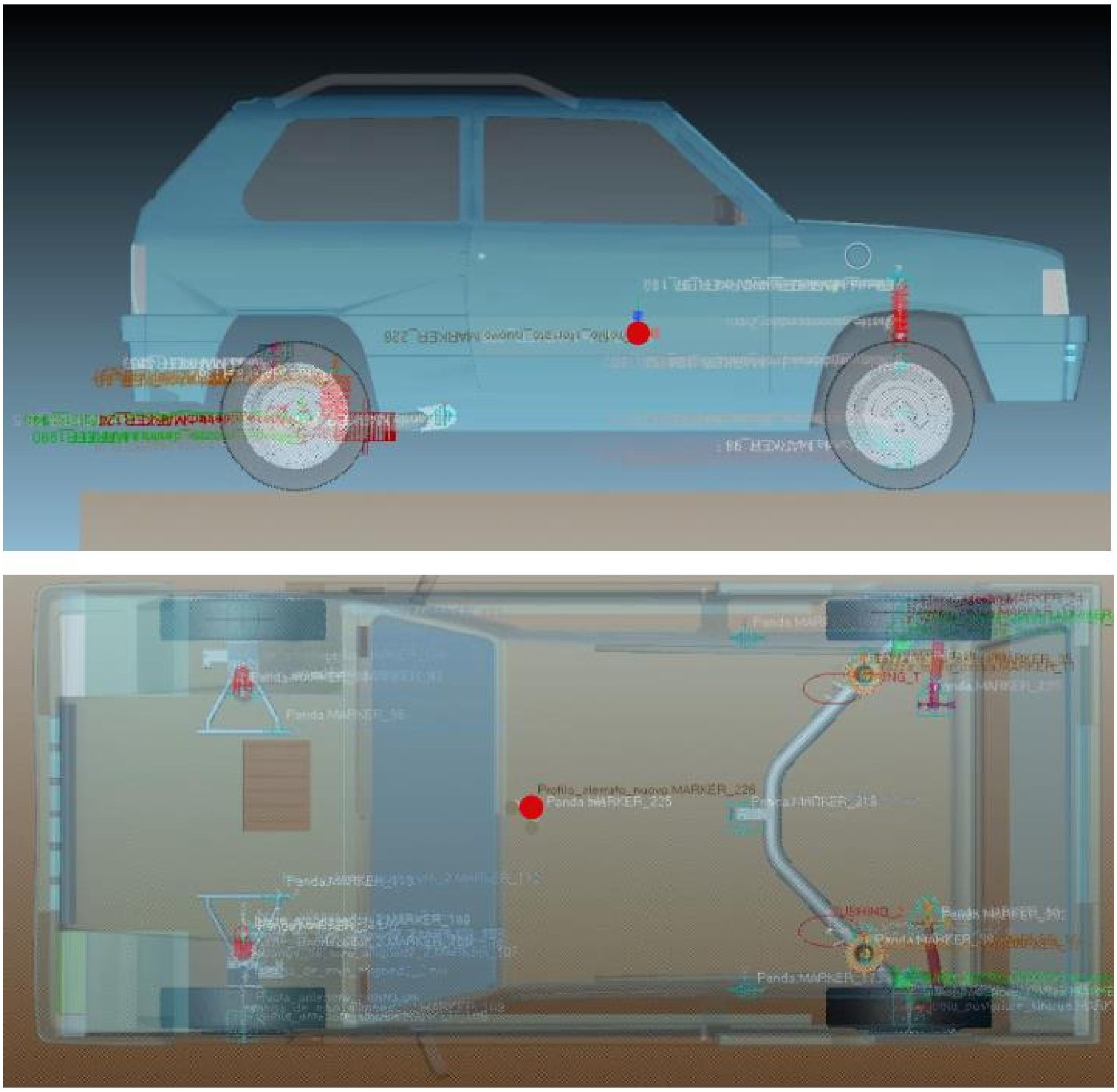


Figure 46: Vehicle model with center of mass highlighted in red

Suspensions model – With the expression “multibody system” it is indicated a system constituted by a set of rigid or flexible bodies connected to each other by means of kinematic joints that limit their relative motion [2], [3].

The rear suspension, being already designed in Solidworks ambient, has been imported to Adams View as a set of components, listed below, forming the suspension system:

- suspension tube;
- swinging arms;
- spring seats (both on the tube and on the chassis);
- connecting flanges;
- wheel hubs;
- coil springs and dampers.

In this model the transaxle was not considered because it is not important for the kinematic behavior of the suspension, apart from the drive shafts which are coupled with the hubs. However, since they are lightweight components and they do not increase the load on the supporting flange, which is the critical component to be studied, they were excluded. The important thing considered about the transaxle unit was its mass and its polar moments of inertia, that are very important for the determination of the suspension loads.

To complete the model, a reference system oriented as explained before was placed and the earth’s gravitational field was added.

At this point it was possible to start with the definition of all the linkages among all the different components described above.

In Figure 47 and Figure 48 the multibody model in Adams View ambient is shown. The brown ground is the profile simulating the off-road terrain, given as input to the wheels.

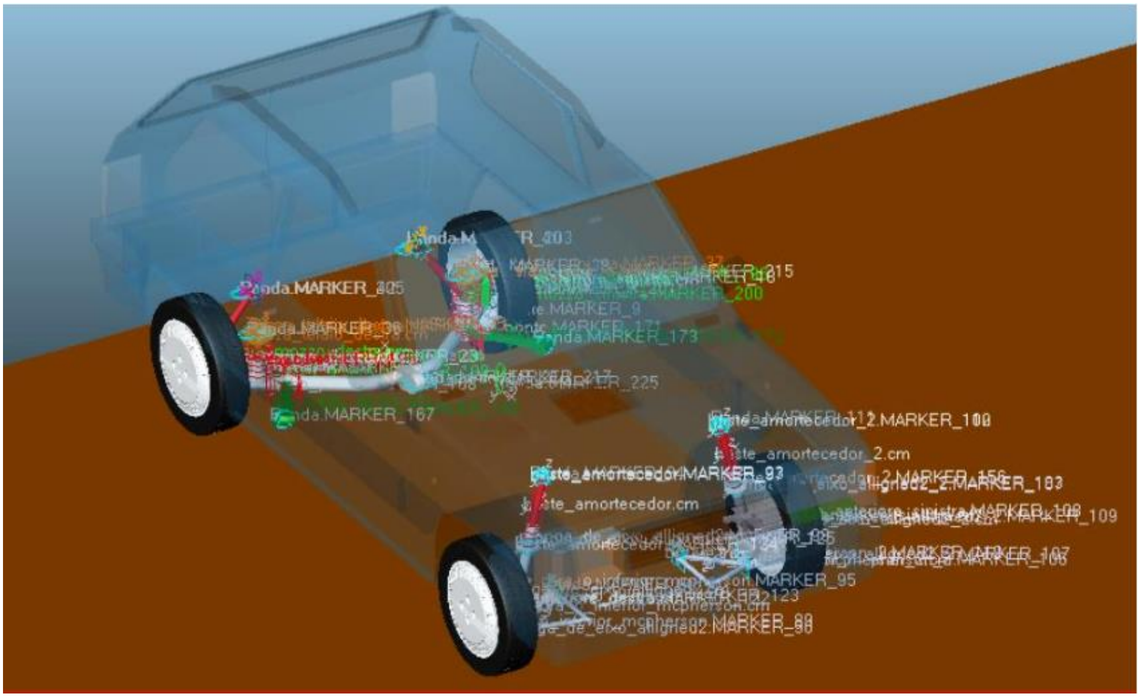


Figure 47:Multibody model

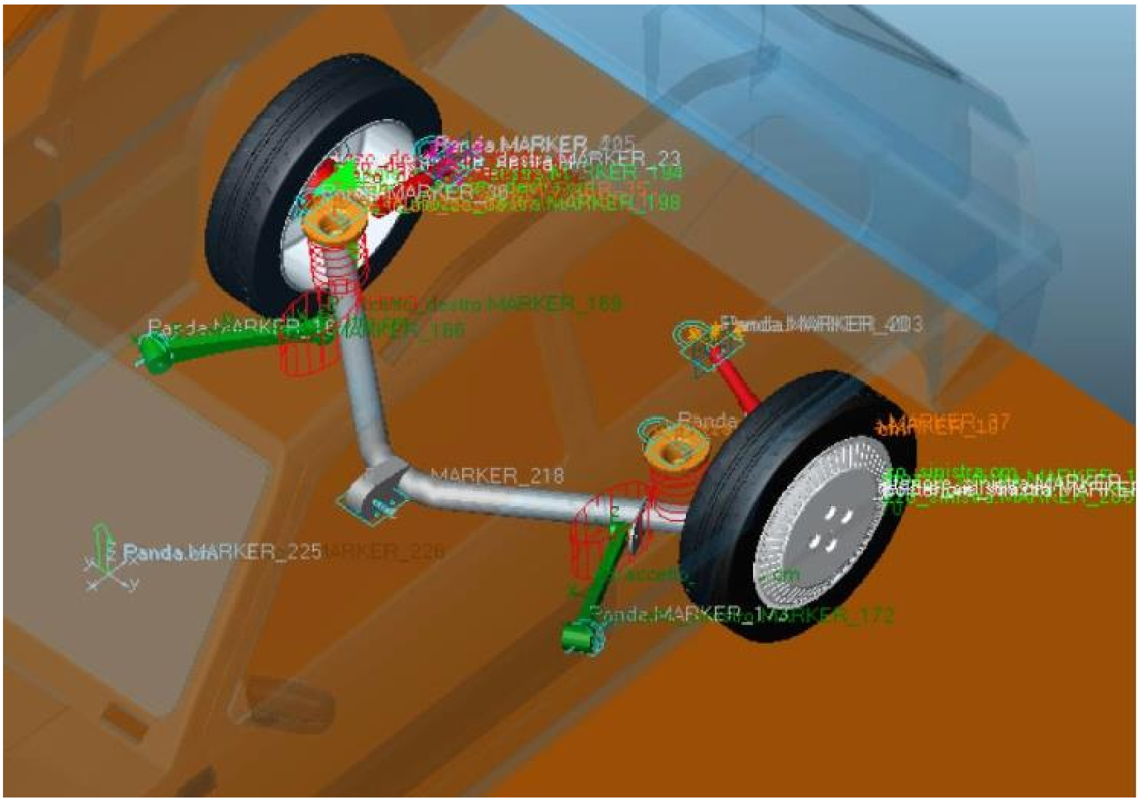


Figure 48: Detail of the rear suspension system

The tube is centrally attached to the frame by an elastic bushing that allows just one degree of freedom which is the rotation about y axis, so the linkage was modelled using a revolute joint (Figure 49).

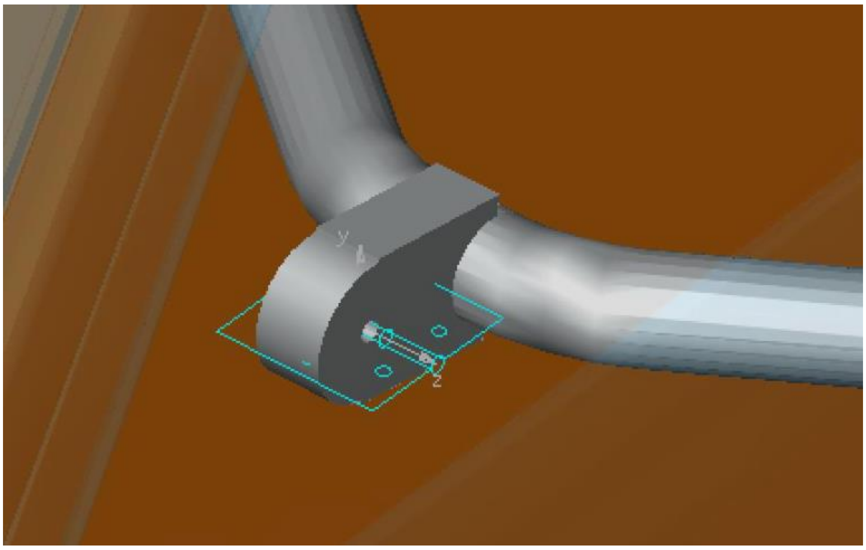


Figure 49: Detail of the central attachment point of the tube to the chassis

The spring seats have been merged to the tube using the join solids function to avoid the introduction of other functions connecting more parts.

The attachment of the springs to the chassis was modelled using a fixed joint, being them totally constrained to the frame (Figure 50).

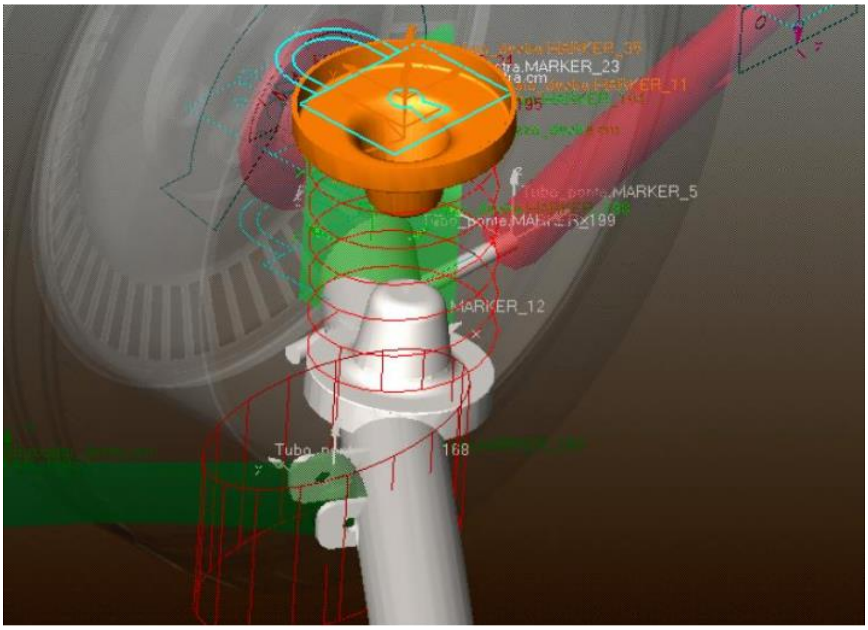


Figure 50: Detail of the spring attachment both on the tube and on the chassis

The swinging arms have been connected at one end to the vehicle frame and on the other end directly to attachments created on the tube. Both extremities have elastic bushing linkages. The chassis attachment was modelled by using a revolute joint, while the attachment to the tube was defined as a “bushing”, meaning a linkage on which it is possible to define a degree of stiffness for the six degrees of freedom. To avoid making swinging arm an hyperstatic element, the stiffness related to the rotation about the attachment axis was removed, to keep the correct kinematic behavior. The other degrees of freedom simulate the elastic bushing movements (Figure 51).

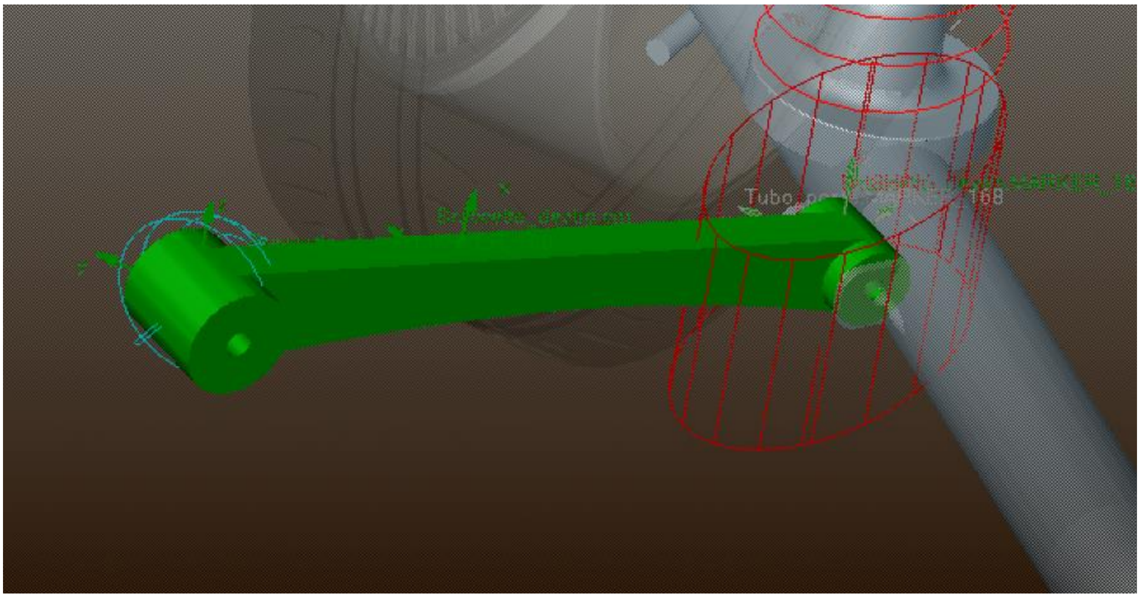


Figure 51: Detail of the swinging arm attachment points

Let’s move now to the flange connecting the tube to the wheel hub. It is welded to the tube along all the lower part to have a big contact area and make the link more solid. As a consequence, a fixed joint was defined between the flange and the tube, being it a linkage without degrees of freedom.

The flange is then attached to the wheel hub by means of a bolted connection that does not allow any movement. For this reason, it was used a fixed joint connection also here (Figure 52).

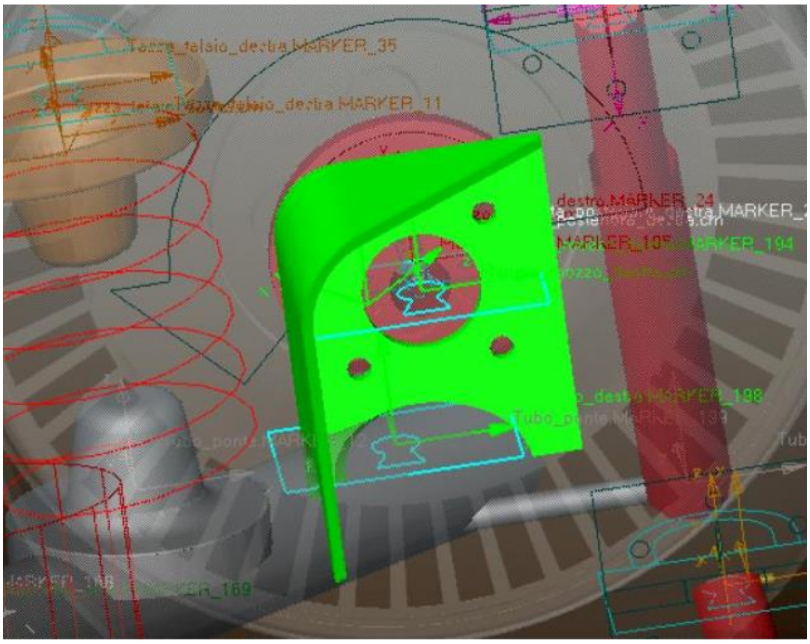


Figure 52: Detail of the flange in the multibody model

The hub then is connected, by means of the brake drum, to the wheel. A revolute joint was inserted between the hub and the wheel to model the wheel rotation through the bearings. To insert the parameters related to the suspension geometry like the camber angle and the toe angle, it is possible to orient the reference system of the joints in a totally arbitrary way, modifying the vehicle settings (Figure 53).

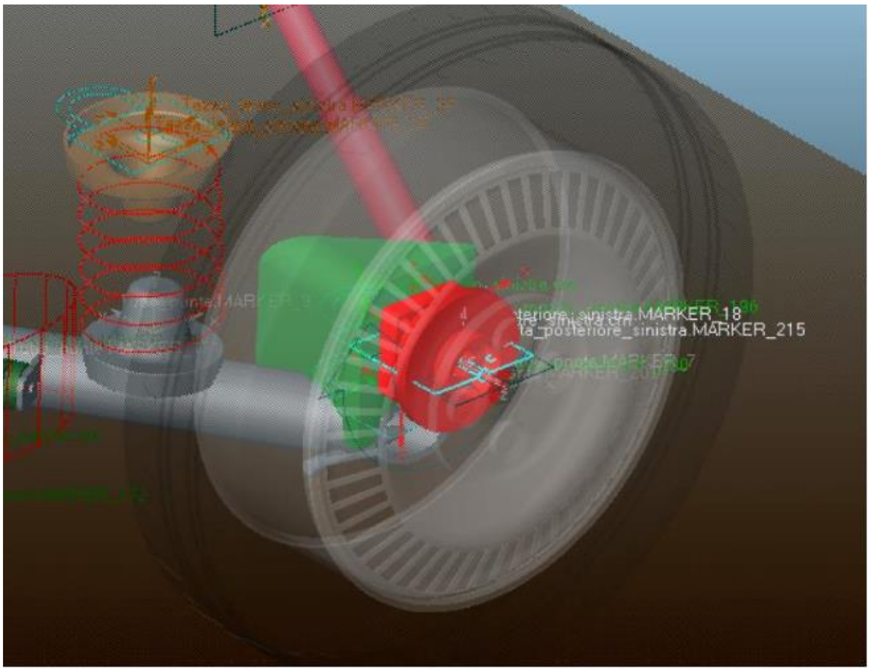


Figure 53: Detail of the coupling between the wheel and the hub

As regards the front suspensions model, also in this case a CAD model was imported as a set of parts that have been accurately linked together and to the chassis.

The wheel hub, included in the strut, was linked to the wheel in a similar way to the rear suspension. Also in this case the presence of the drive shafts was not considered but only the weight was kept into account in the steps described previously.

The strut has been linked to the swinging arms by means of a spherical joint around which they can rotate.

The swinging arms are then linked to the chassis using a revolute joint, since they are free to rotate about an axis parallel to the x axis.

Finally, the strut has been linked to the spring-damper system that is attached to the chassis by means of a spherical joint (Figure 54).

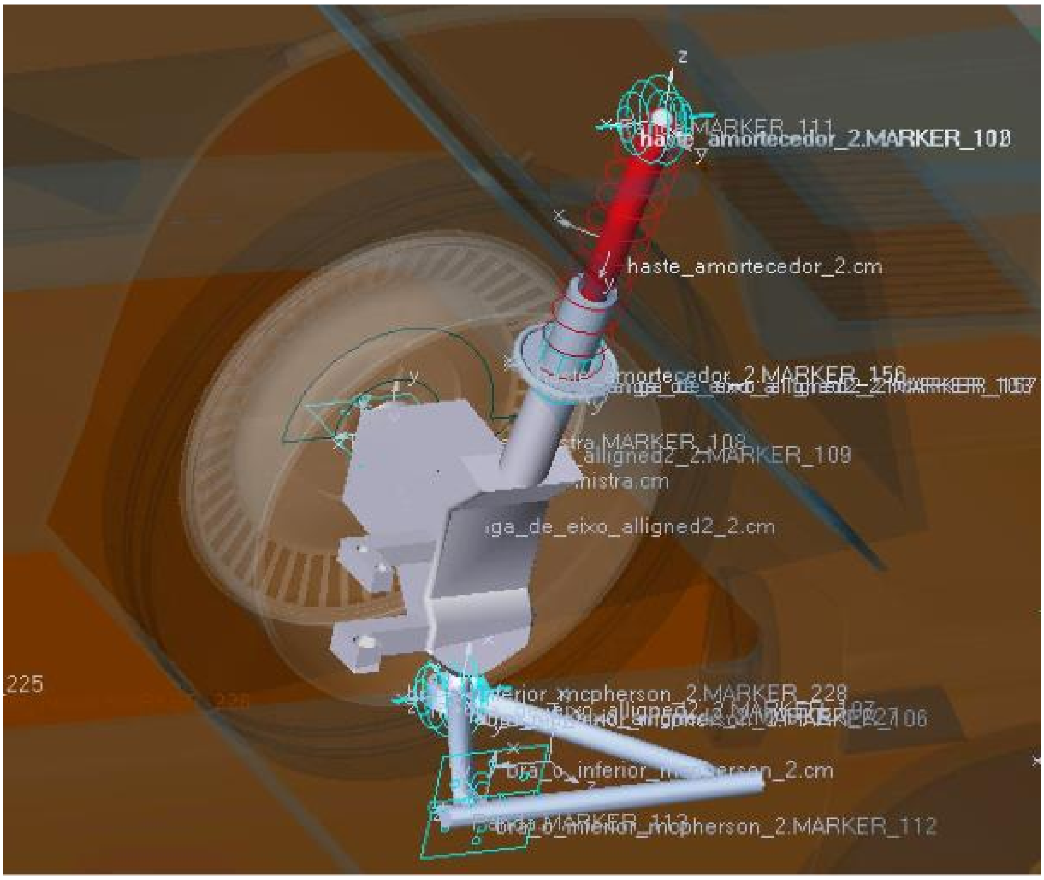


Figure 54: Detail of the front MacPherson suspension model

Coil springs and dampers – The elastic and damping components have been introduced in the model using default functions found in the “forces” section, where it is possible to apply various types of loads and to insert elastic and damping elements with their technical characteristics [2], [3].

In particular, for what the springs and dampers is concerned, the values reported in the manual have been considered as reference and have been set as values of the functions. In Table 16 the values used are listed.

Molle ad elica		769 1000	1000 4x4
Altezza molla sotto un carico di:	208,5±8 daN mm	213	–
	241±10 daN mm	–	231
Le molle sono suddivise in due categorie, identificabili mediante contrassegno:			
giallo (1) quelle aventi sotto un carico di:	208,5±8 daN un'altezza di mm	>213	–
	241±10 daN un'altezza di mm	–	>231
verde (1) quelle aventi sotto un carico di:	208,5±8daN un'altezza di mm	⩽213	–
	241±10 daN un'altezza di mm	–	⩽231

Ammortizzatori		769 1000	1000 4x4
Tipo: telescopico, a doppio effetto		Way-Assauto	
Aperto (inizio tamponamento)	mm	428,5 ± 2	438,5 ± 2
Chiuso (ferro contro ferro)	mm	282,5 ± 2	292 ± 2
Corsa	mm	146	146,5

Molle ad elica		769 1000
Altezza molla sotto un carico di 257 daN	mm	170
Le molle sono suddivise in due categorie, identificabili mediante contrassegno: giallo (1) quelle aventi sotto un carico di 257 daN	un'altezza di mm	>170
verde (1) quelle aventi sotto un carico di 257 daN	un'altezza di mm	⩽170

Ammortizzatori		769	1000	1000 4x4
Tipo: telescopico, a doppio effetto		Way-Assauto		
Aperto (inizio tamponamento)	mm	420 ± 3		340 ± 3
Chiuso (ferro contro ferro)	mm	253 ± 3		195 ± 3
Corsa	mm	168		145

Table 16: Characteristic values of spring and dampers

Contact between wheels and ground – To make possible the relative motion between the vehicle and the ground profile a correct contact linkage was defined among the wheels and the ground itself. With this function a solid to solid contact is defined, specifying all the parameters that governs the contact itself.

The tire con be modelled like a spring-damper system. In fact, to simulate the rubber behavior some stiffness and damping parameters have been introduced, according to the type of wheels used for the competition.

Moreover, some static and dynamic friction parameters have been introduced (Figure 55).

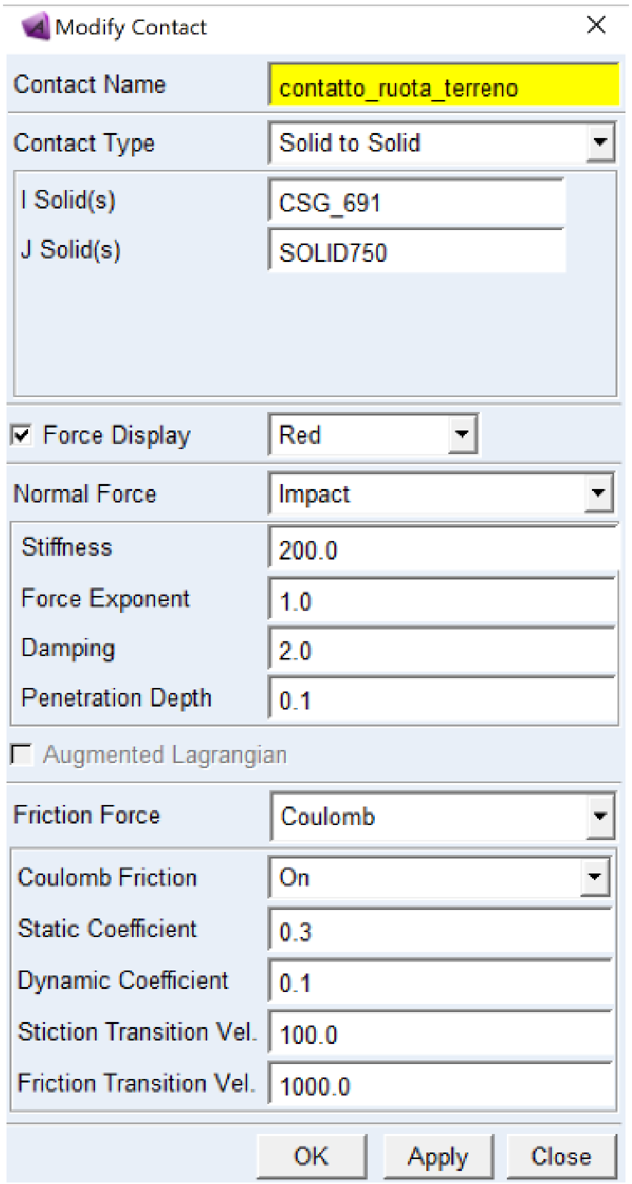


Figure 55: Parameters defining the tire-grond contact

Ground profiles model – To evaluate the loads applied to the suspensions in the most critical conditions, in addition to all the data inserted in the previous passages, the system needs some inputs. A secondary purpose is to check that the suspension kinematic works correctly, though the geometry has not changed so the suspension should work as in the base model.

For this study the choice was to design two sample routes with two different profiles to represent the most critical working conditions for the suspensions. Such profiles, having already determined the wheel-ground contact, constitute the input for the suspension motion, that will result in a load spectrum applied to the system.

The reason for the choice of these input profiles is that the suspension is more loaded when its stroke is very large and sudden, rather than during the vehicle acceleration and braking. For example, the suspension is very stressed where there are many bumps, rocks and other obstacles. Therefore, the final choice for the profile is shown in Figure 56.

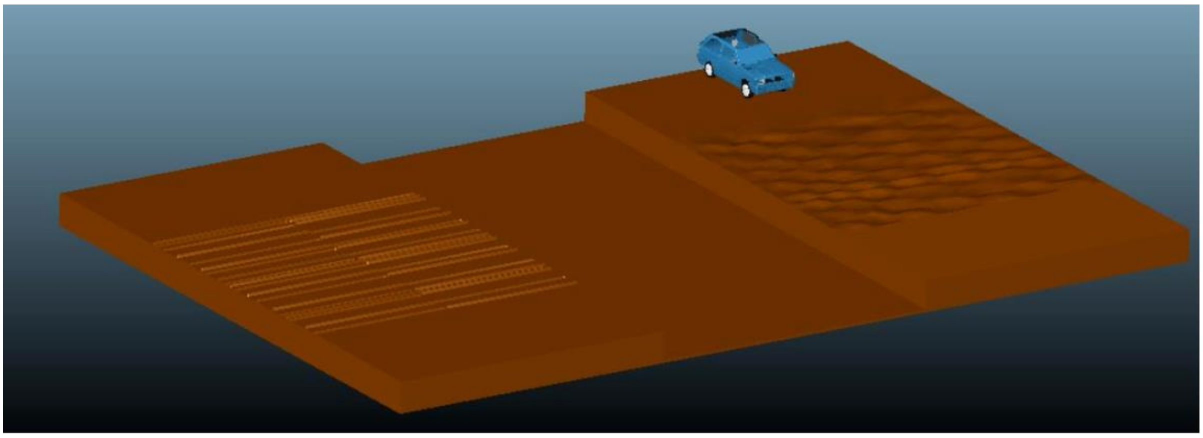


Figure 56: Road profiles model

Profile 1: this profile is characterized by the presence of steep bumps and holes alternated at a variable distance, to make the suspension work in both ways, compression and extension. To make it even more critical, the profile is made so that when one wheel hits a bump, the other wheel of the same axle hits a hole, to have one part of the suspension compressed and the other part extended.

In particular, the bumps are 10 cm high and the holes are 10 cm deep and are very steep, simulating a road with many rocks and jumps. Figure 57 shows more clearly the profile.

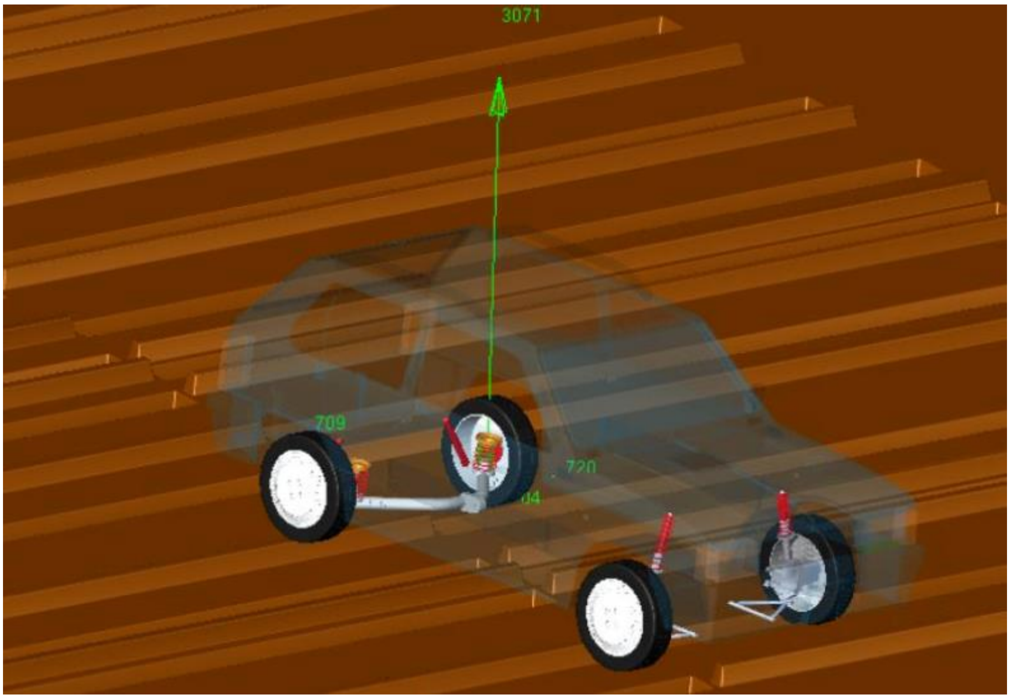


Figure 57: Profile 1. The alternated bumps and holes are visible

Profile 2: the second profile is more similar to a typical off-road route, with smooth holes (Figure 58). The road height variations are in the range of 4 to 8 cm and it is intended to be traveled at a much higher speed with respect to the profile 1. As expected, this path is less critical than the previous one.

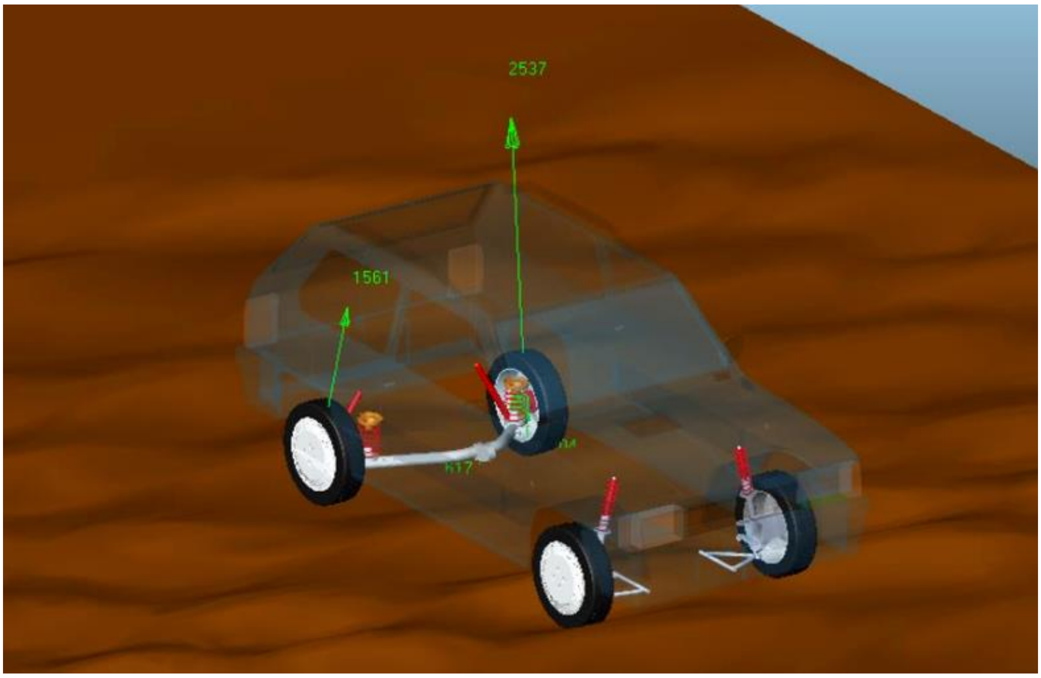


Figure 58: Profile 2. It is clearly much smoother than profile 1

3.2.2. Simulations

Once inserted all the data in the model and once imported the input profile, the following step is to give the motion to the vehicle.

In the previous paragraph the vehicle model was introduced and the important massive parts of the car have been added, to represent the correct weight distribution. In this way it was possible to find the position of the center of mass. At this point, through the software Adams, the tool "general motion" found in the section "motions" was used to assign a motion to the vehicle center of mass.

The main purpose is to obtain the forces acting on the rear suspension system, in particular on the flange element. Since the steering motion was not introduced, and there is no input from the driver, the choice was to assign a uniform straight-line motion along x direction. The tool used in this case is the IF function which specify the starting velocity and acceleration as well as the speed to maintain during the test.

Therefore, all the other 5 degrees of freedom were kept free, to let the vehicle move as in reality. The motions of the vehicle will be following ones: rolling about the x axis (Figure 59), dive and squat motion about the y axis (Figure 60) and yaw motion about z axis.

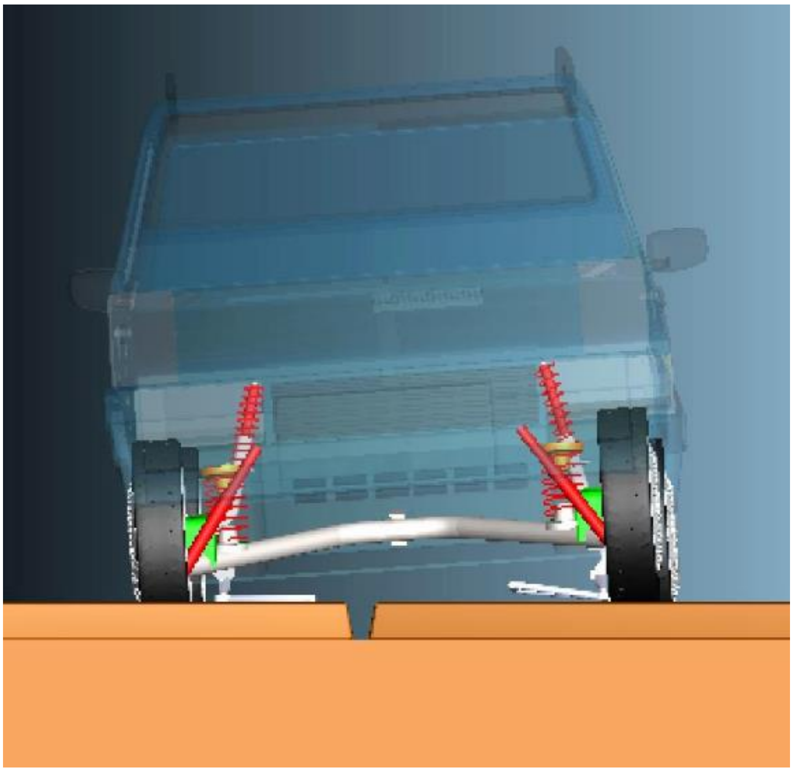


Figure 59: Vehicle rolling about x axis

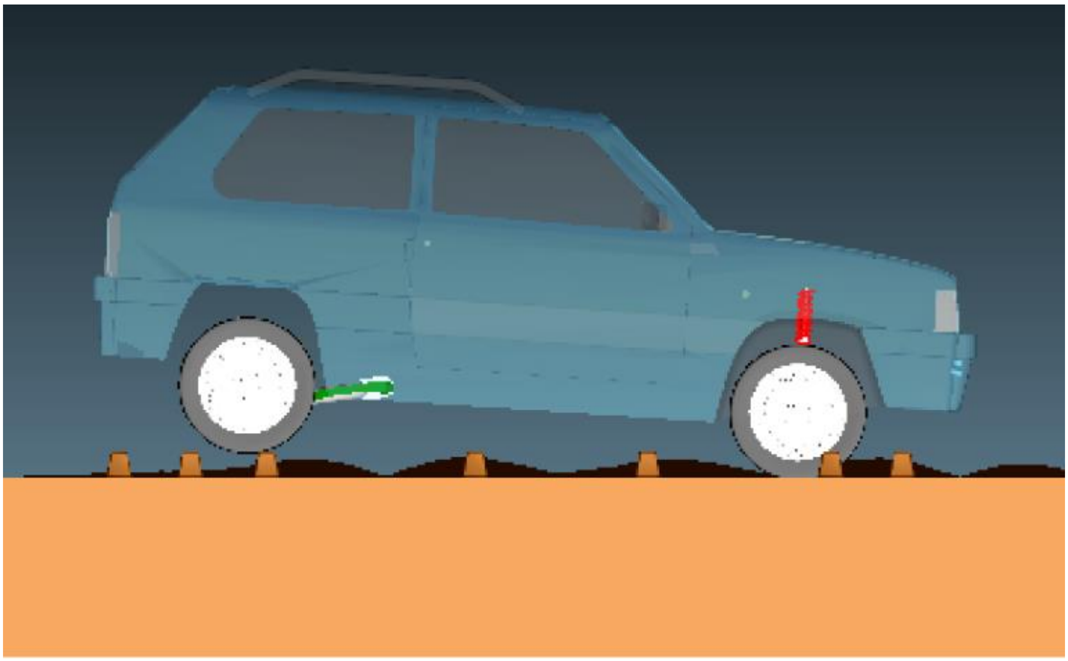


Figure 60: Vehicle dive motion while traveling on profile 1

Different simulations at different speeds have been carried out with the purpose to evaluate the forces between the wheel hub and the connecting flange, and to analyze their correlation with speed.

The expectations are that the forces grow with the increase of the speed and that they are higher in the profile 1 than in profile 2, as explained previously.

Being the profiles very rough, it has no physical meaning to impose high velocities to the vehicle model because the car is forced to go slow to overcome the obstacles.

3.2.3. Obtained results

The forces acting on the constraints are vectors indicating the loads on the suspension. The vectors can be decomposed in triples of vectors directed along the x, y, and z axis. The results obtained from the simulation for each assigned velocity and for each profile are illustrated in Figure 61 and Figure 62. The graphs refer to the following velocities:

- slow: 5 km/h (green color);
- medium: 10 km/h (light-blue color);
- fast: 15 km/ (red color).

PROFILE 1

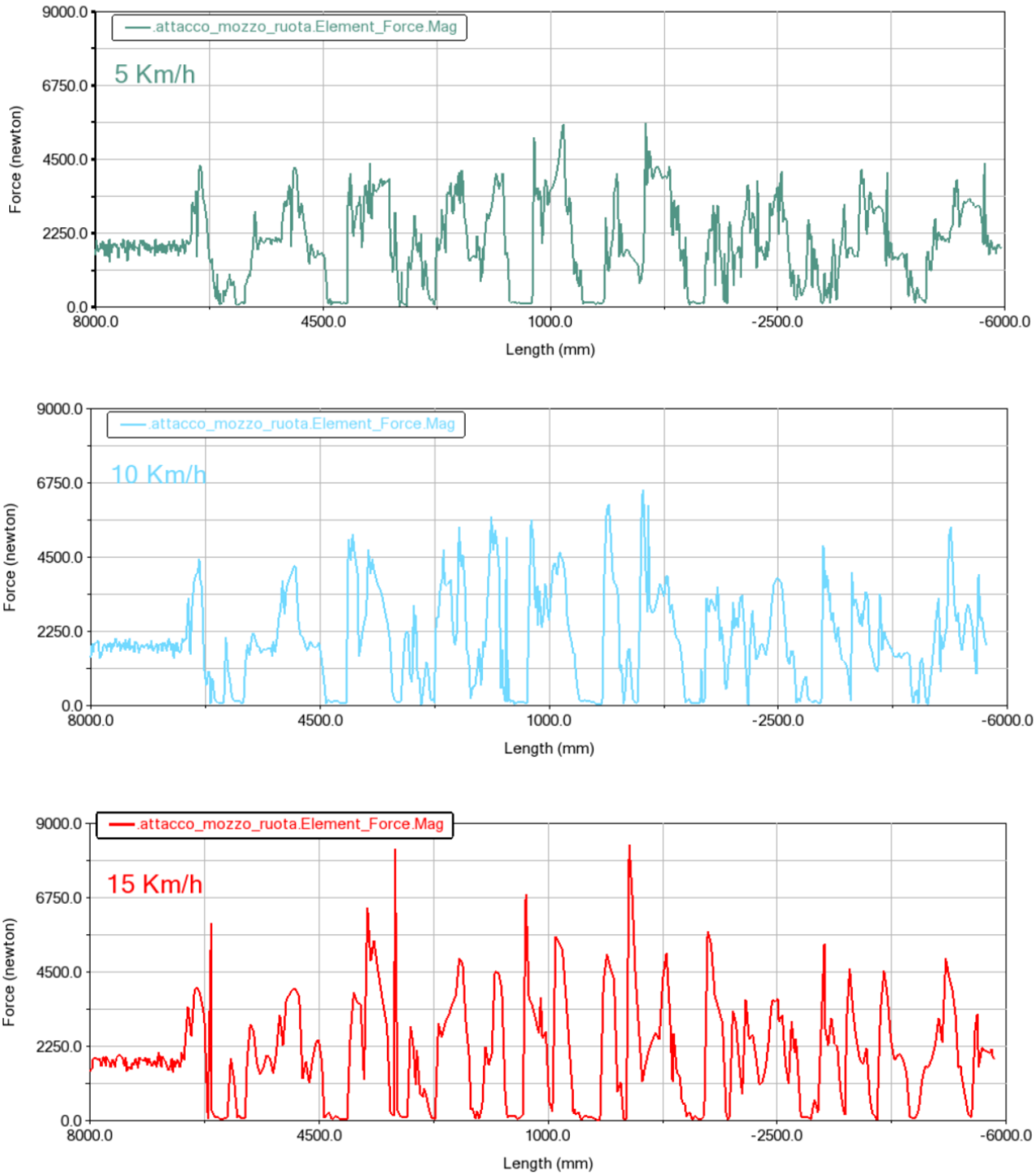


Figure 61: Different velocities for profile 1

PROFILE 2

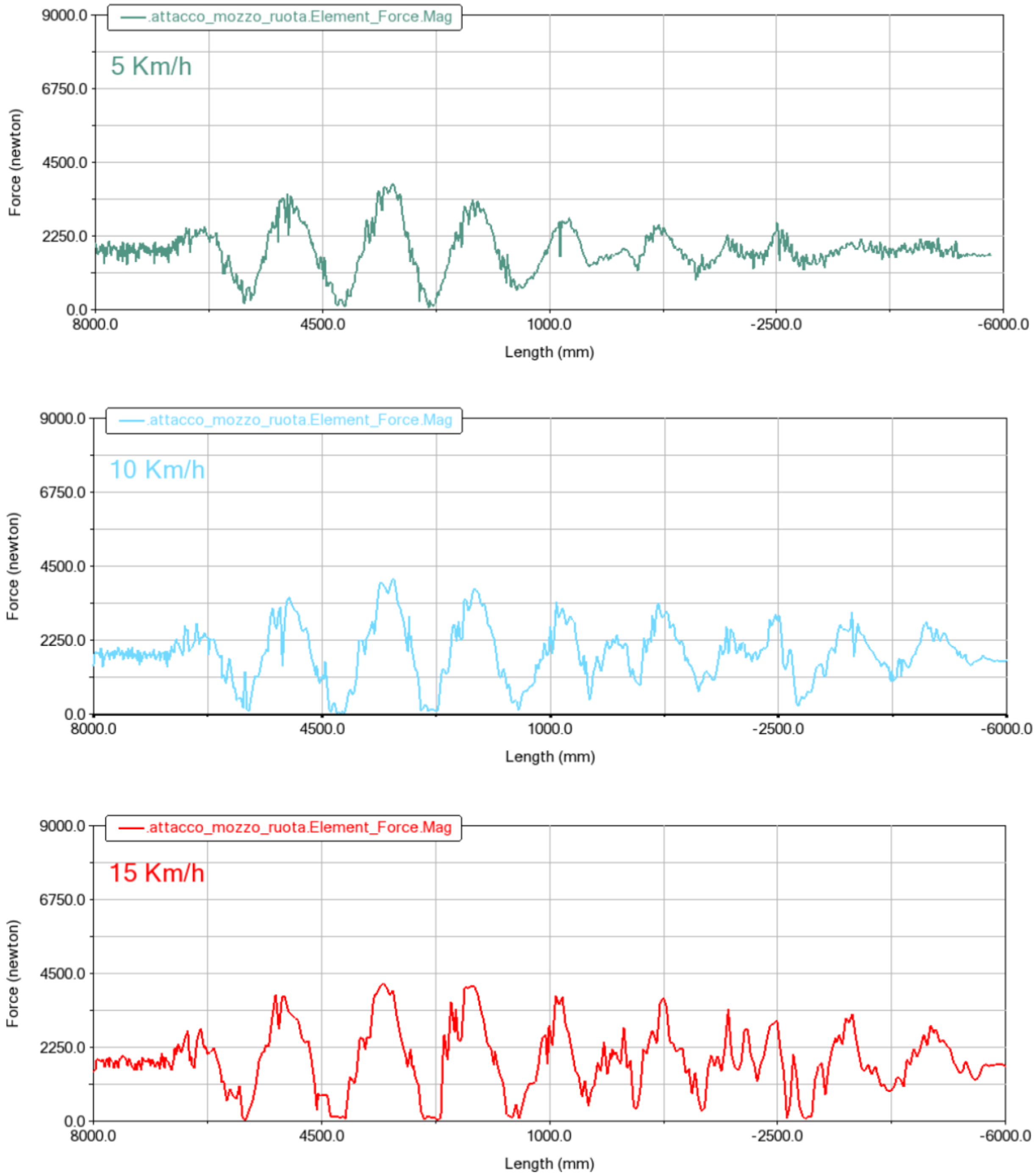


Figure 62: Different velocities for profile 2

Analysis of the results – First of all it is possible to notice that the all the graphs referred to the same profile have a similar trend. In all of them the forces oscillate around a mean value while, at some instants, they show very high peaks. This is caused by the fact that the model is not perfect, in fact it is an approximation of the real vehicle. All the constraints introduced between the components approximate the real behavior. Moreover, also the input profile was created making some assumptions and it is possible that during some contacts the model reacts in a "strangely". By the way in the overall the model is good, and the results are the ones expected. The high peaks have been considered as errors of the model, so they have not been taken into account.

These results are useful to have an idea of the value of the forces acting between the constraints, so to evaluate afterwards if the components can withstand this kind of loads.

It is important to consider also that the vehicle speed is maintained constant for all the duration of the test, independently of its position. In reality the speed can vary very rapidly in this kind of competitions, in fact the driver can slow down if there are critical obstacles. This makes the model more conservative with respect to the reality and guarantee a certain safety margin on the results.

Analyzing the trends once again and comparing the two profiles it is possible to notice that profile 2 is characterized by graphs having the forces oscillating at a lower frequency with respect to profile 1. It means that the forces vary more gently without sudden peaks as in profile 1. This is due to the fact that when the wheels touch the obstacles in profile 1, some shocks are present.

Comparing now the numerical results it is possible to see that increasing the speed of the vehicle, also the value of the forces increases, as expected. For example, in profile 1, there is a peak of 5500 N at 5 km/h while at 15 km/h the peak is 8000 N. In profile 2 this fact is less evident, but the trend is the same: 3500 N at 5 km/h and 4500 N at 15 km/h.

As mentioned above, comparing the numerical results between the two profiles considering the same velocity, it is noticeable that profile 1 generates much higher loads, having a peak force of 8000 N while profile 2 stops at 4500 N.

Finally, the load resulting from the most critical situation (profile 1, 15 km/h) on the connecting flange, is shown decomposed in its three components x, y, z (Figure 63).

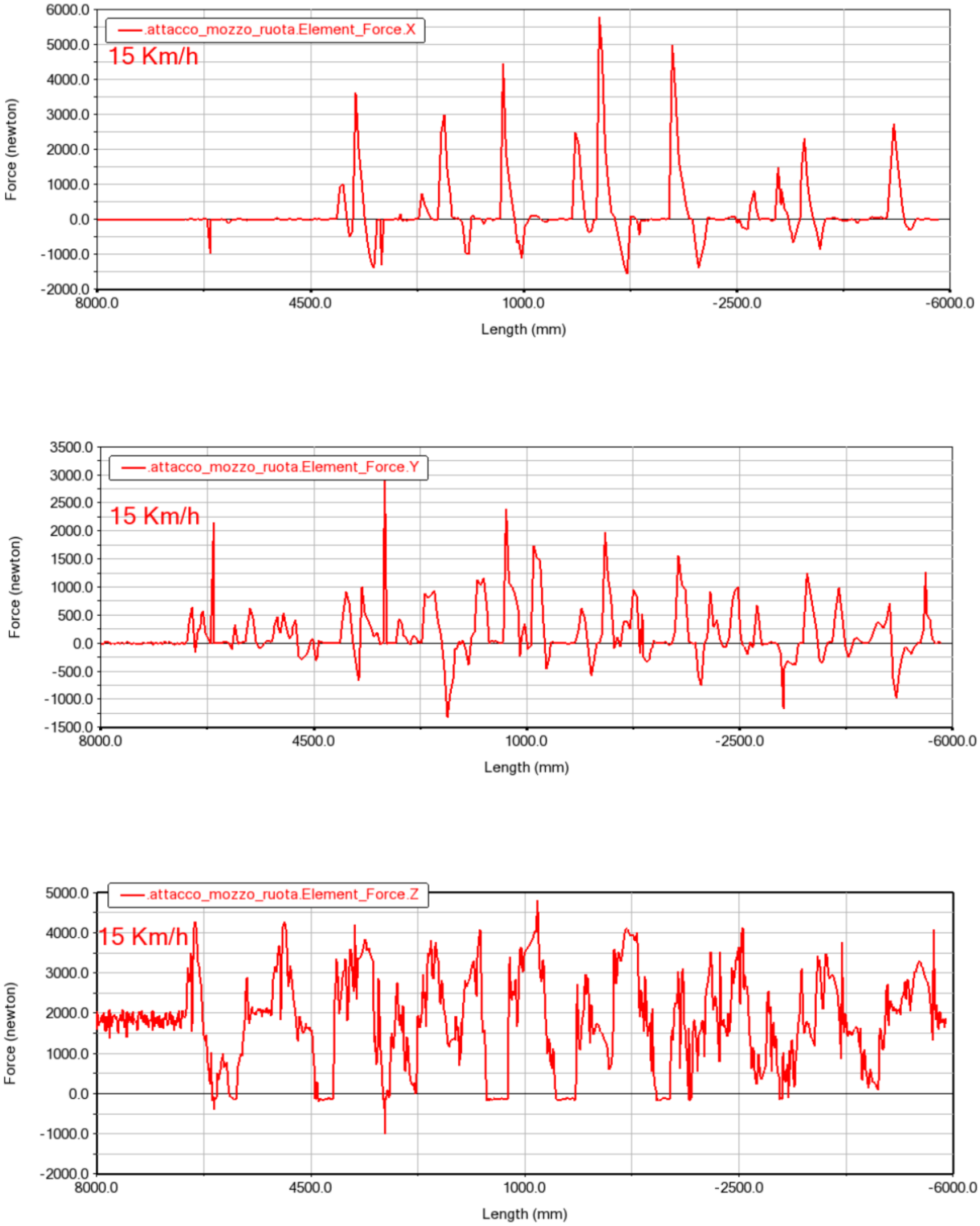


Figure 63: Highest force decomposed in its three components: x (top), y (middle), z (bottom)

3.3. Topological optimization and FEM structural analysis

The finite element analyses are commonly used in the mechanical design field because they allow the study of any kind of structure and of mechanical part. They can be used also when the geometry is very complex and when the boundary conditions are too difficult to be studied from an analytical point of view. Moreover, these software can interface with CAD and CAM software through which it is possible to easily import the geometries and analyze them.

The software used in this project is SolidThinking Inspire [7], a powerful tool that allows to perform analyses and structural optimization in a simple and intuitive way. In its user friendly interface there are a lot of tools: a CAD modeler through which it is possible to generate some geometries to analyze, a pre-processor in which the structural model is generated and discretized and where the simulation inputs are defined, a solver which can solve numerically the analyses, and finally a post-processor through which the results can be evaluated.

In this chapter it will be reported the structural analysis of the flange connecting the tube to the wheel hub. This component, as explained in the first chapters, besides being very stressed by the external forces, is also a part that was not present in the base Panda and that is designed from scratch. Therefore, it must be carefully analyzed and verified.

In the structural analyses it is important to define the discretization level of the model, in terms of dimension and number of the elements. Obviously, the higher the number of elements, the higher will be the time required by the software to compute the differential equations that govern the model. Therefore, also the duration of the simulations will be higher.

For this reason, in the case of the flange the imported geometry was reduced as much as possible: it was imported only the CAD of the flange, with the wheel hub and a piece of the suspension tube.

Once established the correct material for the piece and its size, and once defined the loads applied to it through the hub, a first box geometry of the flange was created. It was linked to the tube by means of a weld on the lower surfaces.

The purpose of this geometry is to look for the optimum dimension of the element of the mesh discretizing the model, to find the right trade-off between precision of the results and low computational time. This procedure is described in the paragraph relative to the convergence analysis of the results.

Afterwards, it has been used a function of the software that allows to realize a topological optimization of the flange model. By doing so is has been possible to obtain the final shape of the component, making it strong and lightweight. In fact, the weight has been minimized respecting a minimum safety coefficient in terms of Von Mises equivalent stress.

The following picture represents the starting shape of the flange before the topological optimization. The flange was imported with the wheel hub and the tube (Figure 64).

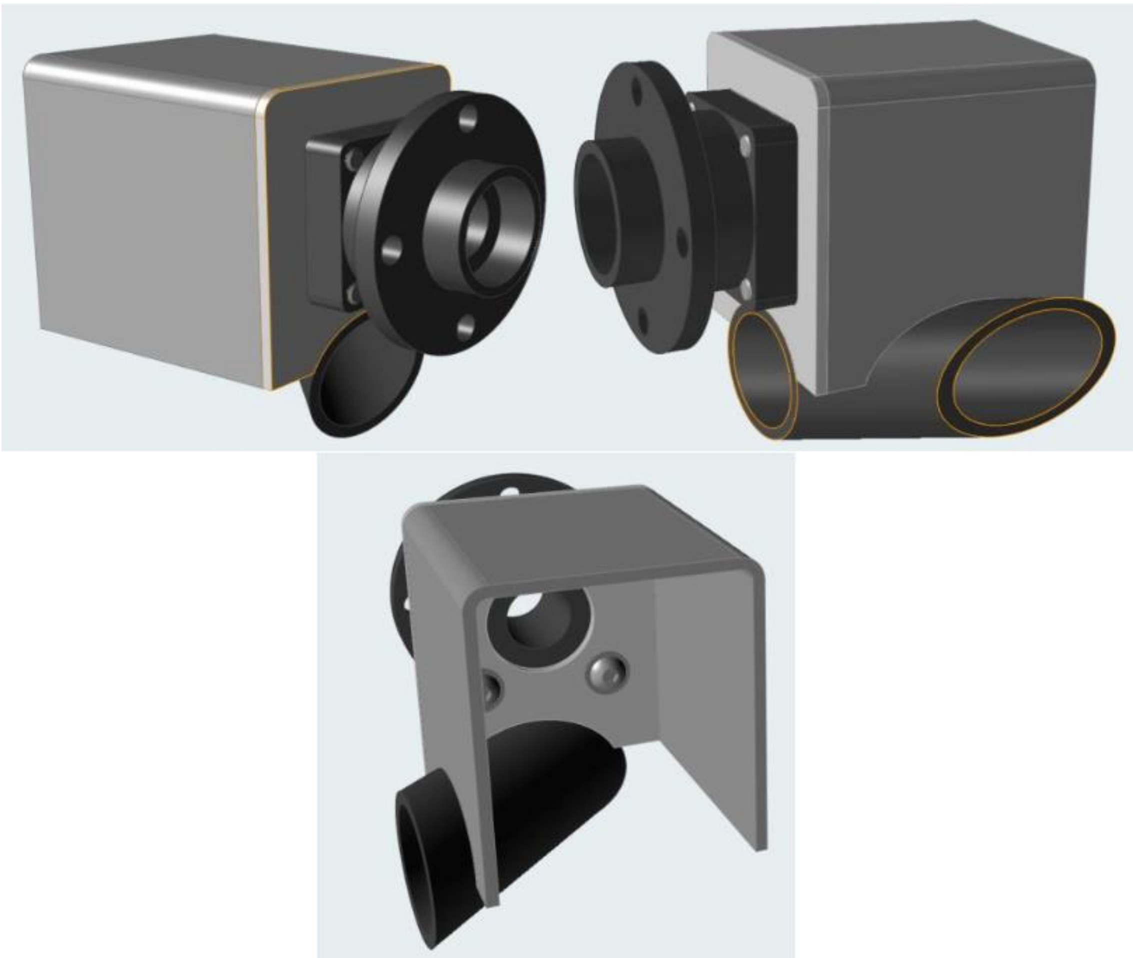


Figure 64: Box shape of the flange before the optimization

3.3.1. Analysis of convergence of the results

It has been performed a study about the convergence of the results of the structural analysis of the flange. The purpose is the determination of the mean element dimension of the optimum mesh. In particular, a reliable result of the analysis is desirable for the best trade-off between computational time and accuracy.

According to theories about FEM analysis, by increasing the number of elements present in the mesh, the model will be more and more realistic, getting closer and closer to the analytical solution of the problem. As a consequence, at every attempt the number of elements has been increased, making the mesh denser every time. Therefore, the degrees of freedom of the model, which depend on the number of elements, has increased as well. For this study it has been used a linear tetrahedral element of the first order.

The results, of which it has been analyzed the convergence varying the mesh, are relative to the Von Mises stress in a generic point P inside the considered model, near one of the four holes where the bolt linkages are placed. In fact, this is the most stressed zone. Moreover, this point P has been accurately chosen at a certain distance from the zones of the model in which the loads and constrains are applied. This choice is due to the fact that in such zones there are often critical points in which singularities and numerical instabilities happen with values without physical meaning [9].

It has been evaluated also the convergence of the value of the movement of the point F, that is the center of application of the load on the wheel hub.

The range of variation of the mean dimension of the elements is between 3 and 1 mm, with a step of 0.25 mm. In Figure 65 it is shown an example of mesh thickening.

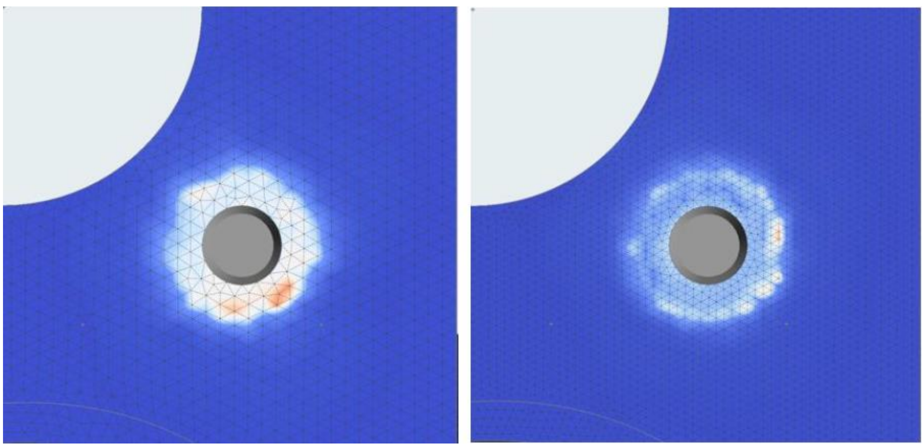


Figure 65: Mesh thickening. On the right the mesh is denser

In Table 17 are reported the values of the mean dimension of the generic element that have been considered in the convergence analysis. The following values are also reported:

- number of linear tetrahedral elements;
- number of nodes;
- Von Mises stresses in a generic point P;
- movement of point F;
- simulation time.

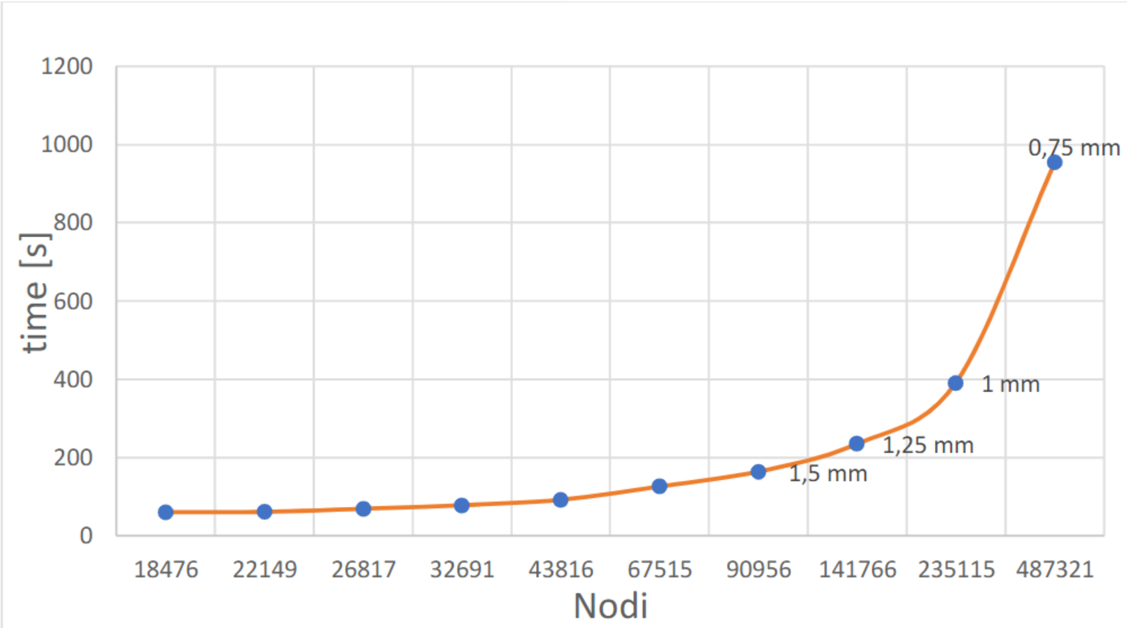
Mesh Dim. [mm]	# Tetra El.	# Grids	Von MIses P [Mpa]	Disp. F [mm]	time [s]
3	64995	18476	75,52	0,1428	60
2,75	79491	22149	65,77	0,1443	61
2,5	97362	26817	66	0,147	69
2,25	118920	32691	69,82	0,1479	78
2	163706	43816	67,69	0,1528	92
1,75	278697	67515	78,34	0,1664	126
1,5	377326	90956	83,12	0,1696	164
1,25	606895	141766	97,02	0,1755	235
1	1030439	235115	107,2	0,1824	391
0,75	2253272	487321	106,5	0,189	954

Table 17: Element characteristic values

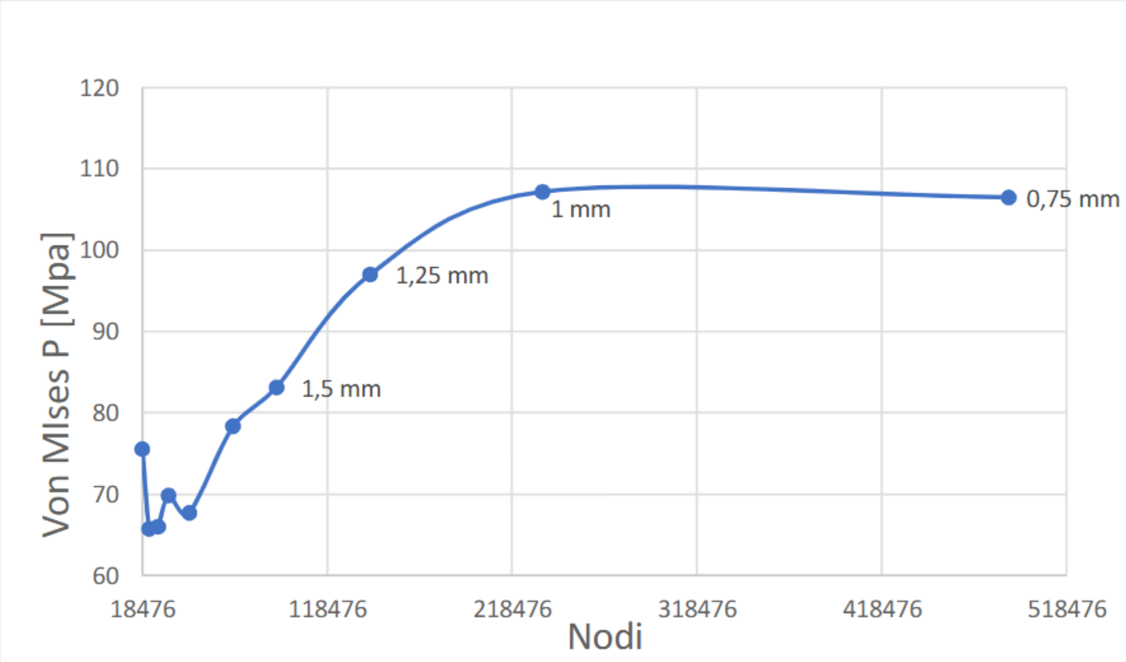
Once plotted these values it is possible to notice that the Von Mises results and the shift of the load application point tend to converge, going below 1 mm as mean dimension. By the way it is also noticeable that good results are obtainable also with a mesh with dimension of 1 mm, reducing drastically the computational time.

In Graph 1: Computational time as a function of mesh thicknessit is plotted the trend of the computational time as a function of the element dimension and therefore of the number of nodes of the domain. The time necessary for the analysis is inversely proportional to the mean dimension of the mesh, as previously observed. In Graph 2 and

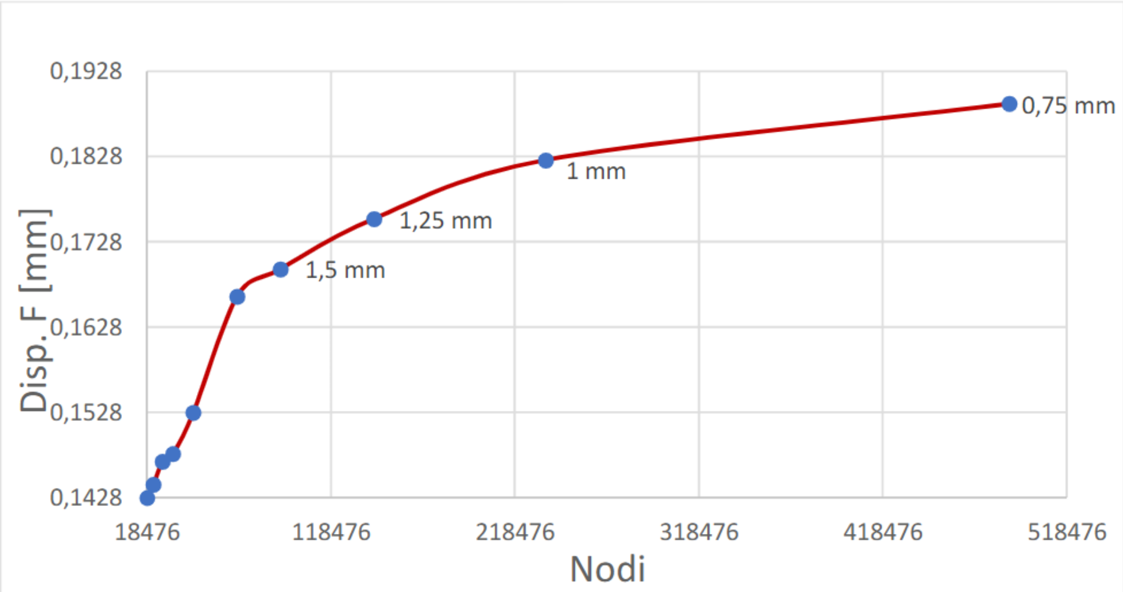
Graph 3 the trends of the Von Mises stresses and the displacement of point F with respect to the mesh thickness are plotted respectively.



Graph 1: Computational time as a function of mesh thickness



Graph 2: Von Mises stress as a function of the mesh thickness



Graph 3: Displacement of the point F as a function of the mesh thickness

3.3.2. Setting of the structural analysis and optimization on the box flange

Project specifications – A minimum safety coefficient of 2 has been defined for the maximum Von Mises tension. The material that will be used for the manufacturing of the piece is S420 construction steel. Its main characteristics are reported in Figure 66.

Yield strength (MPa)	≥ 420

Tensile strength (MPa)	480 – 620

Nominal thickness (mm)	< 3	≥ 3
Total elongation A80%	≥ 16	
Total elongation A5%		≥ 19

Temperature (°C)	-20
Notch impact energy (J)	≥ 40

ELEMENTO	AL	B	C	CEQ	CR	CU	MO	MN	N
Min.	0,015								
Max.			0,12					1,60	

ELEMENTO	NI	NB	P	S	SI	SN	TI	V	ZR
Min.									
Max.		0,09	0,025	0,015	0,50		0,15	0,20	

Figure 66: S420 steel characteristics

Loads definition – The aim is the evaluation of the behavior of the flange during the most critical operating conditions. To do so, the loads exchanged by the components through the linkages must be assigned.

For the component verification, all the loads found with the elasto-kinematic simulations will be used. These forces are the ones exchanged between the wheel and the hub. For this reason, the application point of the loads for the structural analyses will be on the wheel hub, at a certain distance from the flange (Figure 67).

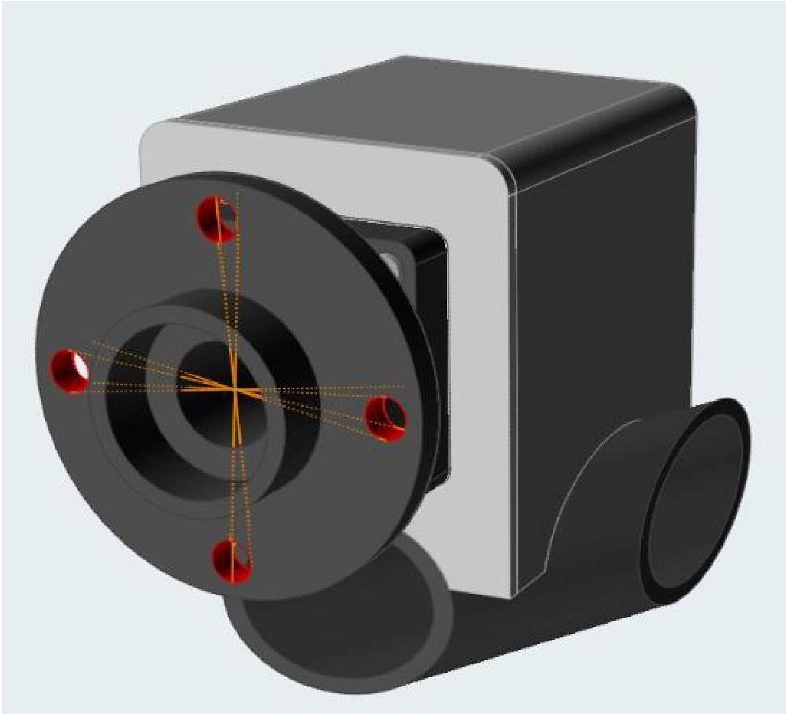


Figure 67: Loads application points (in red)

To perform a more conservative analysis it has been decided to take the peak forces in the three components x, y, z (Table 18). To consider the worst-case scenario, this vectors are applied in the same moment, though it doesn't happen in reality and the force is lower.

Component in **x** direction	**6000 N**
Component in **y** direction	**3000 N**
Component in **z** direction	**5000 N**

Table 18: Components of the force along the x,y,z directions

Contact zones and constraints definition – The first constraint assigned to the model is a fixed bond at the extremity of the tube (Figure 68). It represents the linkage with the central part of the suspension, that will be considered fixed to be on the safe side.

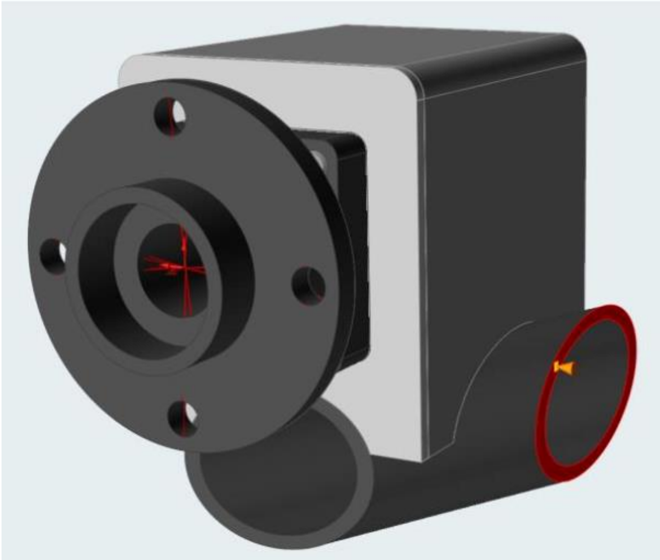


Figure 68: Constraint on the right extremity of the tube

Then it is necessary to define the contact zones between the components. This was a simple task performed using the function “contatti” of the software, through which it is possible to specify the type of contact. In this case they are a welded coupling between the tube and the flange, and a bolted coupling between the flange and the hub (Figure 69).

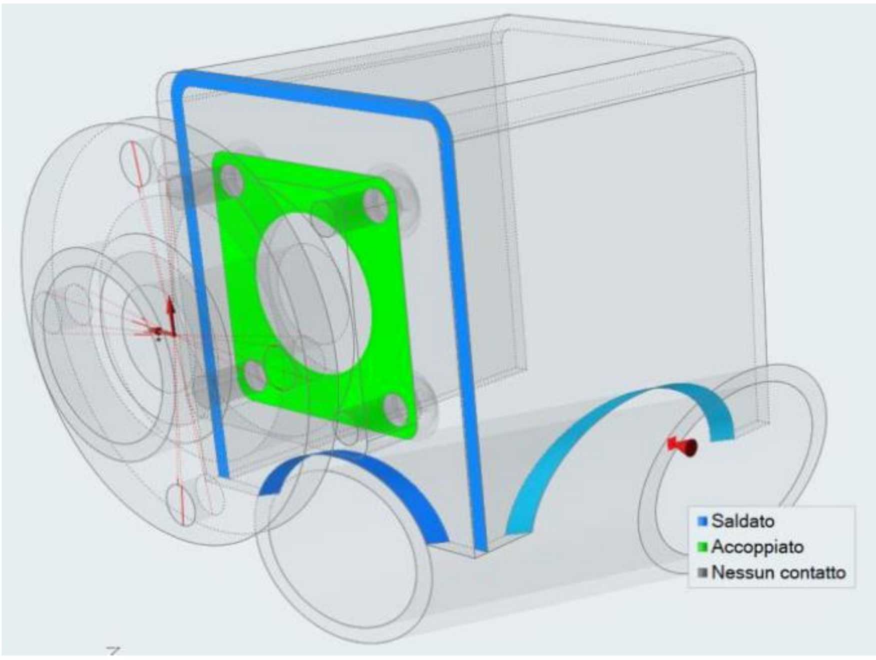


Figure 69: Contacts between the components

The last thing to define is the tightening torque to use in the bolted coupling between the flange and the hub (Figure 70). The workshop manual has been used to find the characteristics of the bolts and the correct pre-tensioning force (Table 19).

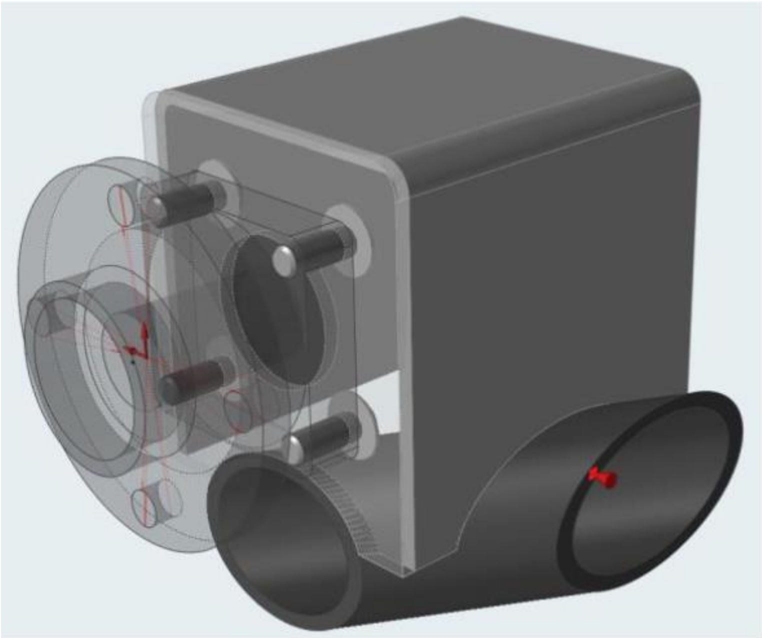


Figure 70: Bolted coupling between the flange and the hub

	Dimension	Tightening torque	Pre-tensioning force	Yield strength
Stud bolt ×4 Material class 12.9	M10 × 1,25	64 Nm	40000 N	60000 N

Table 19: Stud bolts characteristics

Structural analysis of the initial box flange – Below are reported the results obtained from the structural analyses of the initial box flange with the boundary conditions described above. These analyses are shown for a later comparison with the final geometry. Figure 71, Figure 72 and Figure 73 show the obtained results in terms of:

- Displacement;
- Compressive and tractive stresses distribution;
- Minimum safety coefficient and Von Mises tensions.

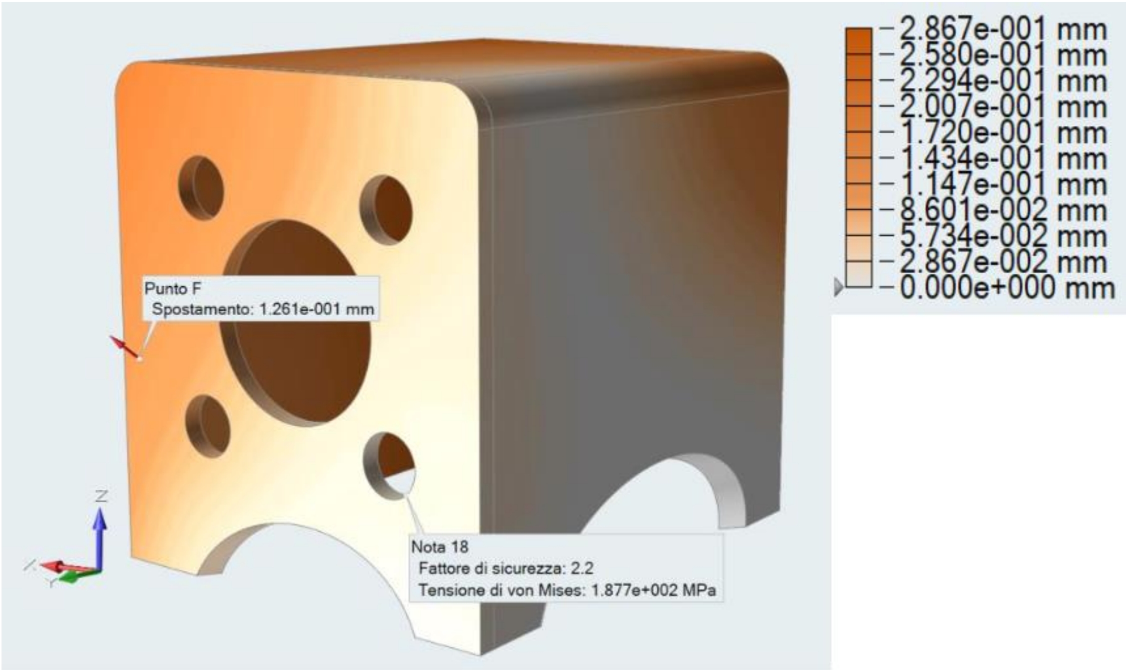


Figure 71: Displacement analysis

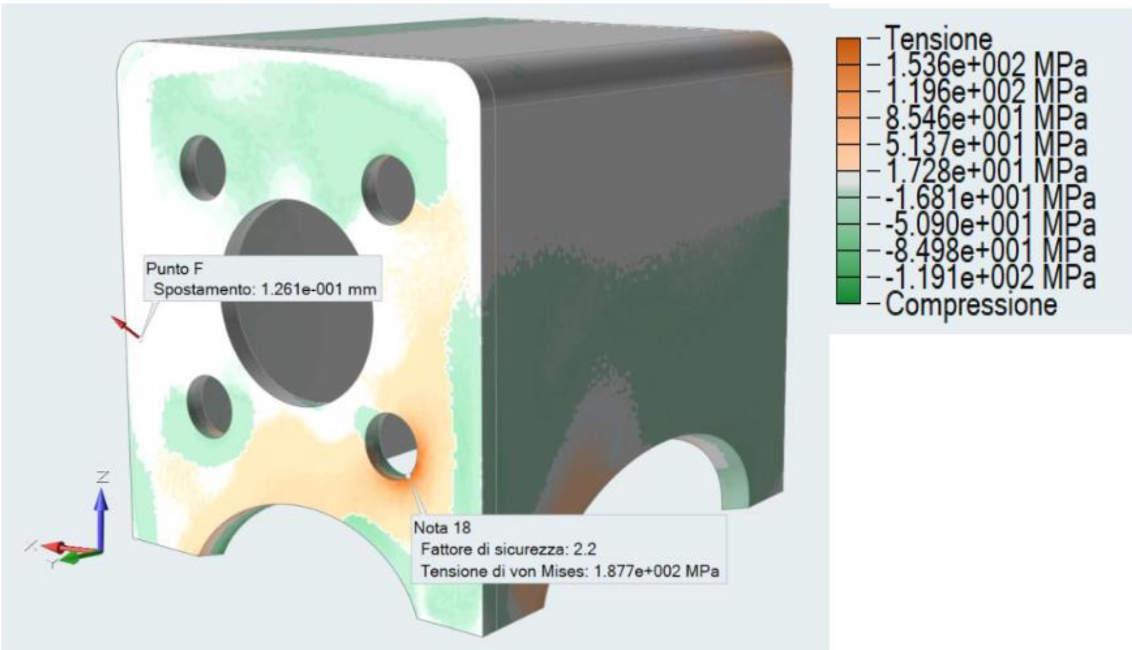


Figure 72: Stresses analysis

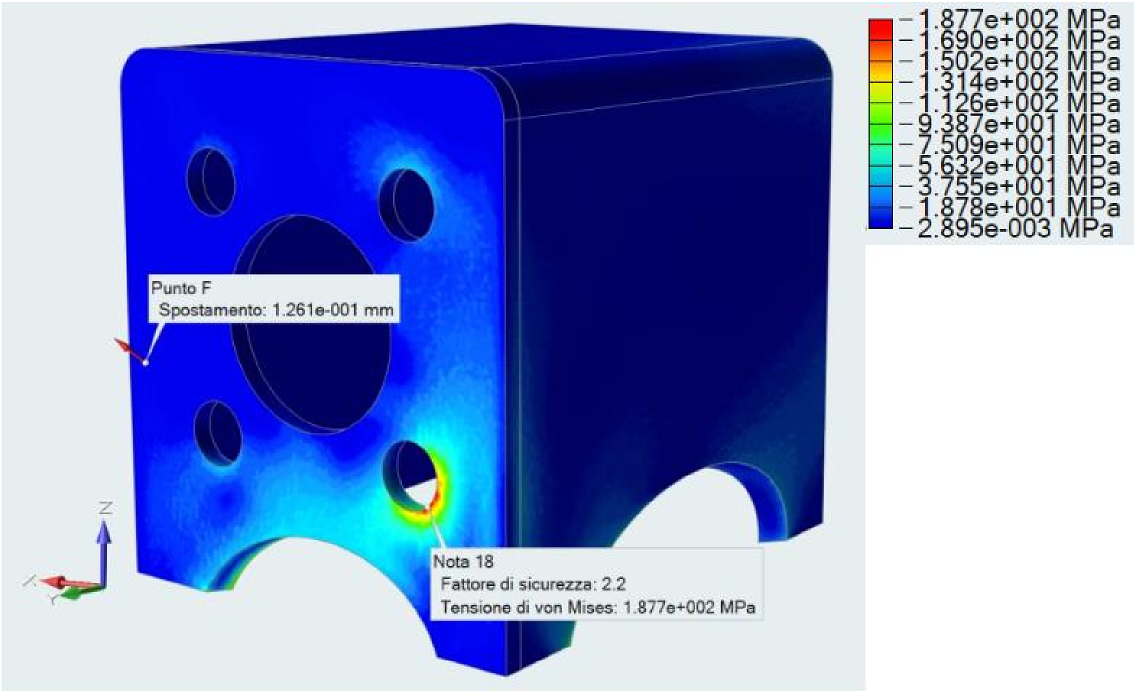


Figure 73: Von Mises tensions analysis

Settings of the topological optimization – The first aspect to evaluate regarding the topological optimization of the flange is the distinction between the non-Design Space and the Design Space (Figure 74). The former represents the zone of the model which should not be considered during the weight reduction process. Therefore, in this zone the mass is defined and cannot vary [10]. The non-Design space in the model is the face of the flange which is coupled to the hub. In fact, this zone is very important from the structural point of view and a high value of stiffness should be maintained.

Moreover, the non-Design Space should be defined also where the constraints and the loads are applied, since these zones often result in numerical instabilities of the model.

On the other hand, the Design Space represents the zones of the model in which the software will analyze the mass and through an iterative process. Then it will establish the optimal distribution of the material on the basis of the load conditions and on the constraints [10].

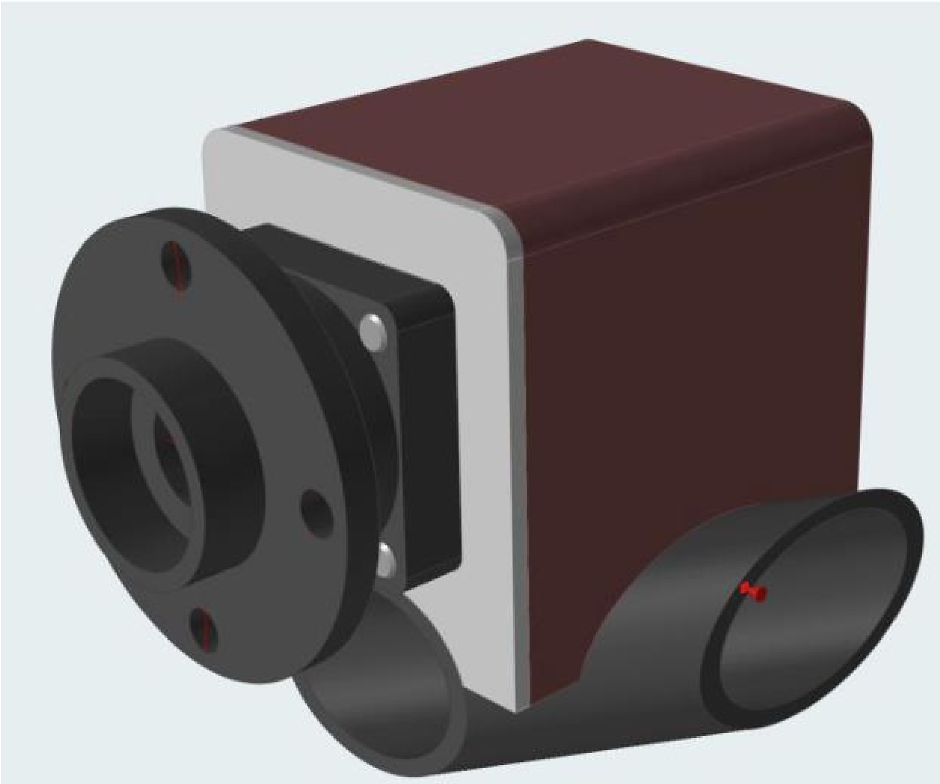


Figure 74: Design Space (amaranth color) and non-Design Space (light-grey color)

Through the software it is possible to perform two different types of optimization, having two different targets: maximizing the stiffness or minimizing the weight.

The first function consists of obtaining the best structure in terms of stiffness, minimizing the compliance of the model. This function imposes to establish a constraint on the discretization of the Design Space, so a minimum dimension of the mesh element must be defined. It will be taken of 1 mm on the basis of the convergence analysis previously performed. Moreover, it has been established to keep a volume equal to the 30% of the initial volume.

The objective of the second function is to minimize the mass of the structure, on the basis of the boundary conditions given, while ensuring a minimum safety coefficient that the represents the constraint of the optimization. In this case the minimum safety coefficient considered is equal to 2.

3.3.3. Obtained results

The procedure through which the model comes to the results consists on the assignment of the mesh to each element. The values of the assigned mesh density varie between 0 and 1, based on the boundary conditions, and following the iso-density curves [9], [11].

A value for the density threshold (cut-off) is set by the user, through the "design explorer" window. In this way the software draws a contour to which corresponds a set of meshes with a higher density with respect to the one established [10].

In this case a threshold value is established to guarantee a continuity in the generated geometry. The mesh density values that satisfy this minimum cut-off value are then automatically modified to the unitary number. By doing so a final topology with homogeneous properties is obtained and used for the generation of the definitive geometry.

The optimization results are shown below (Figure 75 and Figure 76), followed by the final geometry chosen for the best trade-off between stiffness and weight (Figure 77). The results of the two optimizations are very similar and evidence that a large part of the box flange is superfluous for the structural purpose.

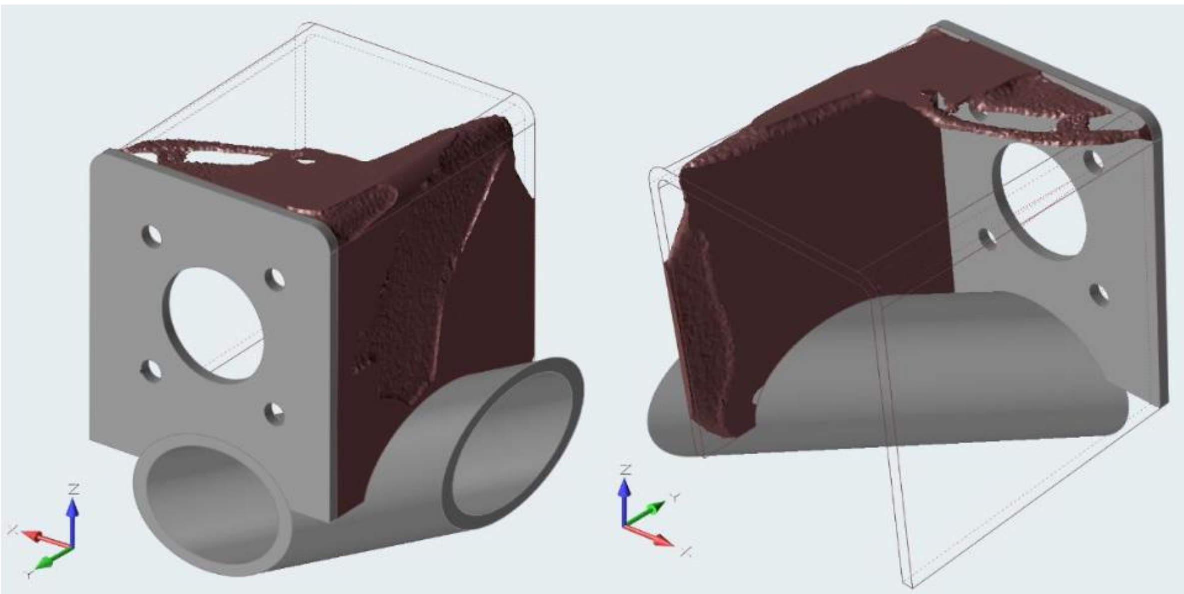


Figure 75: Topological optimization for stiffness

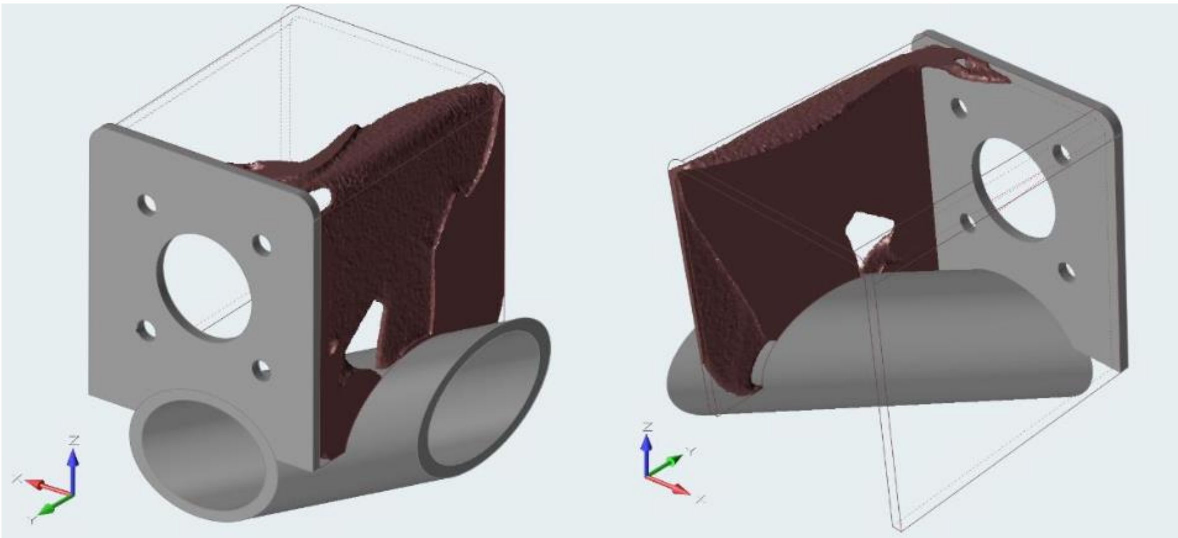


Figure 76: Topological optimization for weight

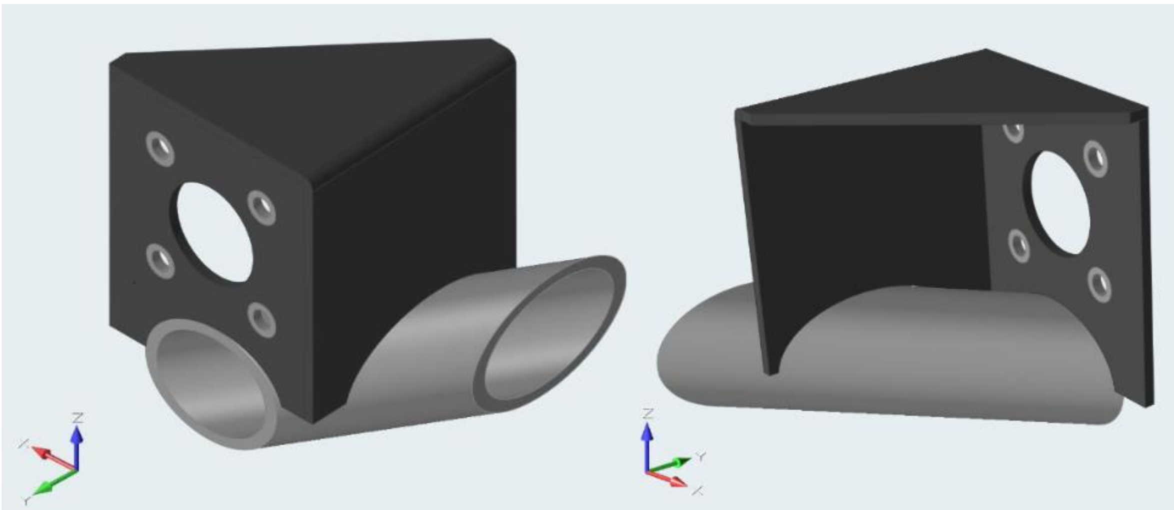


Figure 77: Final geometry of the flange

3.3.4. Verification analysis of the new geometry

Finally, here are reported the results obtained from the structural analysis of the final geometry, to verify that the minimum safety factor of 2 is respected.

In Figure 78, Figure 79 and Figure 80 the results of the analysis in terms of stresses distribution and of displacement are shown, with particular attention to the load application point on the wheel hub.

It is interesting to notice the improvement of the final flange geometry with respect to the box one (Table 20). The weight has been lowered by 43% with an insignificant reduction of the safety factor from 2,2 to 2,1.

	Safety coefficient	**Weight**
Box flange	2.2	2.27 kg
Final flange	2.1	1.29 kg

Table 20: Comparison between box geometry and final geometry

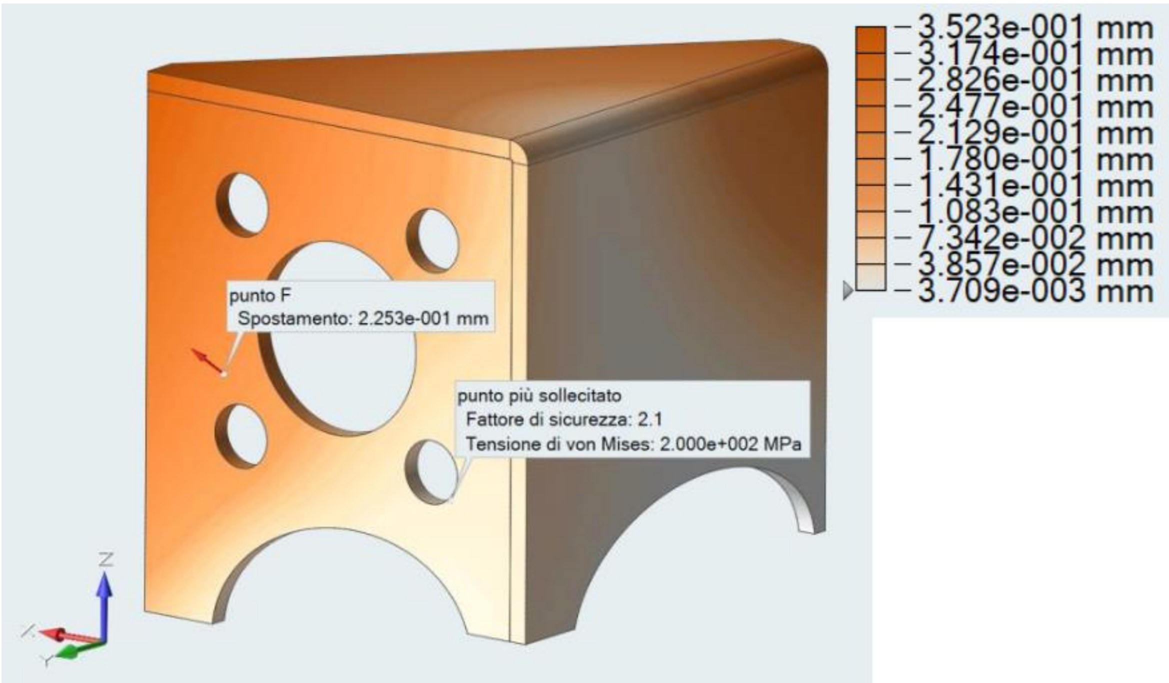


Figure 78: Analysis of the displacement on the final flange

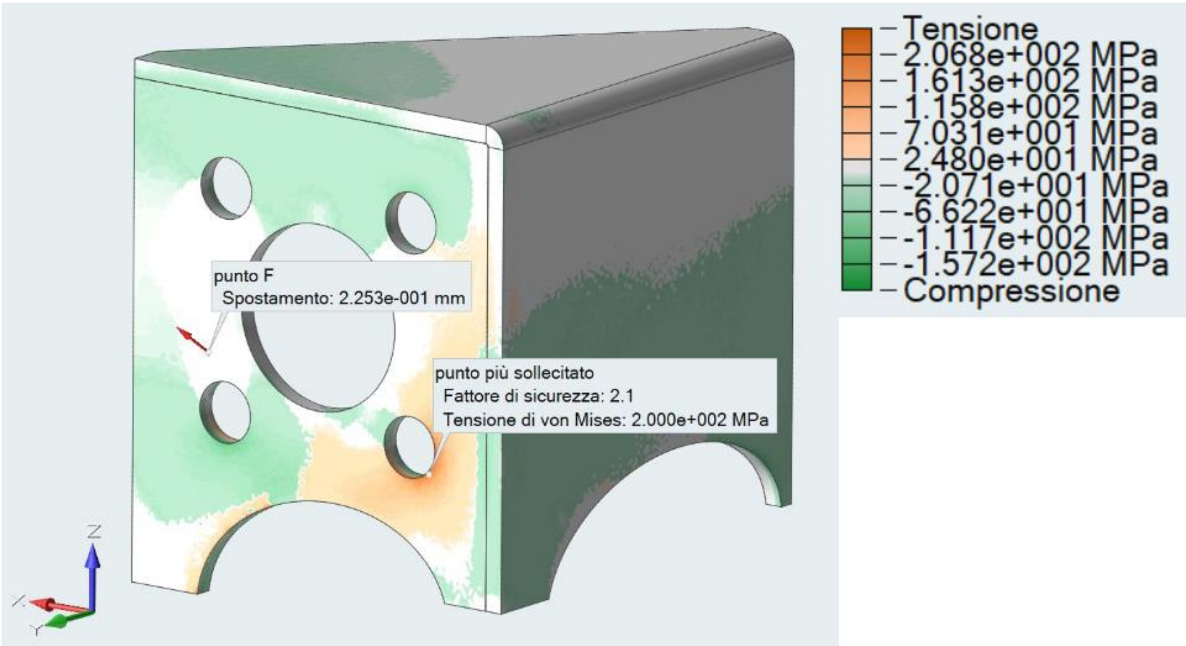


Figure 79: Analysis of the stresses on the final flange

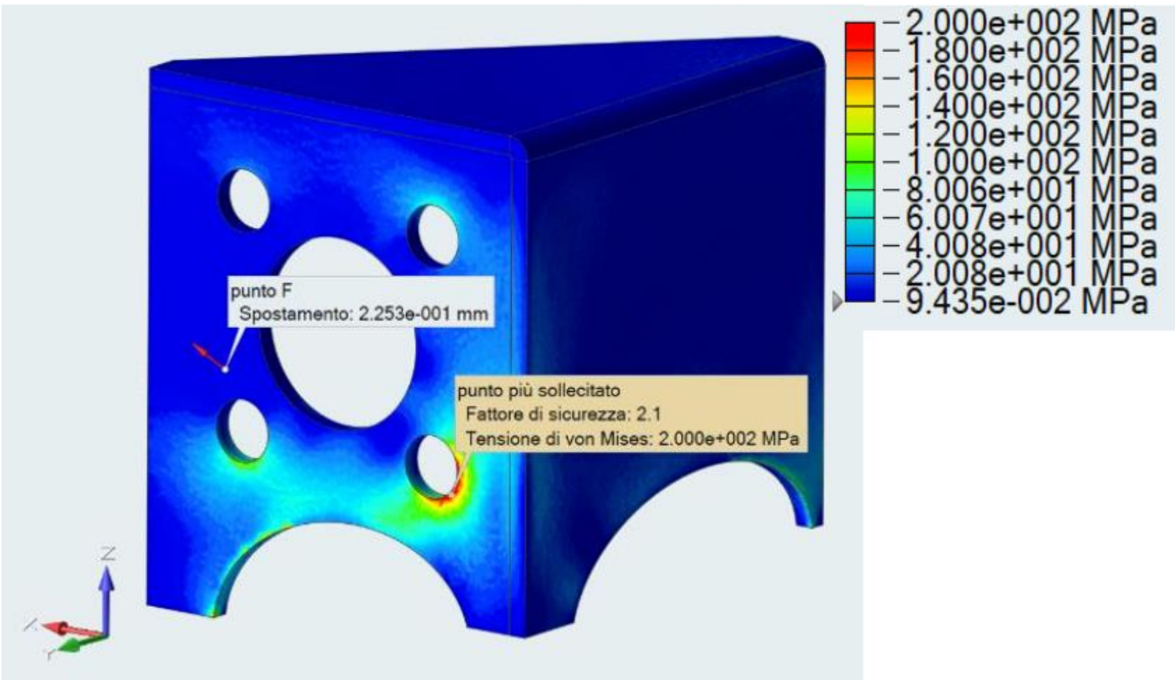


Figure 80: Analysis of Von Mises tensions on the final flange

3.4. From the engineering phase to the manufacturing process

In the previous chapters all the design phases have been carried out, starting from the choice of the best platform, and arriving to the structural verification of the most critical component, the flange.

To conclude this process, the last step is to create the files that can be used by the manufacturing company to machine the new components. Therefore, this is the phase connecting the "on paper project" to the "ready to go" vehicle.
Obviously, to create a real functioning model of the car, other problems have to be solved, for example defining a budget for the modifications to be done and finding a sponsor interested in the project. By the way, these issues are not discussed in this thesis.

To be easily manufacturable and have a low cost, the parts must follow some criteria, on the basis of the material used, the machining process, the tolerances and so on. For this reason, Italtecnica played a fundamental role. With their experience with the suppliers, they were able to give many important advices regarding the creation of the files.

3.4.1. Creation of the 2D drawings from the 3D models

The making of the 2D model starts from the 3D CAD model of the part. The software used for the process was Solidworks, which is very intuitive and has a very good interface with other software.

Since the flange will be made out of metal sheet, in the 3D model it was used the library "lamiera" for the creation of the part. It includes many functions and processes for the correct design of the metal parts, and it can vary its parameters basing on the material characteristics. The most important parameters to be set are the following:

- **metal sheet thickness**, in this case 5mm;
- **bending radius**, defined on the base of the sheet thickness;
- **K factor**, which is a bending property of the metal sheet.

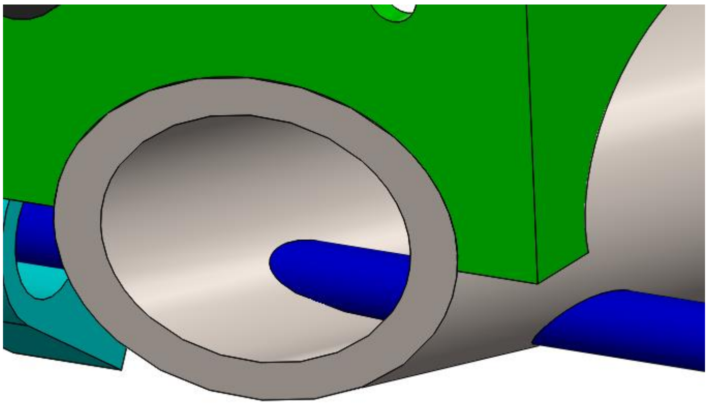
The thickness was chosen on the basis of the structural analysis, while the bending radius and the K factor were suggested by the company.

Critical points

During the design of the two pieces a critical issue was encountered, due to the complexity of the geometry. Since the tube is inclined with respect to the faces of the flange, the contact surface between the tube and the flange has to be adjusted leaving some clearance, according to the manufacturing process.

On the contrary, in the first phase of the modelling, the machining process was not considered so the coupling between the pieces was ideal, but it would be very expensive to realize a piece with such complex geometry. A new design had to be drawn.

To create the clearance between the tube and the flange, a shape following the tube profile was sketched. It was used the projection of the tube on the flange faces, an ellipse. The projection was enlarged to have no interference between the tube and the flange, considering that the flange is 5 mm thick (Figure 81).
Then, the clearance volume will be filled during the welding phase.



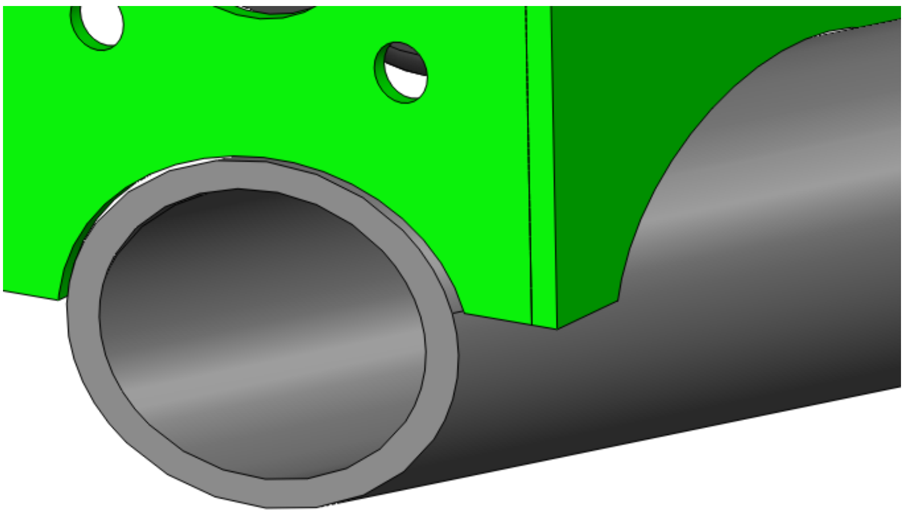


Figure 81: Contact in ideal model (top) and final design for machining requirements (bottom)

After the 3D model was complete, the 2D drawing had to be created. For doing so the flange was designed in two pieces and two different drawings were created (Figure 82). Then they will be welded together to form the final shape already presented in Figure 44.

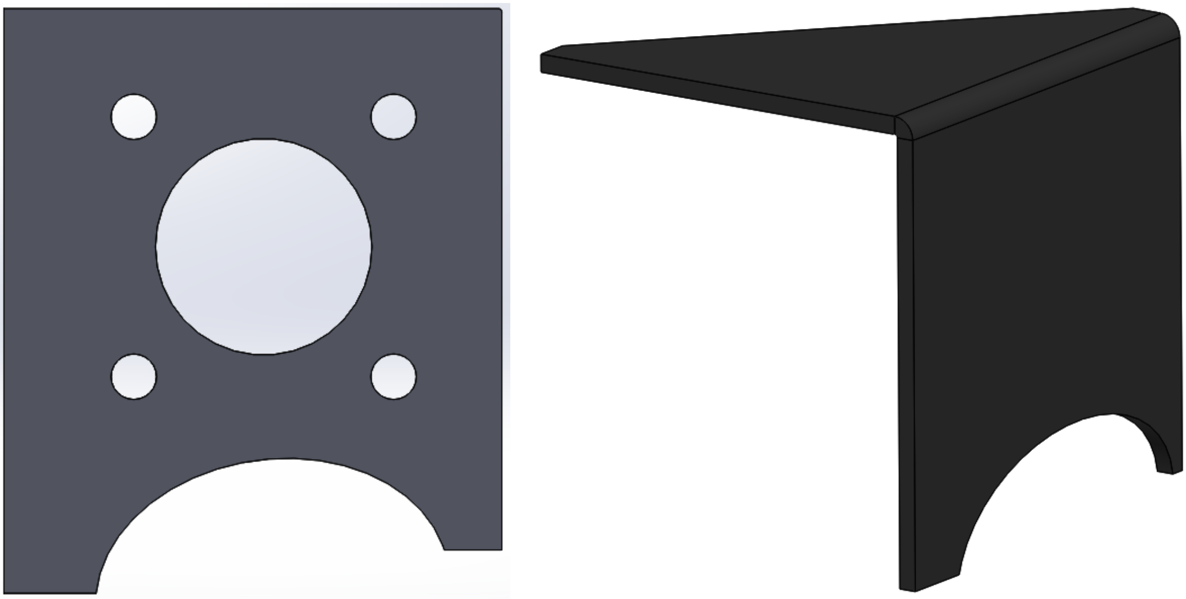


Figure 82: Metal sheet flange divided in two parts

For the creation of the drawings it is possible to go directly from the 3D part to the 2D sketches. In fact, using a dedicated function, the software opens a specific environment in which it is possible to put all the views of the piece with all the dimensions. Then the files can be saved using different formats, according to the supplier request.

3.4.2. PDF files

Since the shaping process for the metal sheet flange is the laser cut, two different files must be created for each component. One is a PDF file containing all the information regarding the part: the dimensions, the useful views and the cartouche of the drawing. The second one is a DXF file which is used directly in the machining process. It contains nothing but the shape of the sheet to be cut.

Since for one piece of the flange the metal sheet is firstly cut and then folded to the final shape, it is necessary that the drawing represents both the profile to be cut (unfolded raw metal sheet) and the final geometry, to make the sketch more clear and to avoid mistakes. By the way, using the function “lamiera” in the 3D model, the software automatically creates an unfolded view of the piece, making the drawing phase very easy (Figure 83). For the other piece (left picture of Figure 82) the problem does not exist because there are no bended surfaces (Figure 84).

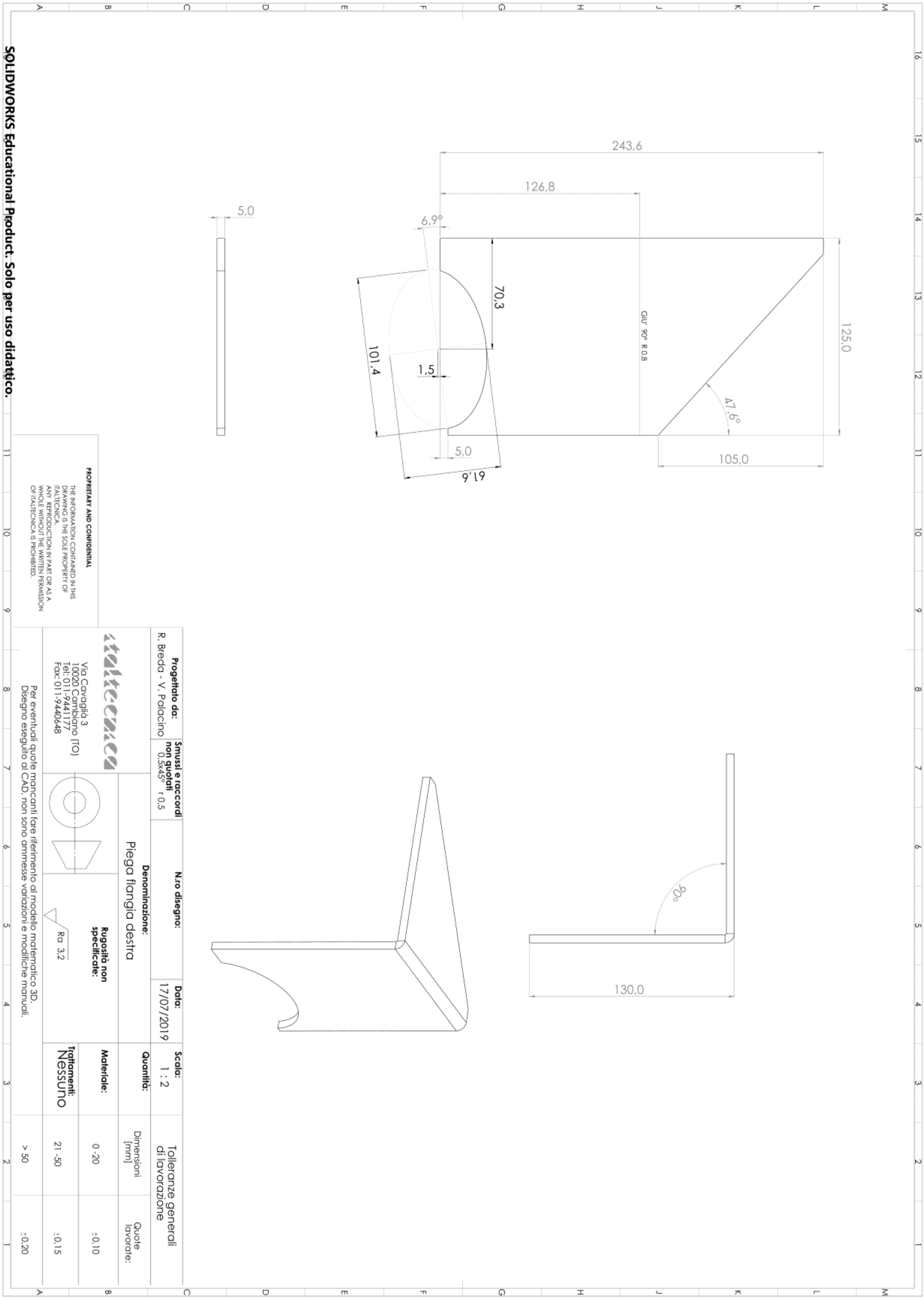


Figure 83: PDF file of the folded piece of the flange

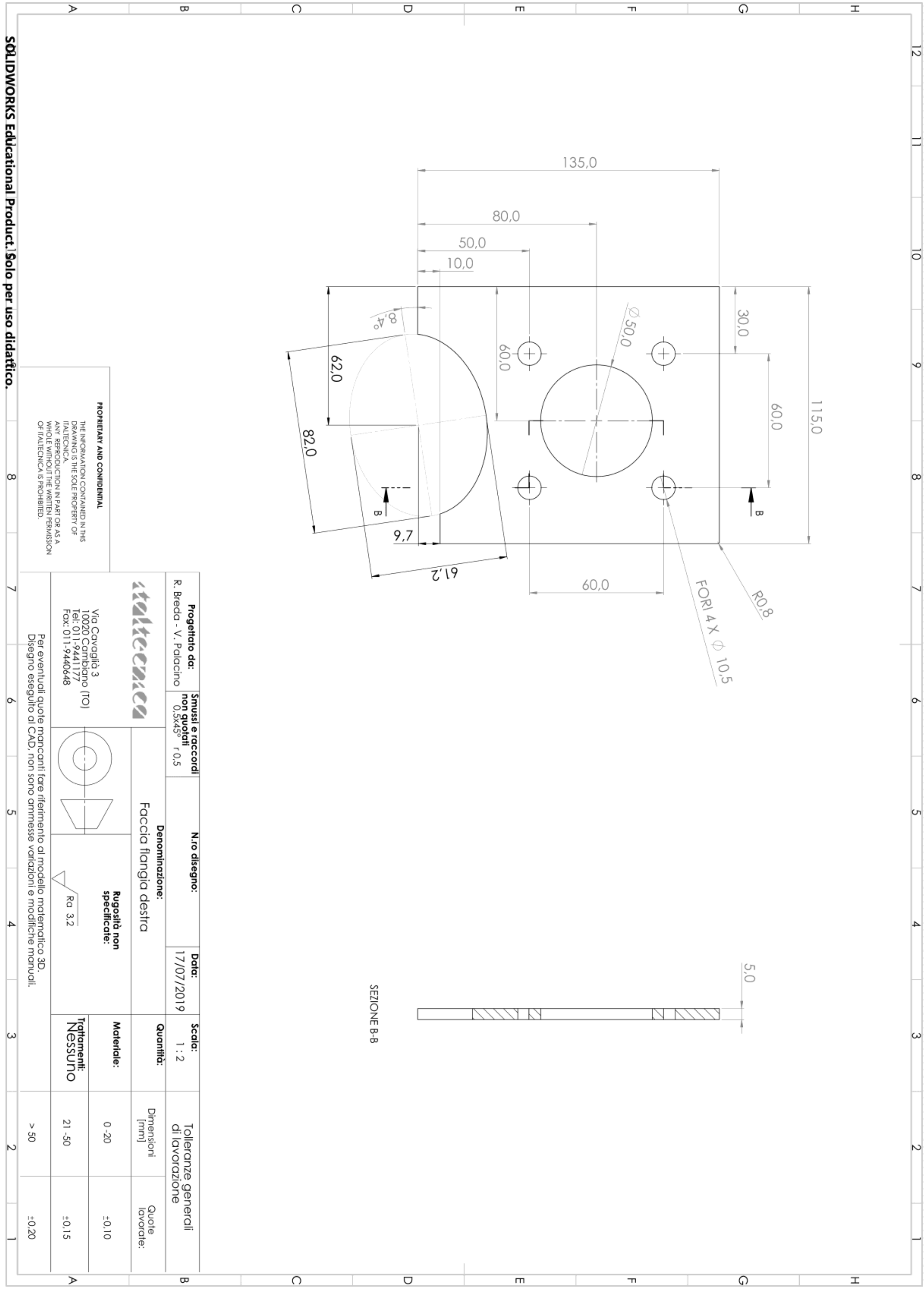
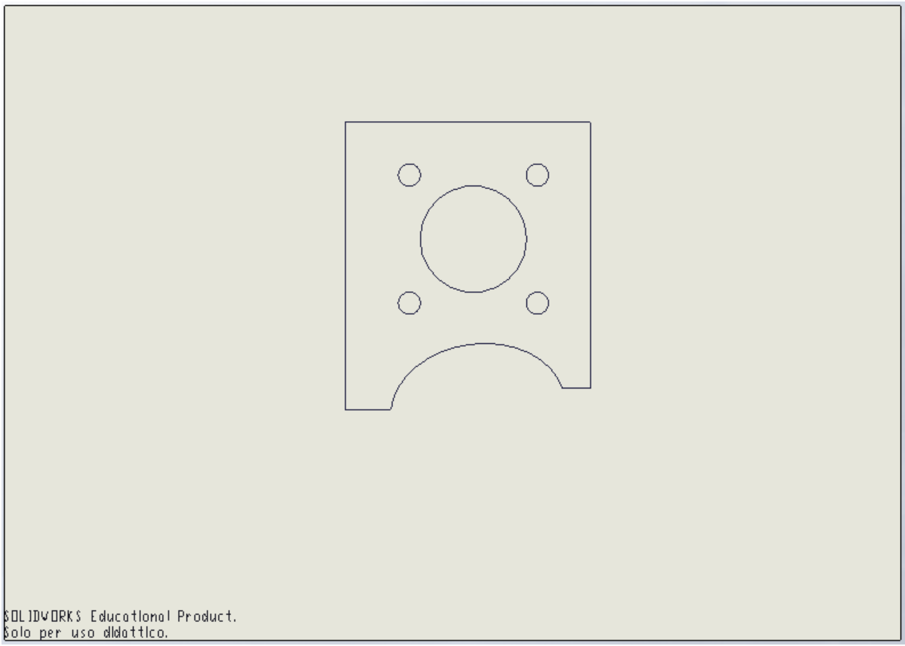


Figure 84: PDF file of the main face of the flange

3.4.3. DXF files

DXF (Drawing Interchange Format) is a file format used as an interface to communicate with other programs. It is requested by the manufacturing process and it can be easily created using Solidworks.

The file must contain just the profile that the machine has to follow to cut the piece and nothing more. The other elements, such as the views, the dimensions and the cartouche are removed (Figure 85).



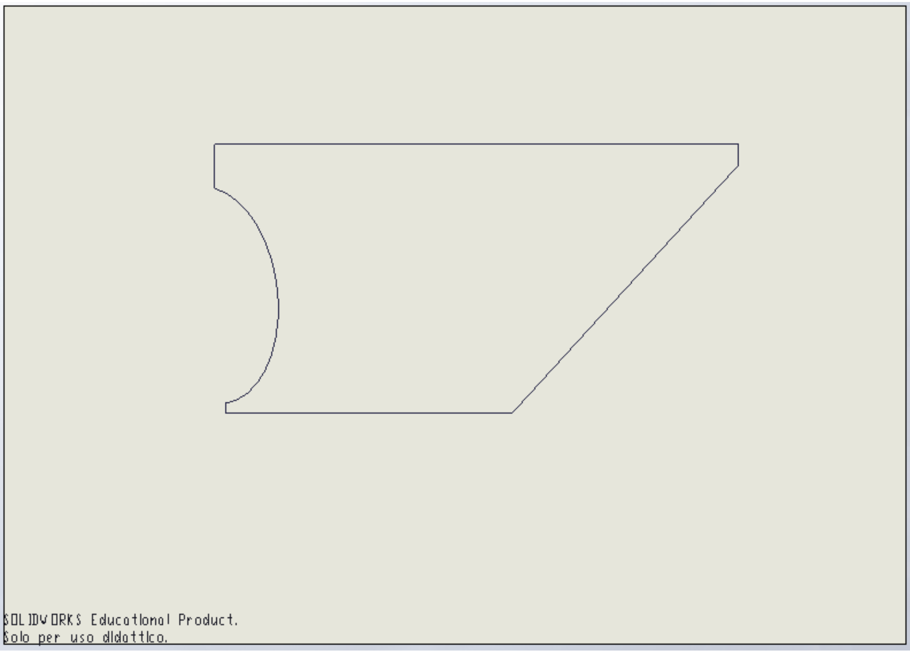


Figure 85: DXF files of the two pieces of the flange

4. CONCLUSIONS AND FUTURE DEVELOPMENTS

The work performed in this thesis is a part of a bigger project which objective is to build a functioning hybrid vehicle starting from the Panda 1° generation platform.

The entire rear suspension system was discussed under the functional point of view, as well as from the feasibility and cost of the redesign, with all the results reported. Nonetheless, many criticalities emerged during the design phases, and a constant communication with the company has revealed to be essential for a successful work.

However, there are still many things to do to be able to participate to the Panda Raid in March 20.

In particular, from the mechanical point of view:

- the first thing to do is to conclude the work on the vehicle trim;
- then it is essential to design a supporting frame for the transaxle unit;
- next a study on the motor connected to the engine should be done;
- as last thing a cooling system for the electrical stuff should be designed.

From an electronic and control point of view:

- the team is working on a model to upload on the vehicle management unit, to manage the torque split in the complex powertrain;
- an efficiency map of the electric motor is under development, to create the best strategy in terms of energy management;
- battery pack dimensioning to be installed, on the basis of specific strategies.

Finally, some important phases involving all the project fields will be:

- costs evaluation for the production of the new components;
- vehicle realization and validation tests on the field.

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