



Prifysgol Cymru
Y Drindod Dewi Sant
University of Wales
Trinity Saint David

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Emanuele Di Crosta

P151088

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Declaration of Authenticity

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree

Signed: 

date: 04/05/2020

*Dedicated to my beloved Brother,
Francesco*

“Everyone is a genius.

*But if you judge a fish by its ability to climb a tree,
it will live its whole life believing that it is stupid.”*

Albert Einstein

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1 Abstract

It is common misconception to identify a motorcycle's frame as a structure that just holds components together, that bears loads or an engineering structure subjected to external forces. The purpose of this thesis is to prove that a motorcycle's frame is a live element, proactive about the handling response and core element to rider's feedback.

Throughout an introduction to mechanics of material, it is explained the behavior of solid bodies under a load, analyzing how a motorcycle's chassis performance is affected and how it reacts with controlled deflections.

For the purpose, it will be used a Honda VFR400 NC30 motorcycle frame due to the availability of either single chassis or a running motorcycle, previously developed and used by university's racing team "Team V4 Racing" in 2015.

The dissertation is developed as the author's opinion over the process that would be followed in the industry for similar analyses. The author acknowledges that the chassis' structural characteristics are identified by its torsional, lateral and longitudinal stiffness values; however, this thesis will focus its attention only to the first one due to the research conducted. Therefore, a torsional stiffness value will be calculated firstly from physical tests data and then evaluated from a numerical model analysis (FEA).

As a validation of the robustness chassis' values previously identified stresses and strains will be analysed, further analysing the influence of several components within the structure.

2 Aims and Objectives

2.1 Aims

The purpose of this thesis is to understand the role of a motorcycle chassis and its physical characteristics that influences its behavior during specific steady state scenarios.

Furthermore, author's interest is to underline the processes used for a reverse engineering analysis approach, highlighting the issues and the limitation normally encountered during a similar 'path'.

The goal of this undergraduate thesis is to gain enough knowledge in order to be applied as a starting point of a further optimization project of either the tools utilized - i.e. the jig, the computational resources or the intellectual analysis' mastery - or the analysed element, the frame.

2.2 Objectives

The thesis is developed around a pragmatistical approach that is validated throughout a computational analysis; the theoretical aspects are presented to explain the development of the argument, so to exploit them for its efficient realization.

On that account, the steps followed for the jig's manufacturing, the tests done, the computational modeling of the frame (CAD) and finally its finite element analysis (FEA) are presented.

However, as it will be highlighted in *chapter 8*, the project was planned so to apply all the results outcoming the previously mentioned fields to a dynamic test: this would have involved the application of strain gauges at first onto the static frame and then to a rolling chassis. The rolling chassis test would have involved basic transient tests as the ones faced in everyday scenarios -e.g. starting from standing still, going through surface disconnections, braking- to finally move on to more extreme ones as on a racetrack.

In conclusion, the data acquired would have been analysed to understand frame's behavior.

3 Timing Plan

Fundamental for an efficient development of a project is its organization which includes risk's foresight since these are what distinguish it from a reiterated process (Smith, 2000).

However, intuitively it is easy to understand that it is impossible to entirely prevent the unexpected because it is peculiar to the development of a new subject/task. Nevertheless, it is possible to minimize risks' influence by breaking down the bigger picture into smaller and tangible goals, the so-called milestones (Maylor, 1996), creating therefore an overall plan that takes into account already possible unplanned delays. This grants the development of a dynamic schedule, which expects a more pessimistic timeline approach, e.g. the machining of a spacer that might take only few hours to be machined is instead made in days.

The overall plan and its sub-tasks is provided in *Appendix A*

Therefore, since the first steps of the project, weekly meetings were hold with the dissertation supervisor for a controlled and constant in time development, allowing to keep track of the progress carried out and dictating the following tasks.

Detailed records of the meetings and task's completion have been kept as proof (*Appendix B*).

4 Background Research

4.1 Understanding a motorcycle 's frame

Implicit in the development of this chapter and the project, is the assumed knowledge of the reader around the basic concepts of a motorcycle. Therefore it will not be discussed the different chassis' designs nor the geometrical parameters that are paramount features for the working principles, i.e. trail, wheelbase, center of gravity, squat and anti-squat etc., and their influence due to the extensive resources already available in the modern literature. This commenced chapter is intended to highlight the role and the more hidden aspects and roles of a motorcycle frame.

A motorcycle chassis' structure must maintain various geometric parameters that characterize the correct and safe behavior of the vehicle according to all conditions. Therefore, it is important to understand the differences between stiffness and strength of the structure. The first is a temporary deformation according if it is loaded or unloaded and it is measured by a comparison between the flexure and the force that cause it (Foale, 2002). On the contrary, the second parameter represents the load that can be applied before failure which is distinguished between breakage or permanent deformation (Foale, 2002).

Failures are a consequence of excess loading, e.g. the ones generated during a crash, or fatigue. Fatigue is a vast and complex subject that influences the performances of a chassis but essentially a good design will experience fatigue failures after the expected life. Furthermore, fatigue failure might occur as well if the structure is under-engineered. Nevertheless, fatigue is an extensive subject that cannot be further extended within this dissertation, the author deliberately avoid discussing it albeit its known importance.

A chassis is a further elastic component on a motorcycle which can be associated as a three coupled suspension system: tires, suspensions and chassis. Before going on exploring the reason why it must have an elastic behavior, it is important to consider where this characteristic comes from: material properties.

The materials used to build chassis can be various and/or a combination of different kinds; metal structures are normally made in steel or aluminum alloys (Bradley, 2003) however nowadays composites can be found either as the main structure or as parts of it. The latter ones, as the name suggests, are a combination of at least two components, normally a resin and a fiber or, for structural parts, a laminate of carbon and aramid cloths (Kevlar) hold together by a resin epoxy based (Bradley, 2003). However, due to the complexity involved, either in their production processes and in the still 'vagueness' of their response under external loads, these solutions will not be considered in this analysis.

On the other hand, most of the frames on the market are metal structures made by alloys, either steel or aluminum. Although, for clarity purposes it is worthy to remember that even steel is an alloy of iron and carbon and many other elements as well. Therefore, it is easy to believe that a variation in the percentages of the elements that composes an alloy can have a big impact over the performances and properties of it (Bradley, 2003) as:

- Strength
- Stiffness
- Ductility
- Brittleness
- Fatigue resistance
- Costs

(Foale, 2002).

The first two parameters have been already defined in relation to a complete structure however, when quantified, they can be used as a yardstick to compare the materials. The tensile stress for instance, is a value to measure strength and embraces multiple parameters: the ultimate tensile stress (UTS) and the yield stress (see *graph 1*). The first is a value at which a material breaks, the latter is the load value where permanent deformation occurs (Hearn, 1977).

As all materials deflects under an external load, the same happens to a motorcycle frame and the amount of this deflection and the rate at which this occurs is crucial. The parameter that describes this change is called *strain* and expresses the proportional variation in dimensions, and is calculated as:

$$\varepsilon = \frac{\delta l}{l_1}$$

Equation 1 Strain

Where the numerator is the length difference from the original length (l_1). The stiffness of a material mathematically is the ratio of the applied load over the deflection as:

$$k = \frac{F}{x}$$

Equation 2 Elastic coefficient

Stiffness is defined by a parameter called Young's Modulus and it can be as well defined as the ratio of stress over strain:

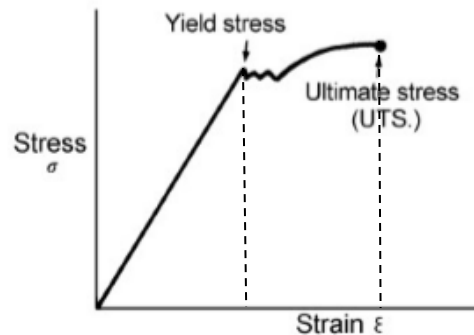
$$E = \frac{\sigma}{\varepsilon}$$

Equation 3 Young's modulus

Moreover, stress is a pressure so Young's modulus equation comprehends a further ratio of load over *area* - for further comprehension see *equation 5* in chapter 4.3.

All these parameters allow the plot of the stress-strain curve that permits to identify the behavior of the material under an external load.

Before proceeding, it is very important to understand how the graph helps to identify the choice of a material over another.



Graph 1 Stress-strain curve graph

Any external load generates stresses: if these are below the yield point it results in an elastic deformation, i.e. the material will go back to its original shape when the load is removed, behaving as a spring. If further load is applied, exceeding the yield point, the deformation will be plastic, i.e. when the load is removed the deflection from its original shape will be permanent. Failure occurs if enough extra load is further applied and material's property wise is represented by the UTS value of a material. Indeed, it is now intuitive picture where motorcycle chassis deflection must occur.

Nevertheless, it is key to keep in mind that failures are as important as all the other engineered features: often, when riding a motorcycle, either on road or on track, unexpected perturbations might occur, i.e. potholes or curbs or worse, and the system reaction even though proximate to failure should be controlled. Therefore, the kind of material's failure is determined by their *ductility* which is the length between the yield point and the UTS (Hearn, 1977) within the stress-strain graph. High ductility values identify a *ductile material* which will undergo more permanent deformation before failure; on the contrary, materials that suddenly fails after the yield point are defined as *brittle* (Hearn, 1977).



Figure 1 Chassis' failures example

As per the above, intuitively is possible to identify which kind of materials should be used for a motorcycle chassis so to withstand overloading without sudden failures.

The above discussion is just a brief outline of a wider argument which must also consider the variation of the material properties according to the technologies used during the

manufacturing process -i.e. heat treatments, influence of welding's types etc.-, that will not be addressed in this dissertation.

It is now time to understand at which extent it is important that a chassis should have a proper structural stiffness, why it must deflect and spring back.

The frame's rigidity is fundamental for either the vital geometrical values and the system handling response; the geometrical values in fact, must be respected to provide the correct and predictable behavior of the system. The elastic characteristic of a chassis is related to its role upon specific conditions during the various steady state motions, e.g. accelerating, braking, cornering etc. Therefore, according to the situation involved, if the stiffness of the frame is inadequate, its reaction forces can generate unwanted scenarios, as the storage of strain energy. As a result, when the external load is removed, because the system is like a spring, the stored energy will be released and, if not controlled/damped, will overshoot and oscillate, i.e. a motorcycle that buckles, weaves and tries to throw the rider (Bradley, 2003).

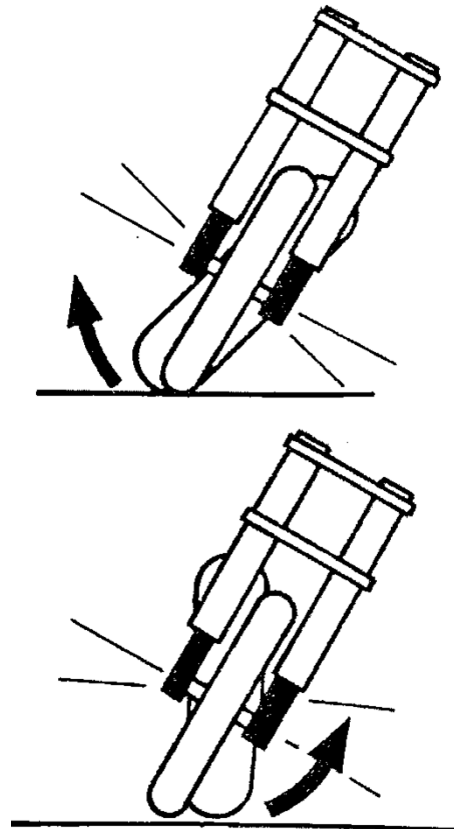


Figure 2 Chassis deformation

Throughout the literature over motorcycle suspensions' theory or suspensions'' manuals, it is often claimed that when a motorcycle leans into a corner the suspension units, due to the inclination, seize up. The system, at this point, is compromised because it is no more providing the required damping action over the surface's unevenness and, consequently, keeping the grip (Thede & Parks, 2010).

In this steady state motion, a controlled deflection of the chassis is a performance savior according to one of the main goals on racetrack: to allow maximum confidence to the rider so to exploit his limits at every part on the racetrack.

In fact, assuming a correct stiffness under upright steady state motions, i.e. braking or accelerating, it is crucial to achieve a *compromise* to cope with the lateral and torsional bending loads and allow high corner speed, paramount feature for a winning rider-bike system. However, the maximum effort experienced by the structure occurs during the transient states motions or under their changes: during these changes of state, the motorcycle structure is rotating around an imaginary axis created by the two contact points wheels-ground, and the

gyroscopic reactions are at their maximum. All rotating elements are identified with the term gyroscope (Bradley, 1996) and yet a motorcycle has several ones: wheels assembly, crankshaft and internal engine components. It is important to understand that any rotating element is identified by its angular velocity (ω) and its moment of inertia (I): the latter identifies its reluctance to change its current state and it is identified by its mass and the distance from the spin axle. The product of the angular velocity and the moment of inertia describes the angular momentum, which identifies a part of the gyroscopic effect, as:

$$M_{angular} = I\omega$$

Equation 4 Angular momentum

Implicit is the understanding that higher angular velocities or heavier masses or bigger radii generates bigger angular moments which can only occur throughout a torque and a reacting force to it. According to the scenario involved, on a motorcycle this role is covered by the chassis.

The angular momentum ① is a vector which length is proportional to $I\omega$, its orientation follows the spin axis and the direction follows the right-hand-screw rule according the sense of rotation. Any variation to the previously mentioned elements will vary vector ① -e.g. speed of rotation-, in figure 3 number ②. Vector ③ represents the variation of angular momentum and applying the hand-screw rule to it, it is displayed the direction of the torque (T) required to change it. As aforementioned, the

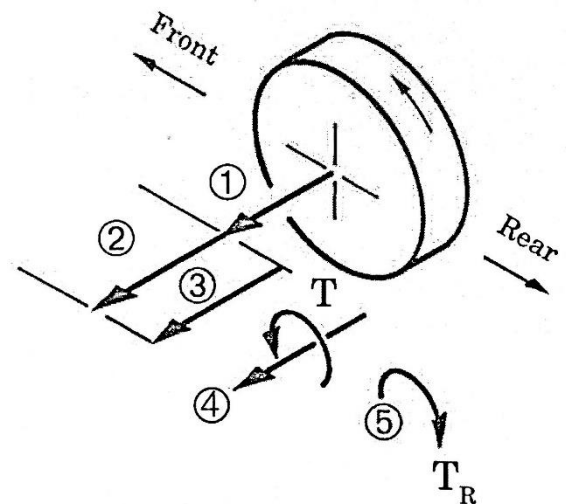


Figure 3 Torque reaction upon the frame (Bradley, The Racing Motorcycle, Volume 2, 2003)

torque can be applied only if there is a resistance that acts against it, therefore there will always be an equal and opposite torque (T_R) generated, that will act upon the chassis. Coherent to the concept of frame's reaction is the understanding that the magnitude of this reaction torque depends on the rate at which the angular momentum variation occurs, i.e. how fast is done. A practical example of this concept is the resistance experienced while leaning into a turn. From the above concept therefore, it is intuitive to understand that the appreciation- or not appreciation- of the behaviour of a motorcycle chassis can only be observed in an environment where fast transients occurs, i.e. racetrack. Furthermore, the reason why similar issues are only experienced by fast racetrack riders, lays in the fact that similar structure's limits are set as high as possible for safety reasons if the motorcycle was firstly engineered and intended for a road use.

4.2 Contemporary topic

Author's interest in the topic finds its reason in the contemporaneity of the subject. In fact, a recent example of frame's rigidities importance is represented by the events that Red Bull KTM Team Ajo, competing in the 2019 world series moto2 class, had experienced throughout the season.

A key aspect that had influenced the inconsistency of team's results during the first part of the season is the lack of chassis development to fit the new powertrain units imposed as mandatory for the 2019 season. In fact, there has been the swap from a 600cc four inline engines to a 765cc inline three cylinders ones.

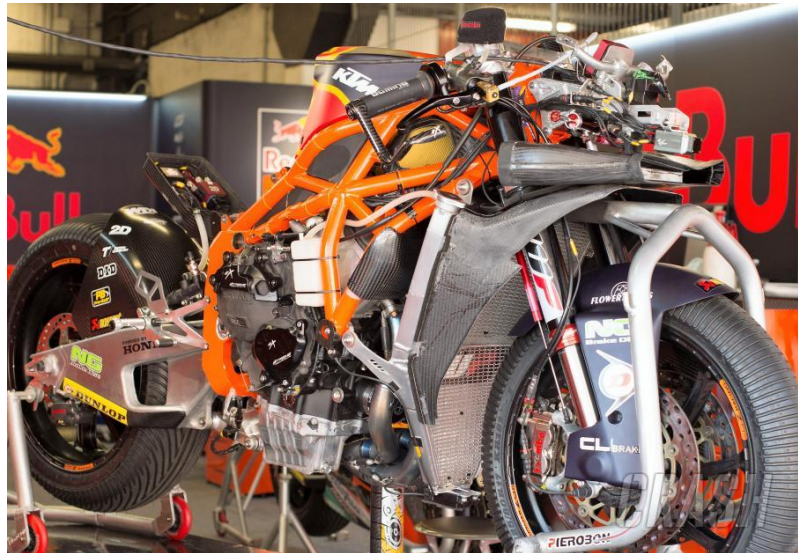


Figure 4 Team Ajo motorsport, moto2

The lack of development not admitted by the Austrian trellis manufacture finds however its confirmation in several ways: the first one is represented by the points over the first seven of out the nineteen races of the championship. In fact, the two riders of the team had gained an average of 6.3 and 0.4 points per race (MotoGp.com, 2019), not considering the experience difference between them. Further confirmation is found in the handling problems highlighted by the riders several times throughout the first races, claiming the necessity *"to make [...] the bike ride easier and improve [...] ergonomics"* (Ajo.fi, 2019). In fact, journalist's photos from the third test sessions of the calendar, highlighted the fitting of new components and parts, e.g. new frame and swingarm (MotoGp, 2019), finding further confirmations in the rider's interviews at the end of the days of testing, stating that *"the parts that KTM have brought worked positively"* (Ajo.fi, 2019). This review of research into the issues experienced by the team, at a closer analysis, reveals that similar substantial updates suggests bigger problems than the ones often claimed: in fact, team members interviewed were addressing the issue to a wrong set-up *"main problem throughout the year was chattering"* (Emmet, 2019). In fact, after the test, riders enhanced that *"we found more rear grip and we can turn better"* (McCallum, 2019) suggesting a positive troubleshooting, finding their rewards in the consistency and quality of the results in the following races, moreover proved by the average points per race, 17.9 and 7.6, and by the battle for the championship title.

4.3 F.E.A.

In order to analyse non-destructively structural behavior and therefore save time by avoiding physical testing, i.e. costs saving, (Hearn, 1977), the engineering field exploits the finite element method which, throughout a mathematical model representation of the element, solves the differential equations, pillars of the structural analysis process (Altair University, 2019) .

The mathematical model is a virtual representation of the analysed element that is broken down into smaller and defined sections, all connected creating a sort of fishing net, the so-called mesh. Every finite element is connected to the next one by discrete points, the nodes, that holds information of the

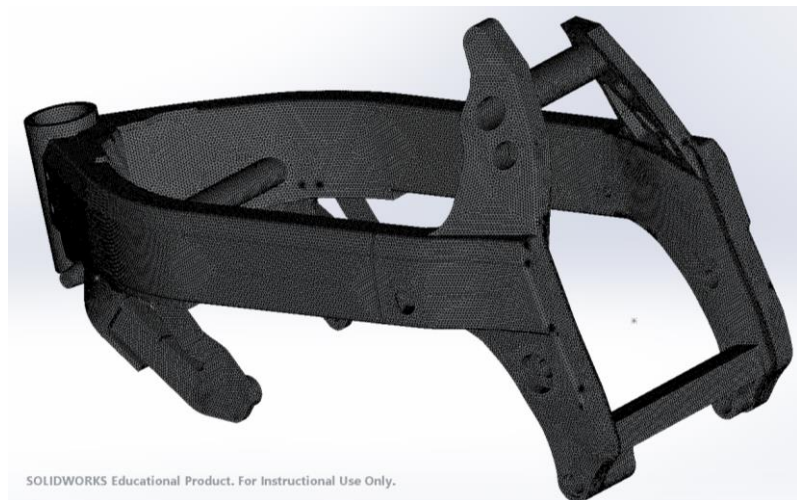


Figure 5 Mesh

element's properties and location within the global coordinate system (Simscale, 2010).

For instance, Young's modulus is one of the parameters hold by each node, and as its formula suggests, multiple set of information are already considered within his definition:

$$E = \frac{\sigma}{\varepsilon} = \frac{\text{stress}}{\text{strain}} = \frac{F}{A} \bigg/ \frac{\delta l}{l} = \frac{\text{Force}}{\text{Area}} \bigg/ \frac{\delta \text{Length}}{\text{Length}}$$

Equation 5 Young's modulus equation breakdown.

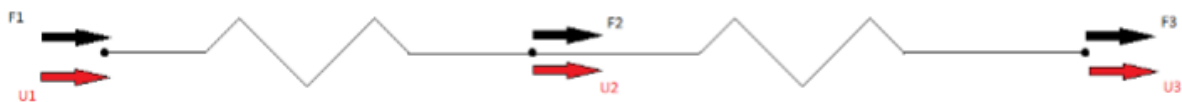


Figure 6 Spring system representation

Therefore, each element holds multiple set of equations that are related to the ones of the near element, so modifying their borders under the external load; the computational power is entitled of solving them simultaneously throughout a matrix form organization (Hearn, 1977). The web of equations can be ideally simplified as a series of multiple springs, which extremes are represented by the nodes, so identified by a force and a displacement, following Hooks law.

$$F_{elastic} = k * x = \text{elastic coefficient} * \text{displacement}$$

Equation 6 Elastic force

Therefore, as outlined above, it is now possible image how, for instance, the overall stiffness value is achieved by the combination of multiple definitions:

$$k = \frac{Area * E}{Length}$$

Equation 7 Stiffness value rearranged

In a 2D system, for instance, each node holds all the information as:

$$\begin{Bmatrix} F_1 \\ F_2 \end{Bmatrix} = \begin{pmatrix} k_1 & -k_1 \\ -k_1 & k_1 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix}$$

Equation 8 Matrix form two nodes element

And the overall system results from the interaction (addition) of each nodes, resulting:

$$\begin{Bmatrix} F_1 \\ F_2 \\ F_3 \end{Bmatrix} = \begin{pmatrix} k_1 & -k_1 & 0 \\ -k_1 & k_1 + k_2 & -k_2 \\ 0 & -k_2 & k_2 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \\ x_3 \end{pmatrix}$$

Equation 9 Overall system matrixial form

Intuitively, from the above, it is possible to imagine the complex web of repetitive mathematical calculations to achieve a single solution and why computational power is necessary to avoid mistakes and obtain time saving, i.e. costs.

Furthermore, for argument's sake, it should be noted that each node holds also the information on its possible movements, the so-called degrees of freedom (DoF) as well, which can be either rotational or translational along its axis (Erochko, 2016); the interpolation of all these information between each nodes are regulated by polynomial *approximations* and the accuracy of the results lays in their degree (Simscale, 2010).

For this reason, it can be said that the parameters influencing the results of a FE simulation are several, however in the following section will be highlighted just the ones that were adopted for the purpose.

This chapter has highlighted the basic concepts around which FEA is based on, however the argument has more complexity behind than the one presented, deliberately not covered for the relevance of the project.

5 Work Carried Out

This project started its first development at the beginning of the academic year by getting in contact with several industries and companies of the field of interest: seventeen, between worldwide factories and smaller ones. The main objective of this action was to gather specific guidance over the entire frame's analysis process: jig's structure and materials, experimental testing solutions and area of interest.

Unfortunately, due to the topic's contemporary constant development, the bigger companies deliberately refused to provide tips or even small data whilst the smaller ones, due to the complexity of the subject and the necessity of specific resources, were not able to give useful guidance.

5.1 The Jig

A jig is “a piece of equipment for holding a tool [...] firmly in position while you work with it” (Cambridge Dictionary, n.d.); as the definition suggests a paramount feature must be its robustness to deflections. However, the realization of the jig, i.e. design and materials adopted, was dictated by the availability in the university's facilities. Therefore, research was conducted to identify the general traits of a tool capable to acquire torsional, longitudinal and lateral displacements of a motorcycle's chassis. However, as the several attempts of getting in contact with the industries had proved, the availability of information over structural analysis' techniques is reduced and, in some cases, its knowledge is a recipe kept secret. For this reason, the final jig's design was a combination of author's and technician's personal judgment and resources available (materials, equipment and tools).

Even though the limited resources did not allow the realization of a heavy-duty jig, it was decided to realize a structure capable of analyzing most motorcycle's frames on the market. To do so, four main bodies were intended to be interchangeable depending on which stiffness values are sought (torsional, lateral or longitudinal) and tailor machined spacers were intended to be used as adaptors, allowing to fix different frames to the jig.

Technical drawings (see *Appendix C*) were realized for those elements and components that needed the usage of machinery restricted to technicians, which therefore involved a waiting list, slowing down the building process.



Figure 7 Flatbed and organization of the layout Rear twisting arm

It was noticed that the twisting mechanism realized and used for the structural analysis of a motorcycle swing-arm (Di Crosta, E; Januario, A. *Motorcycle Swingarm Stiffness and Numerical Analysis* previous author's group project) was the optimal solution to be used: a reinforced 'C' shape section added along the pivot axis of a square box section bar, AISI 1020 cold steel (see figure 7). Moreover, it was acknowledged that the mechanism employed to evaluate the structural stiffness of the same chassis by the university's racing team, Team V4, and presented in the "*Honda VFR400 NC30 frame design*" (Cavalli, 2015) had some unconsidered aspects that had led to errors in the results.

The reason why the author's solution was preferred to the existing one, is linked in its efficiency since:

1. The entire twisting assembly is fixed to a surface, a flatbed, preventing undesired movements.
2. The application of the twisting force is throughout fixed clamping points upon the frame, allowing to always center it and so replicating the experiment consistently.
3. The reaction forces from both clamp arms upon the frame are equal and opposite.

In fact, when conducting a structural analysis, the most important requirement is its repeatability, seeking a method that provides the same results each time. For instance, previous student analysis, despite not fixing the entire assembly to the ground, was also not fixing the twisting element to the frame with the consequent impossibility to consistently repeat the procedure. A schematic representation of the scenario described is provided in figure 8 where the twisting action, even though is applied along a longitudinal mid-plane axis, is not fixed to the analysed structure. Moreover, the

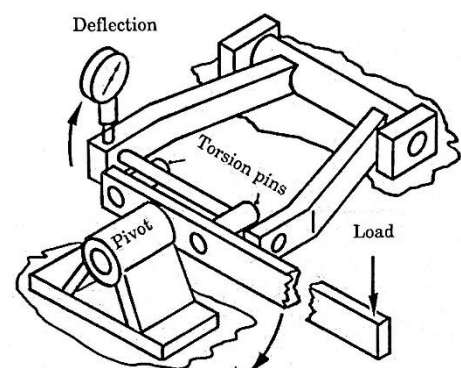


Figure 8 Schematic representation of the solution adopted by previous student analysis. Picture courtesy from Bradley, *The Racing Motorcycle*, Volume 2, 2003

spindles that provides the twisting forces are unrealistically acting upon the swingarm shaft, also causing undesired stresses on an useless element for the conducted analysis so affecting the quality and accuracy of the results.



Figure 9 Machining

A key feature to a correct structural evaluation, as when a frame is built, is the perpendicularity between planes, i.e. headstock axis must be perpendicular to swing-arm pivot (Bradley, 2003). The starting point was the correct alignment of the frame's elements. To do so, were first machined two 6060 T6 aluminum headstock spacers, with a threaded hole to screw the headstock shaft to the jig (see *figure 9*), then the rear 19 mm swing-arm rod and finally its adaptors to the existing twisting-arm mechanism. All of these components were machined and manufactured to allow an accurate perpendicularity between headstock's and swing-arm's pivot axes throughout the usage of a digital inclinometer (see *figure 10*).



Figure 10 Horizontality check throughout digital inclinometer

The headstock's and swingarm's spindle diameter sections were defined according to the guidance given by Raines M., Roe G.E., Thorpe T.E., 2011 paper, where it is specified that a 21% increase in the original section of the shaft gives a 10% increase in the overall stiffness, thus the usage of OEM dimensions.

To prevent bending and buckling of the rear structure (an AISI 1020 cold steel) due to heavier elements involved, two extra supports-in the same material- were added to the original design (see figure 11).

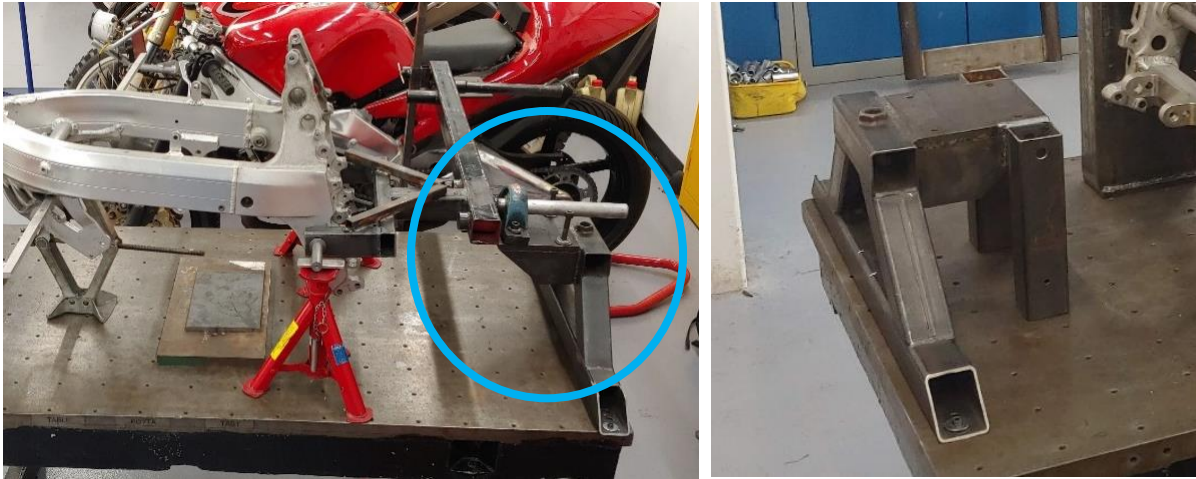


Figure 11 Original and modified version of the rear twisting assembly design

It was then built the front holding section, where an AISI 1020 cold steel post plays the main structural role and, attached to it, an AISI 1010 hot steel "I" beam was bolted to it to clamp the

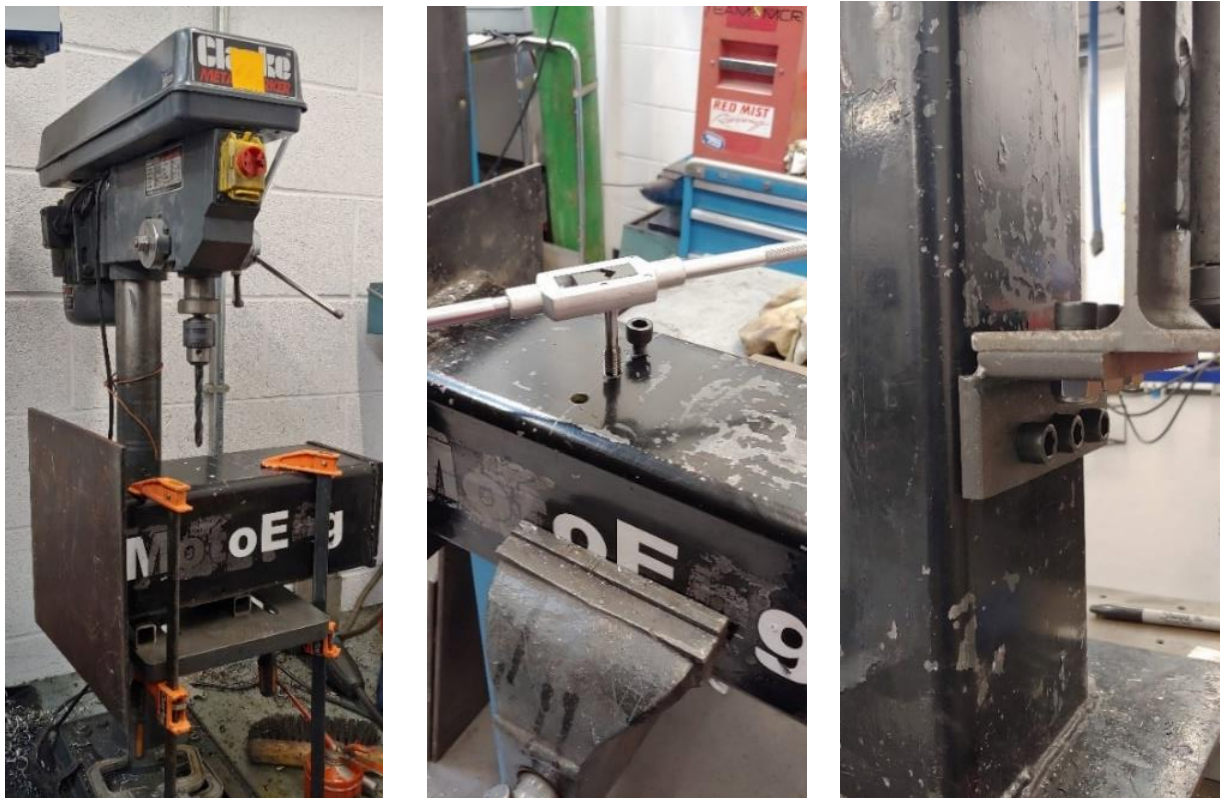


Figure 12 Front holding post

frame's headstock section. Further fixing points were added at the top and bottom throughout "L" shaped gussets (*figure 12*).

To prevent the rising of angular moments from the rear twisting action, two AISI 1020 cold steel plates with four M12 screws were added at the front jig's section. These elements were intended as a clamp, acting upon the casted frame's headstock section, preventing it to pivot.

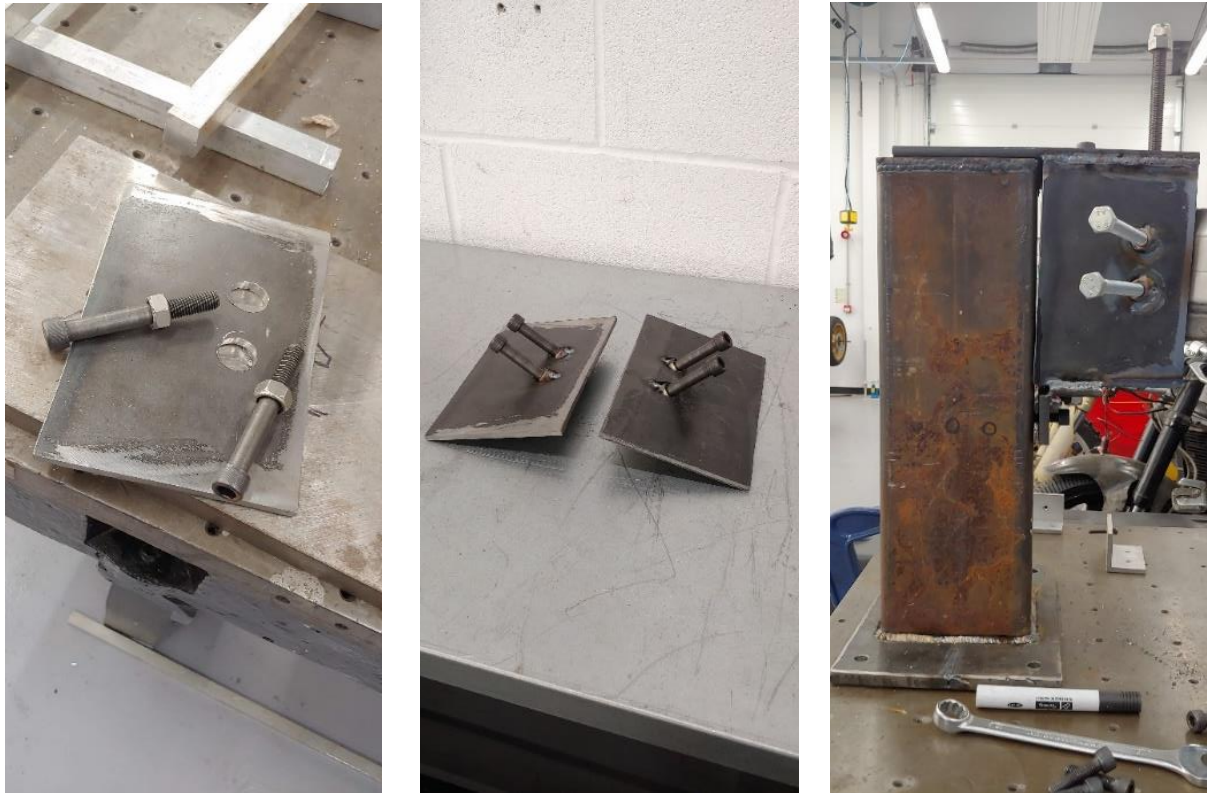


Figure 13 Pivoting lock-mechanism

Further attention was reserved to prevent angular momentum reactions of the rear section as well, throughout a digital inclinometer, checking the parallelism with the surface.

Furthermore, for the front twisting experiment, two halved AISI 1020 cold steel pillar's sections were used as rear holding structure. Along their mid-axis, a hole was drilled to fit the rear custom-made swingarm shaft.



Figure 14 Front twist rear pillars

The small tolerance between parts (halved posts and shaft) was achieved by the usage of plain carbon steel spacers. Furthermore, to prevent distortions, the two rear structures were linked together throughout an AISI 1010 hot steel plate.

The front twisting clamp was built with the same principles used for the front section of the rear twisting layout experiment: it has the same wheelbase and its pivoting axis was intended to be at the centre point of the rectangle shaped plate.



Figure 15 front twist rear spacers and connecting bar

The lack of availability of another “I” beam section, within the workshop facility, led to the manufacturing of a custom made “C” shape element from an AISI 1010 hot steel plate. During the process four gussets at each corner were added to prevent bending. The same reinforcement principles were applied to the main AISI 1020 cold steel post by the application of AISI 1010 hot steel gussets plates at its base. The building process had found its challenge in trying to respect the pre-planned time schedule, due to the time constraints imposed by technicians’ academic commitments: for instance, the machining of a small and simple shaped element sometimes had stopped the building process due to its paramount role.

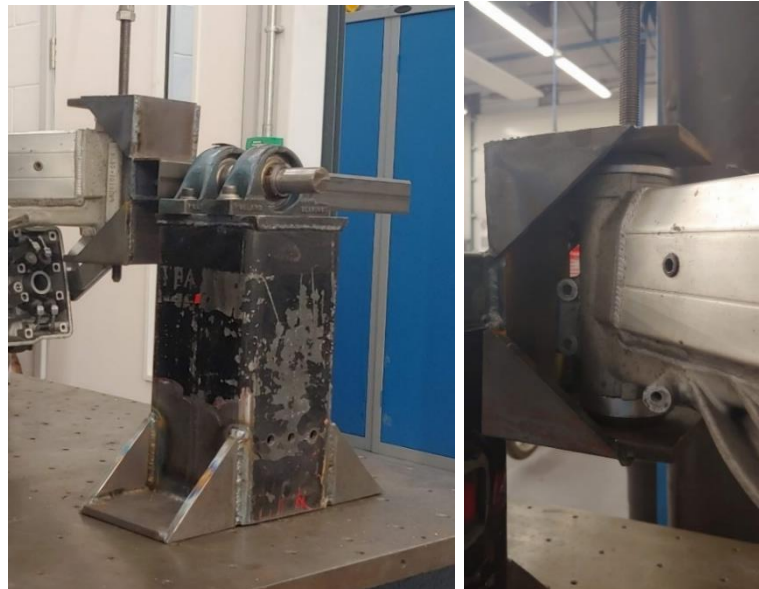


Figure 16 Front twist mechanism and gussets

5.2 Experiments testing

The experiment performed was designed to evaluate the torsional rigidity of the frame, so to mimic those steady state motions where traction or mechanical grip is paramount to the performance. On a motorcycle, most of the external influences tend to twist the structure (Robinson, 1994) despite the numerous ways a component can be deformed. For this reason, an element with a high bending stiffness might not have a high torsional robustness as well and it all depends on how the loads are applied (Hearn, 1977). For the purpose of the project, the influence that forks flexure has upon the overall structure was ignored. Their stiffness analysis is a parallel topic that will not be discussed in this project, however it is known that their oscillation effect upon the steering axis generates a gyroscopic reaction (Bradley, 2003). Moreover, by the author it is also recognized the great influence over the overall performance that higher quality suspension units has over standard components, i.e. less deflection.

The two tests have been conducted by fixing the jig's components at the flatbed throughout the usage of mill's clamp vices. Lateral and longitudinal physical tests due to material and time constrains were never intended to be conducted at this stage. However, the torsional test and its FEA validation would have exploited to perform virtual tests, following the already designed solutions for these physical tests (*see Appendix D*).

Both empirical tests were conducted by fitting the frame at the jig, respecting the torque values as on a motorcycle and using a scale to measure the twisting force applied throughout a scissor jack. Furthermore, four dial gauges were placed upon specific points on the frame to read the deflection under the applied load; dial gauges were also used to read jig's deflections as well - more of this later.



Figure 17 Rear and Front twist experiment layout (with engine)

Either rear and front twisting tests were first conducted with the engine bolt in the chassis and then repeated without it, in order to understand its influence over the frame's robustness.



Figure 18 Rear and front twist experiment (without engine)

When performing the test with the engine in, it was furthermore studied the effects that the engine's fixing points has as well: in fact, both front and rear twisting tests were repeated by unscrewing the middle engine mounts, connecting the front part of the rear cylinder head to the chassis. In fact, the frame available had eight engine's fixing points, two for the front cylinder head, four for the rear one and two for the rear lower section of the engine case.

Furthermore, the load values applied resulted in the same momentum as previous student analysis (Cavalli, 2015) upon the same frame, so to evaluate both set-up and compare results -more of this in chapter 6.1.

The experiments were conducted by applying 5kg at the time, up to a maximum of 60kg, since single readings could have been misleading.

Unfortunately, as previously mentioned, there is no standard procedure to run similar experiments therefore, comparisons between different sources might be misleading, especially because the values are length dependent. However, the loads applied were the same as previous student analysis (Cavalli, 2015) due to the same guiding parameters used, i.e. same twisting arm, frame, dial gauges positioning, moments applied, but this thesis exploits a more efficient and reliable jig's design.

5.3 3D Modelling

Reverse engineering process, as the one here conducted, revolves around approximations and/or assumptions over certain areas due to lack of information or disclosure agreements between the OEM factory and the TIER supplier (Maylor, 1996). This is due to the companies' policy to preserve the winning or the optimal recipe for a specific scenario.

Therefore, to maximize the accuracy of a CAD representation of the analyzed model, the starting point should be a 3D scan of it.

The output of the 3D scan was a point cloud OBJ file format, that has been exploited as base guideline around which the solid model was designed.



Figure 19 3D scan

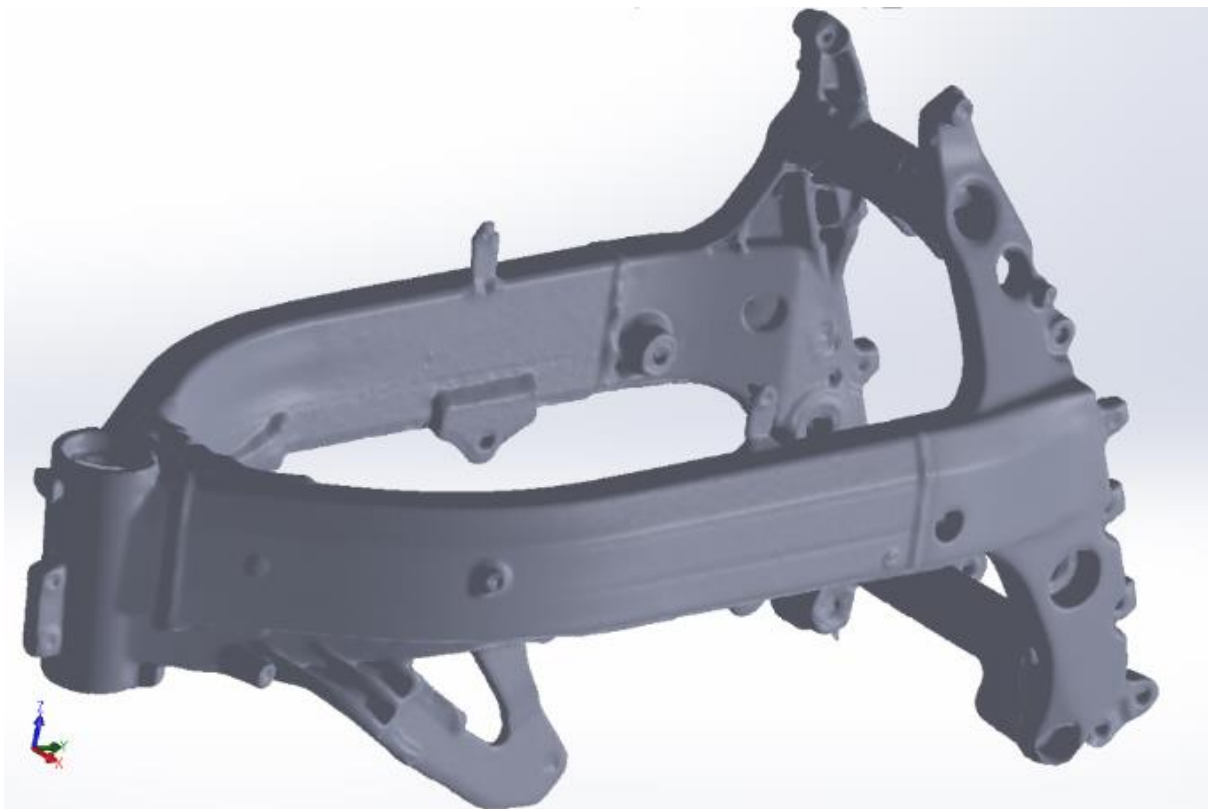


Figure 20 Point cloud of the 3D scan

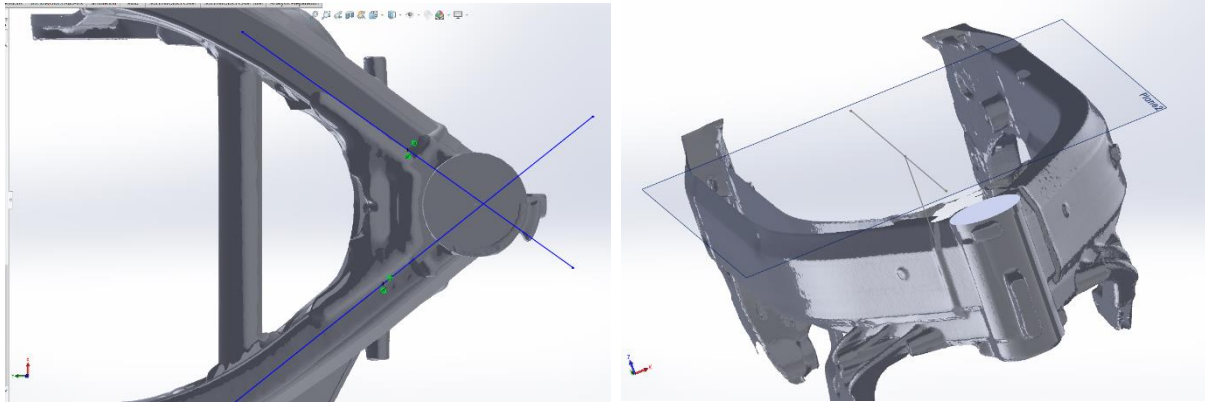


Figure 21 3D modelling asymmetry

During the design process, it was possible to appreciate how, even though the appearance suggests differently, the frame was not a symmetrical element.

The above screenshots, for example, show some of the misalignments. It is highlighted how the two main structural bars, connecting the headstock to the rear section of the frame, neither have their axis meet the headstock's axis nor do lie upon the same plane. Intuitively, these geometrical differences suggest the difficult task of respecting the alignment's parameters during the building process, where thermal expansion and contractions are the main enemies that rises (Bradley, 2003).

Furthermore, especially for aluminum alloy frames, another issue causing distortion is the combination of either casted and extrusions or forged and pressed sheets plates and vice versa (Bradley, 2003). However, this process reduces significantly the strength of the parts due to the change of material condition under the heat (Hearn, 1977) and the ideal rigidity is provided back with stiffening webs (Bradley, 2003). Clearly, three dimensional scans provide a very accurate representation of the external surfaces but not of the internals of the elements; consequently, during the modeling process the stiffening webs were designed and included assuming to be as the ones in figure 22 due to the similarity in the measures showed and the ones of the physical frame. The presence of these "invisible" elements had found its confirmation in CAD models of previous students analyses of the same frame and in the author's physical tests -more in chapter 6.1.

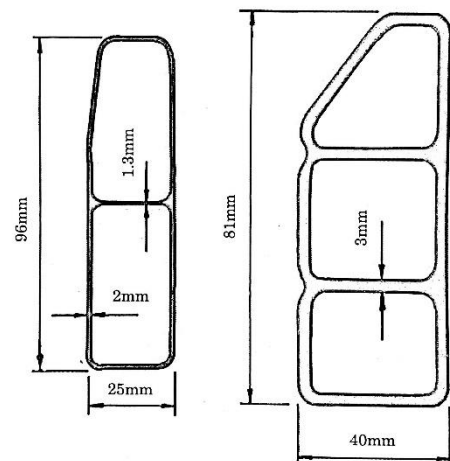


Figure 22 Stiffening internal webs, picture courtesy Bradley, *The Racing Motorcycle*, Volume 2, 2003

The end CAD model is represented in figure 23: two models were generated according to accuracy benefit, that will be explained in the following section.

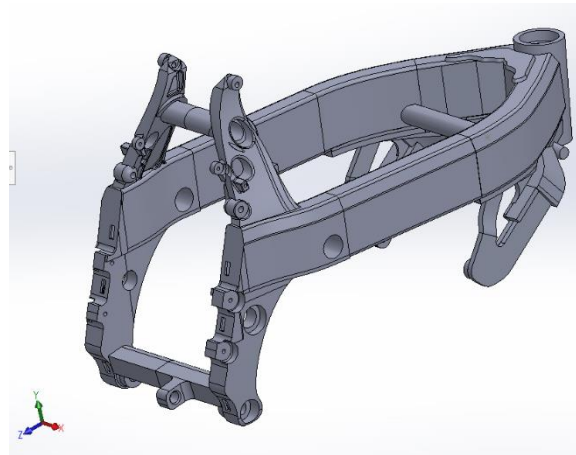


Figure 23 Accurate 3D representation

5.4 Numerical analysis – F.E.A.

The static analysis conducted was performed upon Solidworks2019 simulation package.

For the structural analysis, the designed frame was simplified in its features as in figure 24 shows, to reduce the computational timing and because certain characteristics would not affect the results as well.

To perform the simulation, it was necessary to define parameters and constraints: the material was defined first. Aluminum alloy frames tends to be a combination of casted and forged elements and extrusions or pressed aluminum sheets according to the period of production, where the latter is a more recent and advanced technology. The analysed frame is made by a combination of extrusion bars and casted sections for the headstock and the entire rear section. Within the structure there are also aluminum plates as well as for the engine brackets.

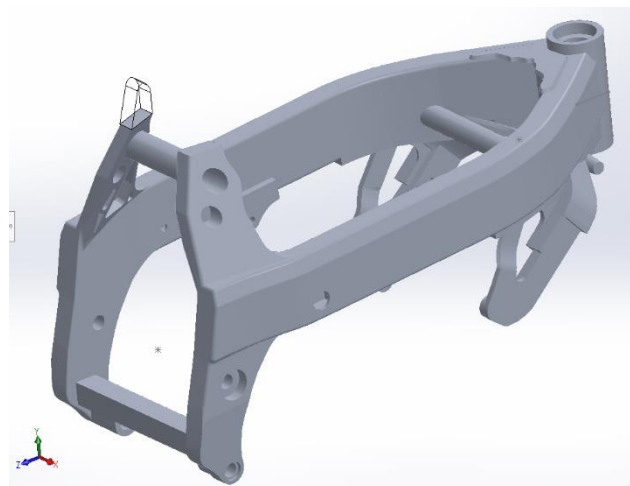


Figure 24 Frame's 3D representation, simplified version

Unfortunately, as already mentioned, reverse engineering processes over production vehicles due to lack of specifications released involves assumptions over the kind of material used. Research over motorcycle's frame building processes had given general knowledge where the largely aluminum types are used: 6061-O or 7020 (Bradley, 2003). However, the latter is

an aluminum alloy largely used into the aero sector, therefore its exotic characteristic allows its usage into the motorcycle sector for those premium vehicles.

By contrast, with reverse engineering processes, the only known values about the analysed chassis are the physical measurements, for instance the weight: 11,2 kilograms.

In addition to the material's property differences, the mass property tool within the CAD software highlights how with either material choice, the frame's weight is 10,8 kilograms. It could be said that the 0.4 kilograms difference from the physical model, is an acceptable 3% difference, especially considering the assumptions carried out throughout the process for defining the structure's shapes. Therefore, for the analysis purpose the author was satisfied with the features and characteristics adopted to run the simulations and consider the results reliable.

Were then defined the fixtures according to which experiment was replicated, either it was the front or the rear twist. For the first one, two fixed constraints were applied to the rear swingarm's shaft holes, whilst for the second scenario at the headstock's lateral bearing surfaces.

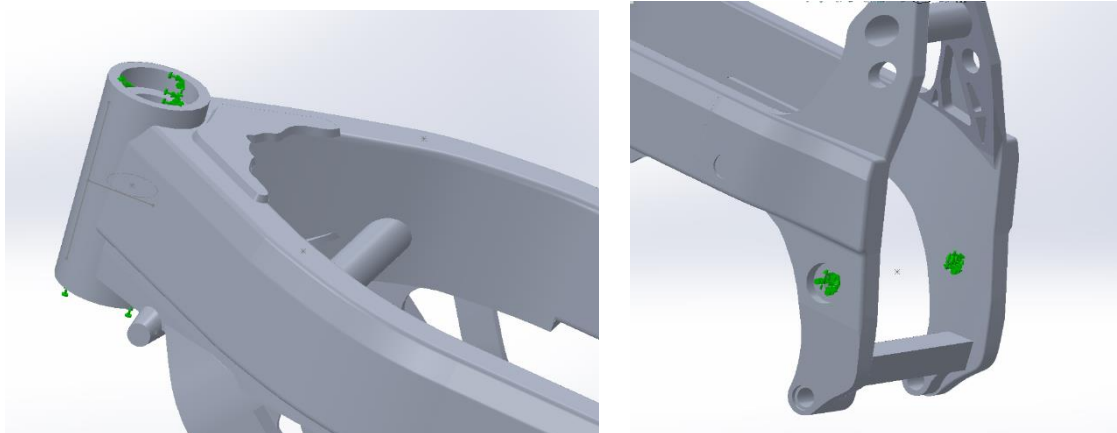


Figure 25 Rear and front twist test fixtures

The application of the external loads was done throughout the usage of two opposing forces or with the application of a torque, either way the results were the same.

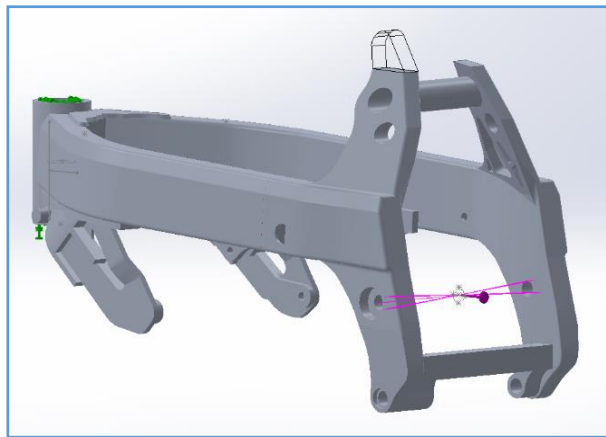
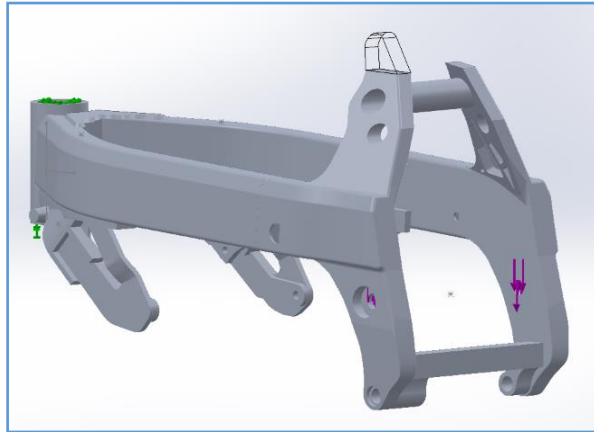


Figure 26 Rear twist loads through force (top) or torque (bottom)

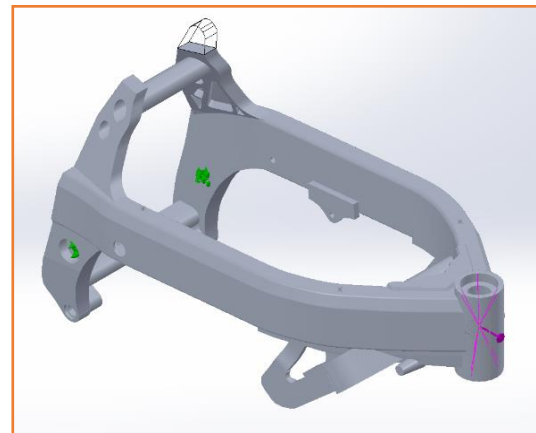
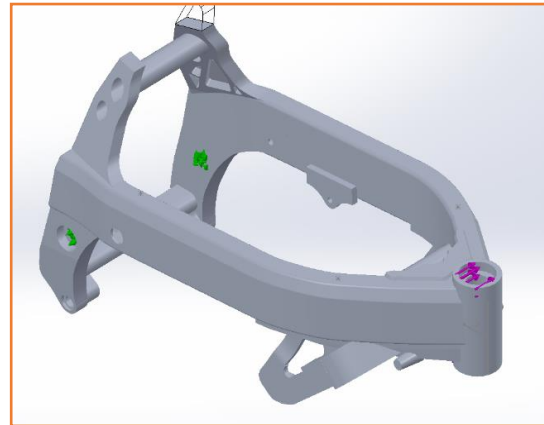


Figure 27 Front twist loads through force (top) or torque (bottom)

The magnitude of the loads was calculated at the application point, due to the fact that the simulations were run without the presence of the jig and its elements – which presence would have allowed the usage of the same loads as in the experiment. Therefore, for either the front and rear twist, it was first calculated the moment around a fictitious frame's pivot mid-plane by the application of the force at a distance of 0,73 m. This value was then used for the simulation employing a torque load and on the contrary, when the two opposing forces were used, it was necessary distinguish both experiment cases. For the rear twist instead, it was assumed that the forces were acting at the same distance from the mid-plane pivot axis, even though in the previous section it was highlighted the no-symmetry of the frame. The mesh setup was mainly dictated by the lack of computational power -more of this in the following chapters. However, it was used a 3 mm blended curvature-based mesh.

6 Results and analysis

The optimal method to evaluate the structural stiffness of a motorcycle's frame should be conducted with the engine fitted (Cossalter, 2006); however, a first analysis is conducted by comparing the results of the previous data acquired by Cavalli's dissertation, where the frame was left engine-less (Cavalli, 2015).

Torsional frame stiffness reference values have different sources, $K_t = 3 - 7 \text{ kNm}/^\circ$ (Cossalter, 2006), $K_t = 2,5 \text{ kN}/^\circ$ (Bradley, 2003), $K_t = 2,7$ (Foale, 2002)

Torsional frame's stiffness is calculated as:

$$K_t = \frac{F}{\alpha}$$

Equation 10 Torsional stiffness

where α is the angle created by the deflection of the frame. As previously mentioned, these values are length related therefore, it all depends on where the displacements are read.

6.1 Rear twist torsional test

The rear twist test was conducted in different layouts: with the engine installed, with the engine installed but with not all engine's bolts installed and without the engine. However, the latter test was only conducted to compare previous student test results. Dial gauges (DL) have been used to identify deflections of either the frame and the jig's front post, identified with numbers as *figure 28* show.

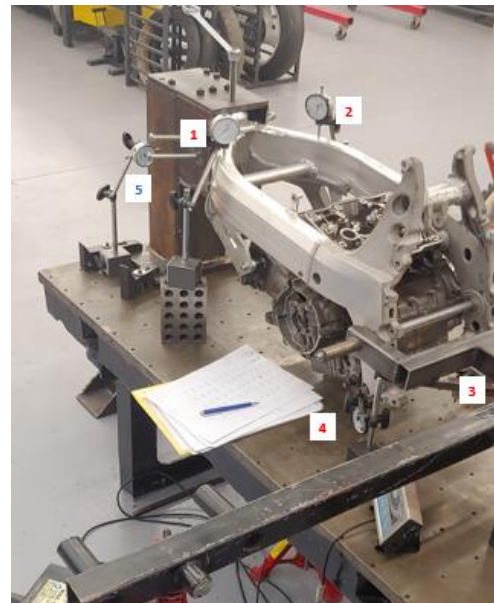


Figure 28 Dial gauges nomenclature

REAR TWIST OFF ENGINE						Cavalli's Test		
LOAD	DISPLACEMENT [mm]					LOAD	DISPLACEMENT	
kg	DL 1	DL 2	DL 3	DL 4	DL 5	kg	DL 3	DL 4
5,35	0,185	-0,14	0,05	-0,31	0	1,95	9	-9
10,15	0,28	-0,17	0,01	-0,55	0	10,16	25	-24
15,05	0,36	-0,23	0,05	-0,69	0			
20,2	0,44	-0,3	0,11	-0,84	0	20,32	43	-43
25,15	0,525	-0,38	0,2	-0,96	-0,05			
30,15	0,6	-0,45	0,27	-1,09	-0,05	31,66	59	-58,5
35,1	0,73	-0,52	0,34	-1,2	-0,08			
40,1	0,74	-0,59	0,41	-1,33	-0,1			
45,3	0,81	-0,66	0,47	-1,47	-0,14	43	78	-77
50,3	0,85	-0,69	0,49	-1,58	-0,15			
55,35	0,91	-0,75	0,53	-1,73	-0,19			
60,25	0,96	-0,78	0,56	-1,865	-0,2			
-	-	-	-	-	-	65	116	-115,5

Table 1 Rear twist test comparison with previous student's results

Table 1 presents the deflection's values of the authors test compared to the previous student ones. It is immediately clear the paramount role of a well-built jig. It has been already highlighted in previous chapters the issues that the previous student's test-rig solution rises and finds confirmation in the frame's displacement results.

EXPERIMENT REAR TWIST NO ENGINE					Cavalli's Test		
	DL 1	DL 2	DL 3	DL 4		DL 3	DL 4
load [kg]	60,25				load [kg]	65	
force [N]	591,1				force [N]	637,7	
lever arm [m]	0,73				lever arm [m]	0,73	
T@midplane [Nm]	431,5				T@midplane [Nm]	465,5	
avg dist lever arm	0,1275				avg dist lever arm	0,1275	
force@FRAME [N]	1692,03				force@FRAME [N]	1825,4	
displacement read	0,96	-0,78	0,56	-1,865	displacement read	116	-115,5
distance DL	0,242		0,282		distance DL	0,282	
α [degrees]	0,45	0,37	0,23	0,76	α [degrees]	55,4	55
K [N/degrees]	3760,1	4573,1	7356,7	2226,4	K [N/degrees]	32,9	33,2

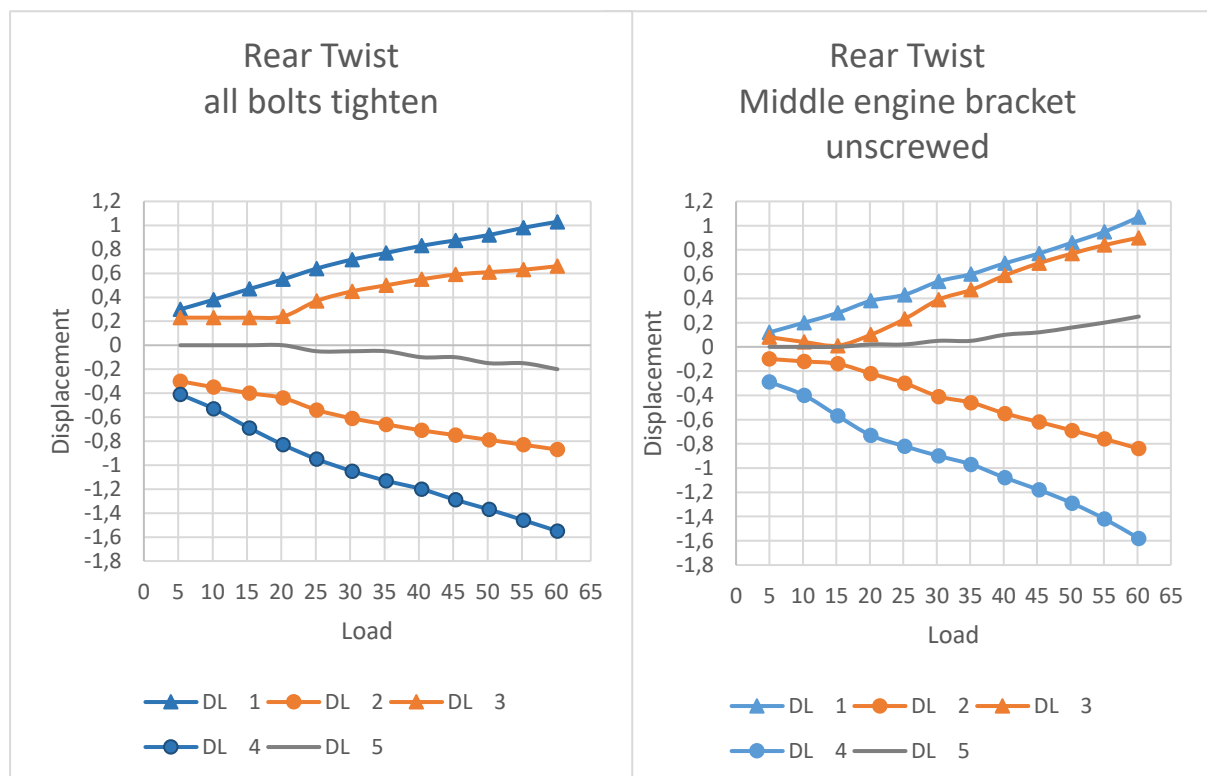
Table 2 Stiffness' values comparison

Table 2 above further enhances the unreliable results from the previous test, highlighting unacceptable stiffness values; the absence of displacement's readings of the jig does not allow to evaluate correctly the test. It must be highlighted that from photos of the previous student's test, the dial gauges upon the frame were in the same points as the author's ones therefore, it was assumed the same distance from the pivoting mid-plane, allowing the calculation of the angle α .

On the contrary, here are presented the experiments data with the engine installed and in different layouts: all engine's bracket bolts tighten and rear head bolts unscrewed -i.e. the rear cylinder has four attachment points at the frame, here are unscrewed the front pair.

REAR TWIST ENGINE IN, all tight							ENGINE IN & middle bracket UNSCREW						
LOAD	DISPLACEMENT [mm]						LOAD	DISPLACEMENT [mm]					
kg	DL 1	DL 2	DL 3	DL 4	DL 5		[kg]	DL 1	DL 2	DL 3	DL 4	DL 5	
5,3	0,3	-0,3	0,23	-0,41	0		5	0,12	-0,1	0,08	-0,29	0	
10,15	0,38	-0,35	0,23	-0,53	0		10,2	0,2	-0,12	0,04	-0,4	0	
15,35	0,47	-0,4	0,23	-0,69	0		15,2	0,28	-0,14	0,01	-0,57	0	
20,25	0,55	-0,44	0,24	-0,83	0		20,15	0,38	-0,22	0,1	-0,73	0,02	
25,15	0,64	-0,54	0,37	-0,95	-0,05		25,2	0,43	-0,3	0,23	-0,82	0,02	
30,3	0,715	-0,61	0,45	-1,05	-0,05		30,3	0,54	-0,41	0,39	-0,9	0,05	
35,25	0,77	-0,66	0,5	-1,13	-0,05		35,15	0,6	-0,46	0,47	-0,97	0,05	
40,4	0,83	-0,71	0,55	-1,2	-0,1		40,2	0,69	-0,55	0,59	-1,08	0,1	
45,35	0,875	-0,75	0,59	-1,29	-0,1		45,3	0,77	-0,62	0,69	-1,18	0,12	
50,25	0,92	-0,79	0,61	-1,37	-0,15		50,25	0,86	-0,69	0,77	-1,29	0,16	
55,25	0,98	-0,83	0,63	-1,46	-0,15		55,1	0,95	-0,76	0,84	-1,42	0,2	
60,15	1,03	-0,87	0,66	-1,55	-0,2		60,2	1,07	-0,84	0,9	-1,58	0,25	

Table 3 Rear twist experiments



Graph 2 Rear twist comparison

From table 3 and graph 2, it can be noted the influence of an extra engine bracket fixing point: the variation of frame's rigidity calculated in fact varies between 5% and 25%. Furthermore, the front post's deflection is read at the same height of the headstock's center point, between the top and bottom bearing's housing. Consequently, its deviation from the original vertical position only occurs at 66% through the test, reaching a peak value of 0.2 – 0.25 millimeters, which the author consider an acceptable value.

Deflections values are read at the same points upon the frame during both experiments. It is peculiar how their magnitude is the same (see table 3) however, by contrast the gradient of their curve is different, i.e. maximum deflection is reached at different loads (see graph 2).

It seems that the frame with the unscrewed middle bolt tends to damp more the external loads especially at lower load magnitudes, then reacting with more robustness at higher loads. However, an important data missing is the reading of the displacement of the rear twisting arm; additionally, it must be pointed out that this lack is due to the availability of extra pair of dial gauges within the workshop facility; nevertheless FEA will be used to evaluate in more dept and accuracy the deflections of the jig as well. Although, the graphs allow a first analysis of the jig's behavior.

Up to 25% through the test, the DL3's readings highlight some unexpected gradient of the curve, suggesting some minimal flecion in the rear twisting arm, damping the application of the force, leveling out at higher force values. On the whole, the graph highlights expected frame's movements where, in the case of a clockwise moment, the left-hand side tends to rise and the right to lower. Table 4 shows the calculated stiffness values of the frame following equation (1).

REAR TWIST ENGINE IN all tight				
	DL 1	DL 2	DL 3	DL 4
load [kg]	60,2			
force [N]	590,1			
lever arm [m]	0,73			
T@midplane [Nm]	430,8			
avg dist lever arm	0,1275			
force@FRAME [N]	1689,2			
displacement read	1,03	-0,87	0,66	-1,55
distance DL	0,242		0,282	
α [degrees]	0,49	0,41	0,27	0,63
K [N/degrees]	3447,4	4120,1	6256,4	2681,3

ENGINE IN & middle bracket UNSCREW				
	DL 1	DL 2	DL 3	DL 4
load [kg]	60,20			
force [N]	590,6			
lever arm [m]	0,73			
T@midplane [Nm]	431,1			
avg dist lever arm	0,1275			
force@FRAME [N]	1690,6			
displacement read	1,07	-0,84	0,90	-1,58
distance DL	0,242		0,282	
α [degrees]	0,51	0,40	0,37	0,64
K [N/degrees]	3315,0	4226,6	4569,3	2641,6

Table 4 Rear twist stiffness values

The angle α values were identified throughout basic 2-dimensional representation of the frame's displacements (see figure 29), assuming the rotation occurred around a longitudinal midplane.

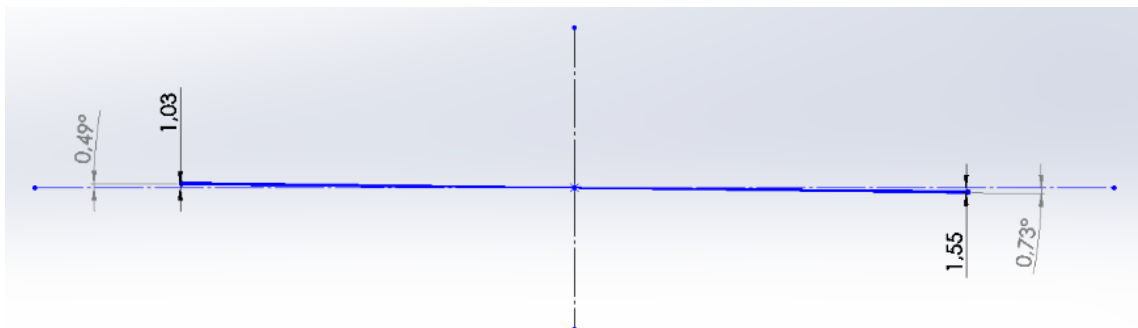


Figure 29 Angle identification schematic

The calculated frame's stiffness values can be considered correct because within the indicated range by several authors. Nevertheless, as mentioned, torsional stiffness values are

length dependent and the table highlights the concept showing the value's difference according if the measurements are taken from the front or the back section of the frame, relatively DL1 and 2 and DL 3 and 4.

Furthermore, it is possible to appreciate a possible deflection of one of the two rear twisting arms by noticing the very low frame's stiffness value given by the DL4 readings.

6.2 Frontal twist torsional test

The front twist test was conducted as for the rear one either with the engine installed and without, mainly for consistency with the amount of data acquired. However, the analysis will be focused only upon the scenario with the engine installed due to the validity of data acquired as previously explained. As for the rear test, the frame's deflections are read at the same points whilst the jig is analysed for lateral movements around its rear holding pillars. In a perfect scenario DL5 and DL6 are not supposed to read the jig's lateral displacements as DL 3 and DL4 the vertical displacements of the frame's rear section.

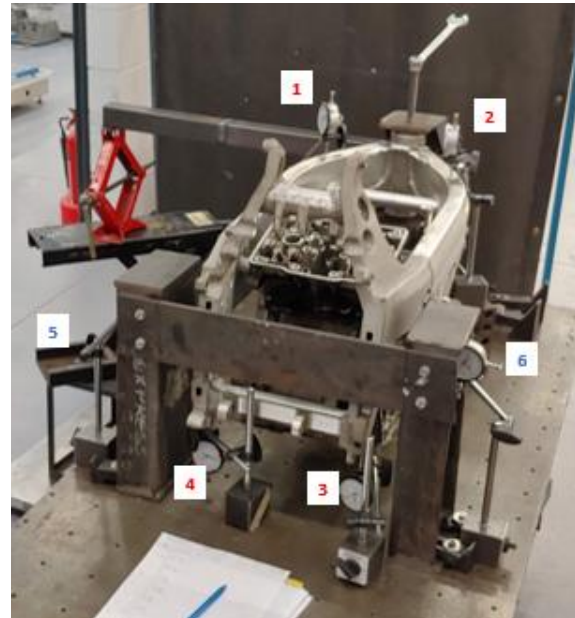


Figure 30 Front twist DL's layout

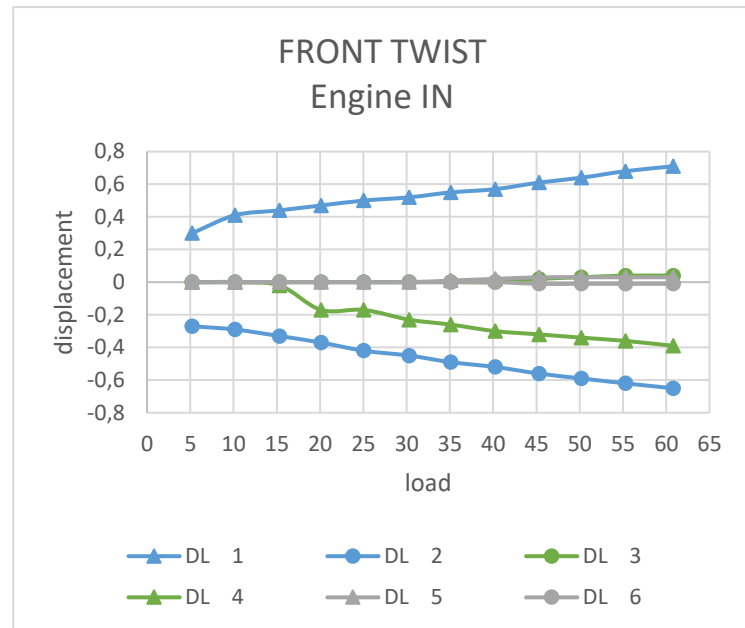
However, some movements are read but, how it will be explained in the following section, can be neglected.

As *graph 3* highlights, the frame's front part is flexing as expected and the linearity of the curve's gradient suggests a good performance of the jig's stiffness. However, it could be pointed out DL4's higher displacements readings compared to its counterpart DL3 which are approximately to none: the explanation is found in *figure 31*.

FRONT TWIST ENGINE IN, all tight						
LOAD	DISPLACEMENT [mm]					
[kg]	DL 1	DL 2	DL 3	DL 4	DL 5	DL 6
5,2	0,3	-0,27	0	0	0	0
10,15	0,41	-0,29	0	0	0	0
15,3	0,44	-0,33	0	-0,02	0	0
20,1	0,47	-0,37	0	-0,17	0	0
25,05	0,5	-0,42	0	-0,17	0	0
30,3	0,52	-0,45	0	-0,23	0	0
35,1	0,55	-0,49	0	-0,26	0,01	0
40,25	0,57	-0,52	0	-0,3	0,02	0
45,3	0,61	-0,56	0,02	-0,32	0,03	-0,01
50,2	0,64	-0,59	0,03	-0,34	0,03	-0,01
55,3	0,68	-0,62	0,04	-0,36	0,03	-0,01
60,8	0,71	-0,65	0,04	-0,39	0,03	-0,01

Table 5 Front twist displacement readings

The fourth dial gauge's 0.4 millimeter maximum movement finds its explanation in the collapsed metal shim used to fill the very tight gap between swingarm custom machined axle and the frame itself. The tolerance between the two components was acknowledged only after the axle was machined and, whilst the wider hole for the swingarm adjuster allowed the machining of a 6061 T6 aluminum spacer, it was not possible to do the same on



Graph 3 Front twist experiment displacements

In fact, the reduction of the axle's diameter, to allow the usage of a similar solution, would have provided more important and severe aspects, i.e. structural failures, already discussed in chapter 5.1. Nevertheless, this solution was used as well for the rear twist experiment and some unexpected results can now find their reason and explanation.

Therefore, the 0.06 millimeter difference between DL1 and 2 can be ignored; also negligible are the lateral displacement read by DL5 and DL6, thus proving a good performance of the jig's structure. Table 7 shows values used to calculate the front stiffness value of the frame. It is so clear how the overall value is out of the reference's values, highlighting an over-rigidity of the frame. However, it must be kept

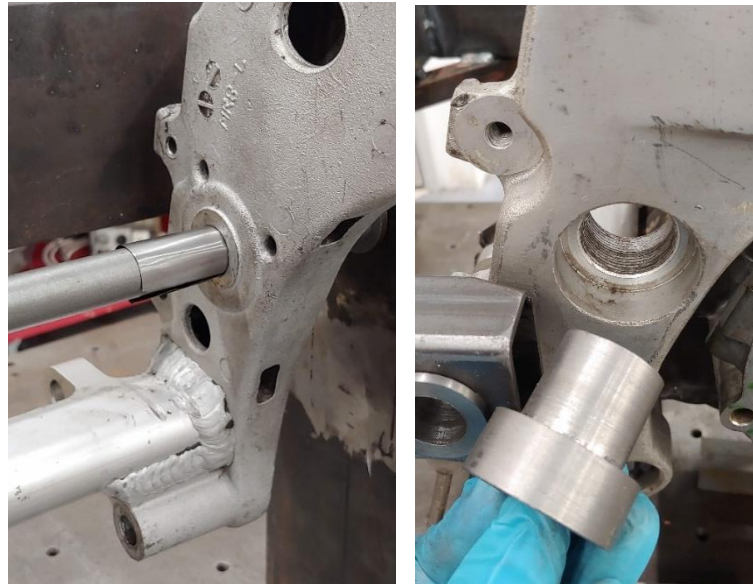


Figure 31 Metal shim and spacer solution

into account that the process to acquire the values does not reflect an actual scenario that the frame undergoes under working conditions. In fact, there is no such condition while the motorcycle is on track or on the road where the rear section of the frame is lock in one position, nevertheless is not a value that is meaningless. As aforementioned, the necessity to reduce the harsh feeling is often discussed (Gubellini, F; Gubellini, Diego; Finelli, A; AIM-Sport, nd)

to enhance riders' confidence, especially entering a turn; therefore, this value might come handy to understand what it is its influence in this steady state, assuming all the other variables -damping set-up, geometries, tires' pressure, rider psychological state etc.- are "in the ball park". By contrast, the author also wants to highlight that is not a value that should be analysed on its own or used to identify a frame's characteristic.

FRONT TWIST EXPERIMENT ENGINE IN, all tight				
	DL 1	DL 2	DL 3	DL 4
load [kg]	60,80			
force [N]	596,45			
lever arm [m]	0,73			
T@midplane [Nm]	435,41			
avg dist lever arm [mm]	0,06			
force@FRAME [N]	3628,39			
displacement read [mm]	0,71	-0,65	0,04	-0,39
distance DL [mm]	0,242		0,282	
α [degrees]	0,34	0,31	0,02	0,16
K [N/degrees]	10671,74	11704,49		

Table 7 Front twist stiffness calculation

For this reason, identifying torsional stiffness values of a motorcycle frame is a tedious process that must be done with maniacal accuracy and precision and the end value is a combination of several data, as table 8 shows.

EXPERIMENT OVERALL RESULTS engine IN [N/°]				
	DL 1	DL 2	DL 3	DL 4
K rear twist	3447,40	4120,06	6256,39	2681,31
K front twist	10671,74	11704,49		
K avg	7059,57	7912,27		
K total	7485,92		4468,85	

Table 8 Frame's overall stiffness calculation

The overall torsional stiffness value calculated is an average of the four K averages which are in turn already an average between the front and rear twist stiffness ones. The author considers the calculated stiffness value to be acceptable, even though it is slightly beyond the reference's one, finding its confirmation in the rider's feedback (Cavalli, 2015). In fact, the rider highlights a very stable motorcycle under braking and entering turns, where a stiffer frame gives confidence enough to allow him to follow the wanted racing line effortlessly; however, it is also enhanced by the rider's feedback, the motorcycle's unpredictability at mid-corner, where the torsional stiffness plays a key role. As the screenshot from an onboard footage shows (see figure above), this unpredictable behaviour often had led to low-side crashes. This type of crash suggests the too much robustness of the structure, incapable to maintain contact with the surface -i.e. grip- especially when the rider picks up the throttle to start accelerating again or when the surface is not entirely smooth.



Figure 32

Cavalli's onboard footage, photo courtesy of Cavalli, 2015

6.3 FEA – NO engine installed

The FE study focused explicitly on the application of the external loads directly upon the frame sections likewise, it is important to highlight the method used. In both cases, front and rear twist, it was firstly identified the moment generated by the external load, assumed to occur at the frame's mid-plane; the external load, in the laboratory test, was applied by a twisting arm, at a distance of 0.73 meters from the pivot axis. In the front twist test, the pivoting axis was assumed to be at the mid-point of the headstock whilst in the rear test at the mid-point of the inner surfaces of the swingarm hole - red dot in *figure 33 on the right*. Either cases experiencing a calculated moment following:

$$M_t = F * d = \text{load} * \text{gravity} * \text{lever}$$

For the used values see *Appendix E*.

This value was used for the FEA cases studied with the application of a torque; however, it was still necessary identify the forces generated at the application points upon the frame. It is assumed the presence of two equal and opposing forces F_1 and F_2 acting at the same distance from the rotational axis: an average of 0.06 meters was used for the headstock bearing housing where the forces are assumed to act along the vertical surface (*figure 34*), whilst an average of 0.1275 meters from the frame's mid-plane and the rear surfaces of the swingarm shaft holes (*figure 35*). Therefore:

$$M_t = \sum F * d = (F_1 * d_1) + (F_2 * d_2)$$

$$\therefore F_1 = F_2 = F_t$$

$$\therefore d_1 = d_2 = d_t$$

$$M_t = (F_t * d_t) + (F_t * d_t) = 2F_t * d_t$$

$$F_t = \frac{M_t}{2d_t}$$

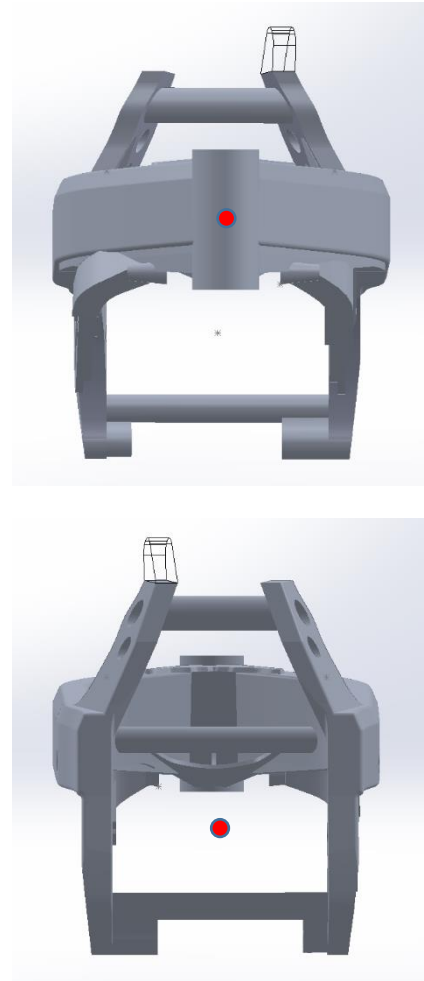


Figure 33 Front and rear pivot axis

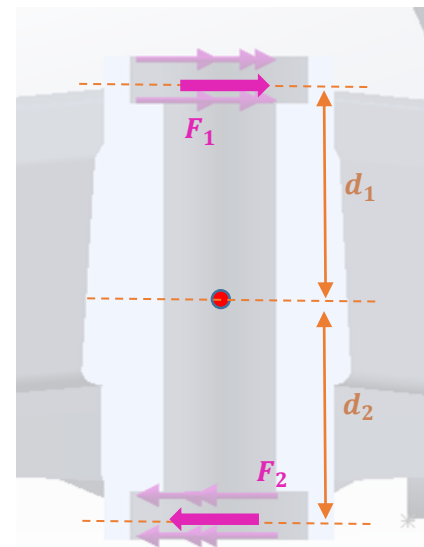


Figure 34 Headstock's section view

In the light of all parameters defined, it was the run of the FEA for both scenarios: the displacement readings have been read at the same locations where the dial gauges were placed in the experiment.

As it will be explained in the following chapters, the FE simulations for the front and rear twist were unfortunately run neither with the jig's elements nor with the engine installed, but just with the bare frame.

Notwithstanding, the analysis process is the same in both cases and the only difference stands in the values that are analysed.

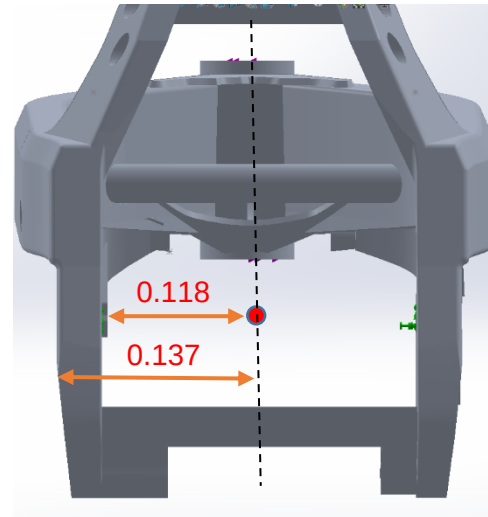


Figure 35 Rear's section view

FEA REAR TWIST NO ENGINE														
type	load	force	DL 1	ERROR %	STIFFNESS	DL 2	ERROR %	STIFFNESS	DL 3	ERROR %	STIFFNESS	DL 4	ERROR %	STIFFNESS
experiment	60,25	591,1	0,96		3760,07	-0,78		4573,1	0,56		7356,7	-1,865		2226,4
FEA FORCE		1692,0	0,0362	3%	84601,63	-0,0454		84601,6	0,568	1%	7356,7	-0,5373		7691,1
FEA TORQUE		431,5	0,035		84461,21	-0,043	5%	84461,2	0,5729		7344,5	-0,52	3%	8043,9
FEA average			0,0356	96%	84531,424	-0,0442	94%	84531,4	0,57045	-2%	7350,6	-0,52865	72%	7867,5

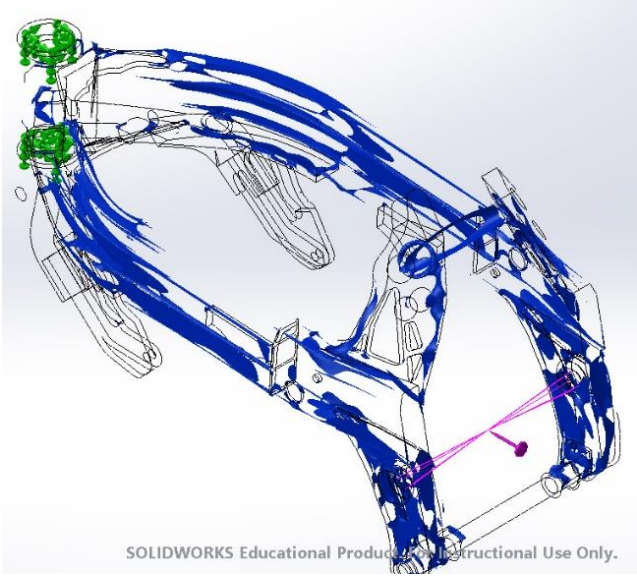
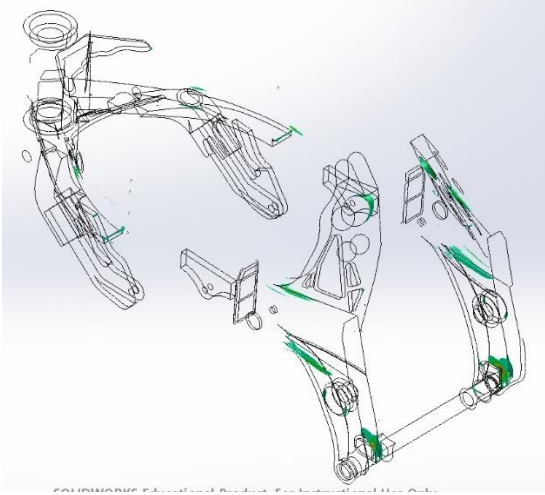
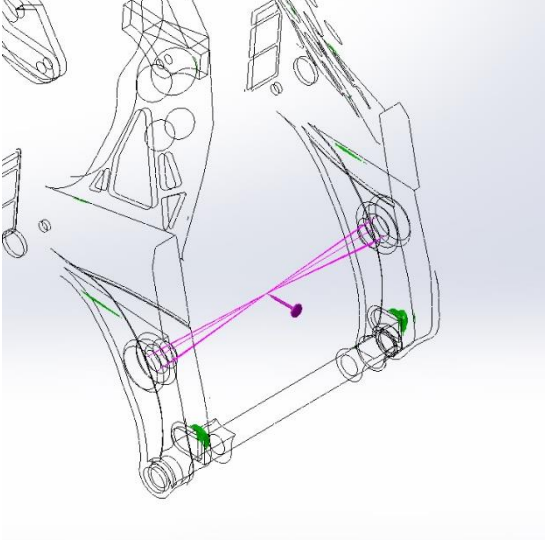
Table 9 FEA rear twist NO engine, results comparison with the physical test

FEA FRONT TWIST NO ENGINE														
type	load	force	DL 1	ERROR %	STIFFNESS	DL 2	ERROR %	STIFFNESS	DL 3	ERROR %	STIFFNESS	DL 4	ERROR %	STIFFNESS
experiment	60,20	590,6	1,06		7185,2	-0,81		9454,2	0,20			-0,530		
FEA FORCE		3592,6	0,43	16%	17962,9	-0,42	14%	17962,9	-0,006	0%		0,005	20%	
FEA TORQUE		431,1	0,36		21132,9	-0,36		21132,9	-0,006			0,004		
FEA average			0,395	63%	19547,9	-0,39	52%	19547,9	-0,006	101%		0,0045	101%	

Table 10 FEA front twist NO engine, results comparison with the physical test

The tables 9 and 10 highlights the parameters acquired and calculated from the experiment and the ones resulting the FE simulation for the rear and front twist scenario. For the FE it is showed the percentage difference between the two external loads application methods, i.e. throughout two forces in opposing direction or the twisting moment, and even though the difference is small it was used an average between them. It is immediately clear how the FE simulation underpredicts the displacement's readings of the frame's front section-i.e. DL1 and 2. Moreover, higher percentage differences are generated within the rear twist scenario than the front one, i.e. relatively average 95% and 57.5%. However, on the contrary, the rear twist scenario presents less percentages difference between DL1 and DL2, i.e. 2%, compared to the front twist one, i.e. 17%.

By contrast, the trend is inverted when analysing the rear section of the frame, i.e. DL3 and DL4. In fact, in the rear twist scenario the percentage difference between the physical test and the FE simulation is reduced by a fair amount, conversely to the front twist where it occurs the opposite. However, from the previous chapters it was acknowledged that the readings from the experiment, due to highlighted issue with the jig's robustness, e.g. the metal sheet plate

10 MPa	 SOLIDWORKS Educational Product. For Educational Use Only.	
20 MPa	 SOLIDWORKS Educational Product. For Educational Use Only.	
30 MPa	 SOLIDWORKS Educational Product. For Educational Use Only.	

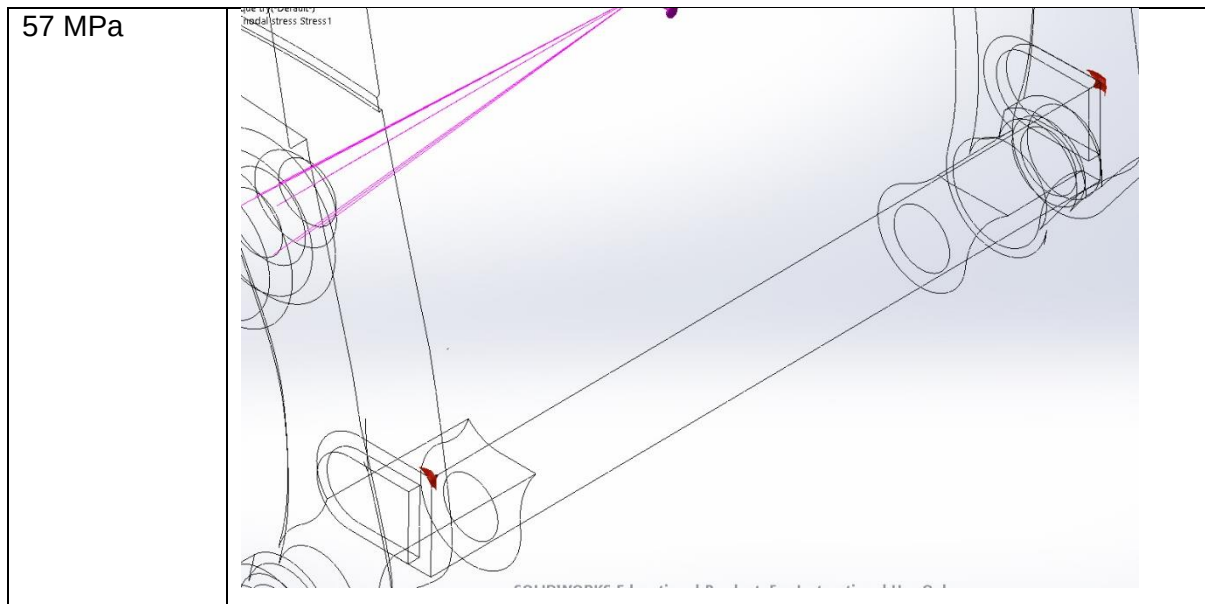
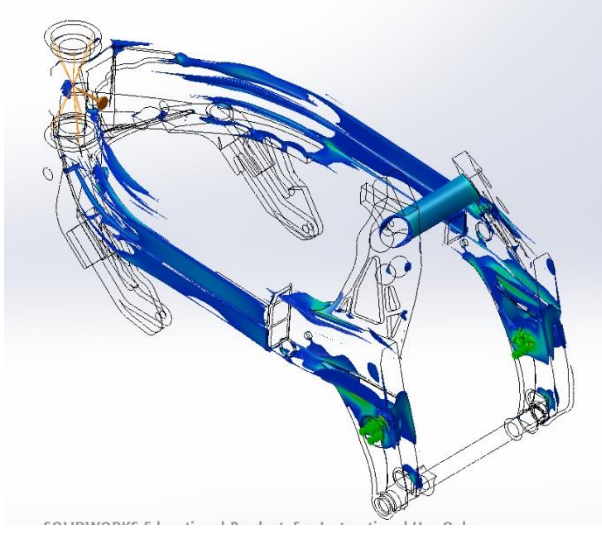
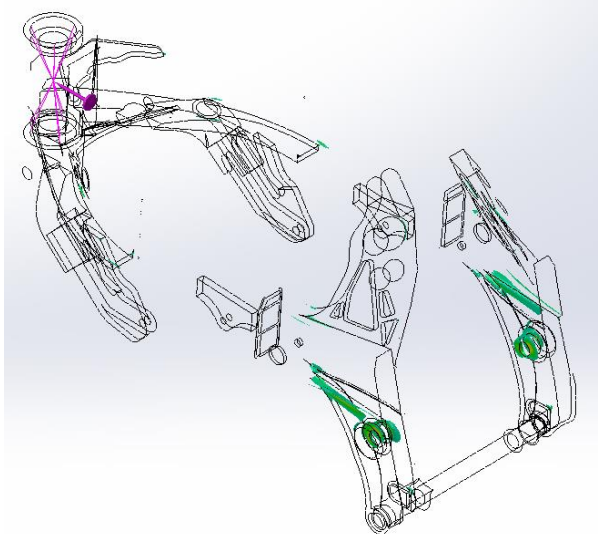
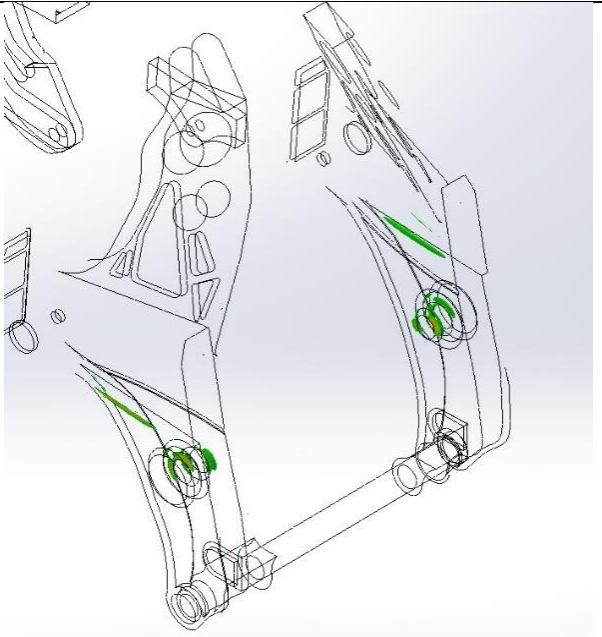


Table 11 Rear twist Stresses

Table 11 presents the stress values that the frame undergoes over the application of the external loading; the designed model results having a tensile strength of 55.15MPa. In this context it is peculiar the jump in stress concentrations between 10 and 20MPa; furthermore, stress concentrations values close and above the yield limit are displayed at the attachment points between the rear reinforcement cross bar and the casted 'arms'. Clearly, in a limit scenario, failures may occur or start from those points, however it does not come as a surprise as Foale's pragmatism tests shows: in fact, the end of attachment points of gussets or brackets or edges of a weld or any junction between a more flexible and a rigid part are typical areas where stresses concentration are (Foale, 2002).

Looking back at table 10, results seem closer to the ones gathered from the experiment. In fact, the percentage difference between the tests and the computational results are between 30-40% more precise than the counter case previously discussed. However, it was already highlighted in previous chapter, the particular case that the front twist experiment without the engine installed has represented. In another way the difference between the application of the external load throughout two coupling opposing forces or a moment is further enhanced between the two cases: in fact, for the rear twist there is a 4% average difference whilst for the front case there is a 14% average difference. In both cases, it has always been used an average between the displacements generated by the application of the forces and the torque ones. Although, the FEA still overpredicts the stiffness of the frame with a final value 105% higher than the calculated one. However, a previous section had confirmed the validity of the front twist physical test results even though considering the issue caused by the metal shim plate, so confirming the influence of assumed features within the model or their absence, as well as the material choice.

10MPa	 A 3D wireframe model of a motorcycle chassis with blue and green stress contours. The blue areas indicate higher stress concentrations, particularly around the main frame rails and the rear swingarm pivot. The green areas indicate lower stress levels. The model is shown from a perspective view, highlighting the overall structure and the distribution of stress.	
20 MPa	 A 3D wireframe model of a motorcycle chassis with green and yellow stress contours. The green areas indicate lower stress levels, while the yellow areas indicate higher stress concentrations. The model is shown from a perspective view, highlighting the overall structure and the distribution of stress.	
30MPa	 A 3D wireframe model of a motorcycle chassis with green and yellow stress contours. The green areas indicate lower stress levels, while the yellow areas indicate higher stress concentrations. The model is shown from a perspective view, highlighting the overall structure and the distribution of stress.	

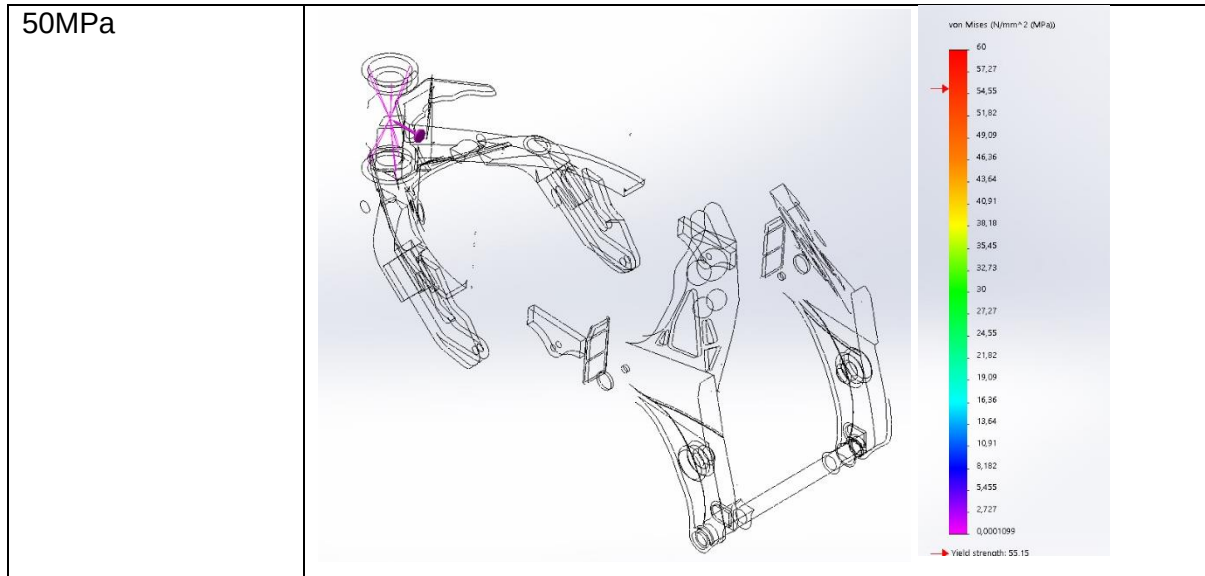


Table 12 Front twist Stresses

Table 12 above, as for the previous case, highlights the stress concentrations under the external load influence. As previously, the model keeps its yield strength value to 55.15 MPa and this time there are no areas where this value is reached or exceeded- explained further on in this chapter-, however the interested areas are in different sections of the frame, e.g. front and rear sections of the frame.

Finally, it was identified the maximum strain value that should not be exceeded, following Young's module definition:

$$E = \frac{\sigma_{max}}{\varepsilon} = \frac{stress}{strain} \rightarrow \varepsilon = \frac{\sigma_{max}}{E}$$

Throughout the Young's modulus values of the aluminum alloy used, i.e. $E = 6,90E + 10 \text{ N/m}^2$, and from FEA's maximum stress figure, i.e. $\sigma_{max} \equiv yield\ strength = 5,52E + 07 \text{ N/m}^2$, it was calculated the maximum strain value:

$$\therefore \varepsilon = \frac{5,52E + 07}{6,90E + 10} = 7,99E - 04$$

This value was exploited to redefine the used scale within the FEA simulation as upper limit for either front and rear twist simulations.

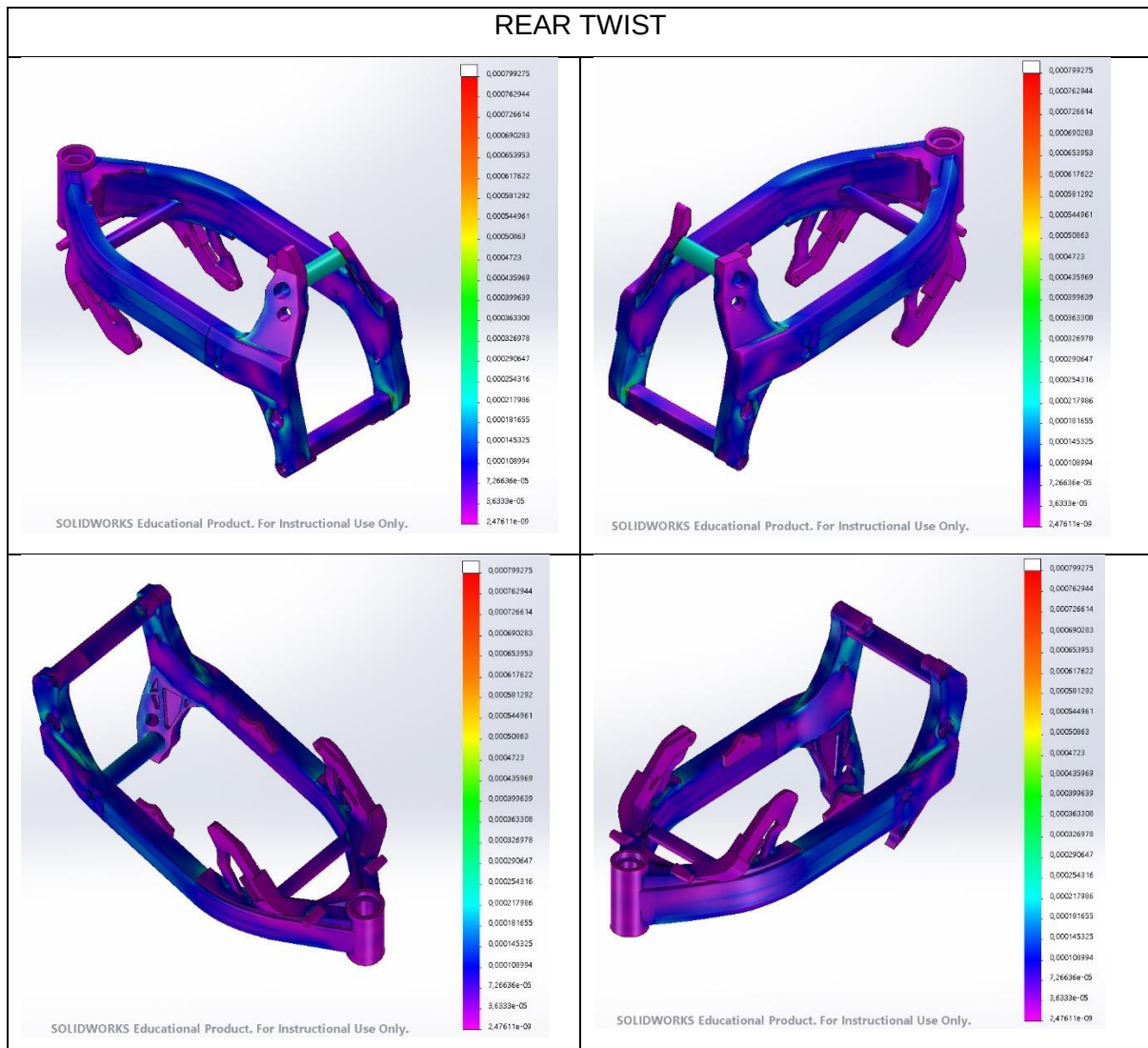
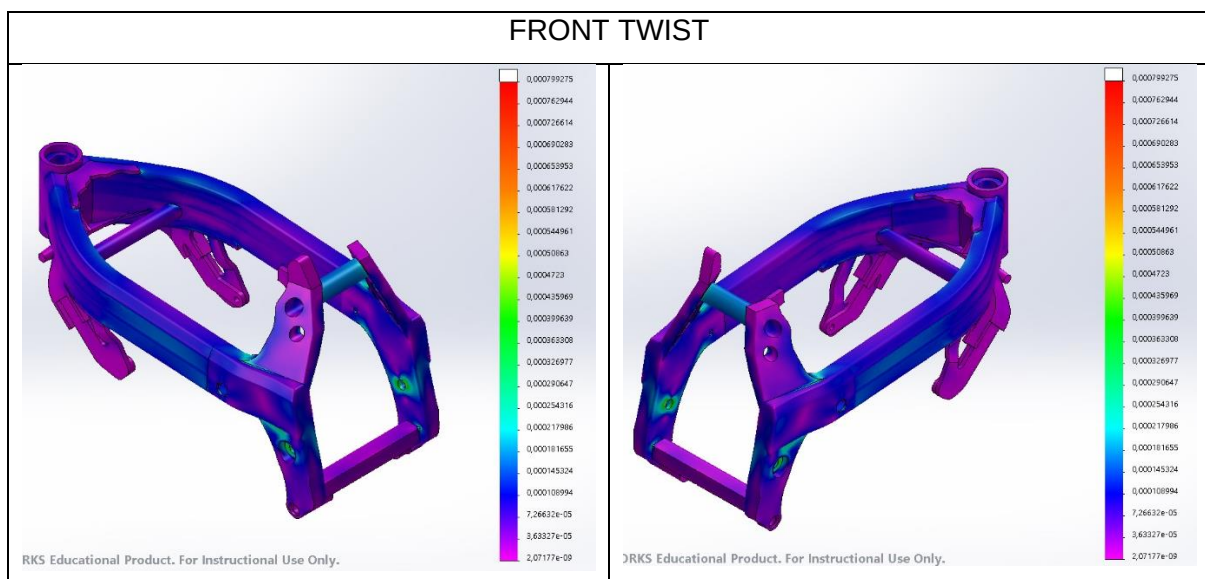


Table 13 Rear twist strain display



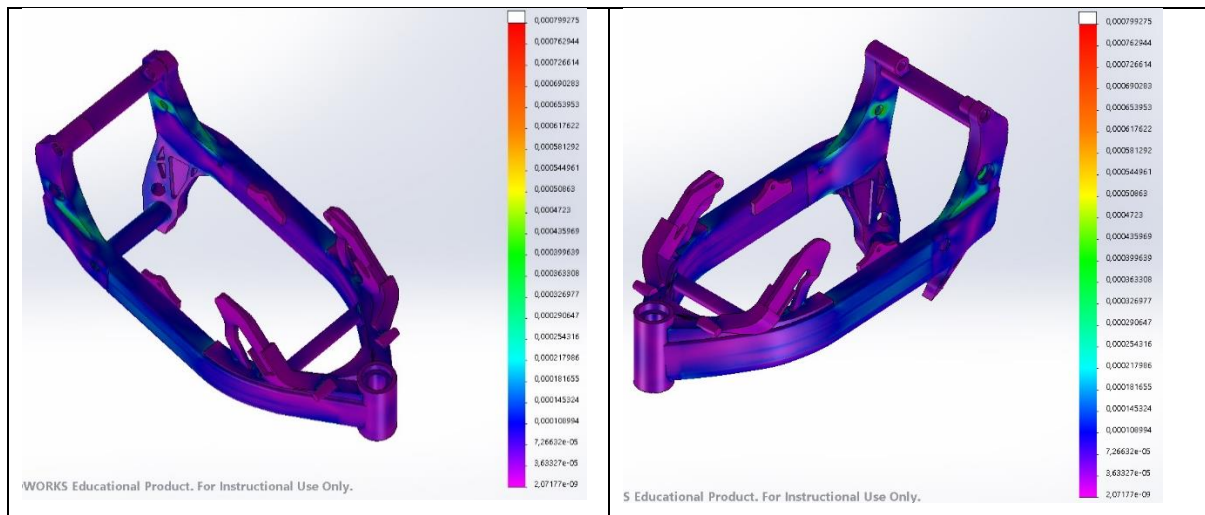


Table 14 Front twist strain display

It must be highlighted that further checks must be conducted upon internal sections as well.

7 Further Developments

The report has highlighted numerous processes for the development of a frame structural analysis; however, many uncovered features can still be part of a similar development, improving the quality of results:

- The rear twist experiment should include the application of extra dial gauges, reading the displacements of the rear clamping arms and of the main twisting arm. This would allow a first analysis of the twisting assembly robustness by reading and comparing the vertical displacements of the components.
- Both experiments and simulations should be repeated on each rotational side; in fact, it was only applied a clockwise moment.
- Further experiment should be conducted to analyse the lateral and longitudinal deflection.
- A further improvement would be the application of the twisting loads as it occurs according to vehicle's geometries. This means applying the loads at the entire structure as it occurs during normal loading conditions, achieving the whole motorcycle stiffness.

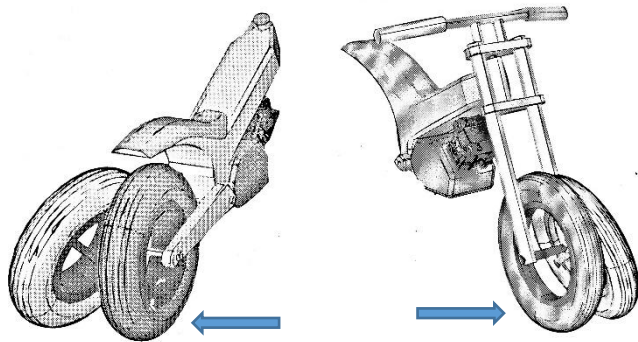


Figure 37 Loadings

- A new jig design would be necessary: attachment points to the surface should be upon the engine case, allowing the twisting action to act directly onto the frame.
- The influence of different torque values of the engine bolts as well as the influence of the added rigidity given by the fuel tank, air box or of the rear subframe when installed upon the main chassis, could also be studied.
- A vibrational test should be done to identify the resonance frequencies.
- An ultrasonic test should be conducted to refine the inner geometries, shapes and surfaces' thickness values which were assumed during this project.
- Given the observed trends of factors that influences the rigidity of the structure, further development of the influence of engine mounts should be carried out, already proved by the author's attempt of unscrewing one of the attachment points.

8 Year 2020 - Pandemic Impact

Unfortunately, not even an over scheduled time plan could cope with the outbreak of the 2020's pandemic. The interruption of the face-to-face lectures and the subsequent closure of the university's site had generated an unpredict impact over the scheduled plan. The following chapter is intended to highlight the issues that had been raised and their impact.

- The FEA was initially intended to be conducted using Design Studio software, installed upon university's computers. The software was preferred upon others due to its feature of choosing different materials within a single piece structure, therefore identifying and distinguishing for instance, casted or machined sections in a straightforward way. Consequently, the computational power required to analyse a single body, instead of multiples as an assembly, would have been less. *Figures 38 and 39* shows an example of the first simulations run with the university's licensed computers.

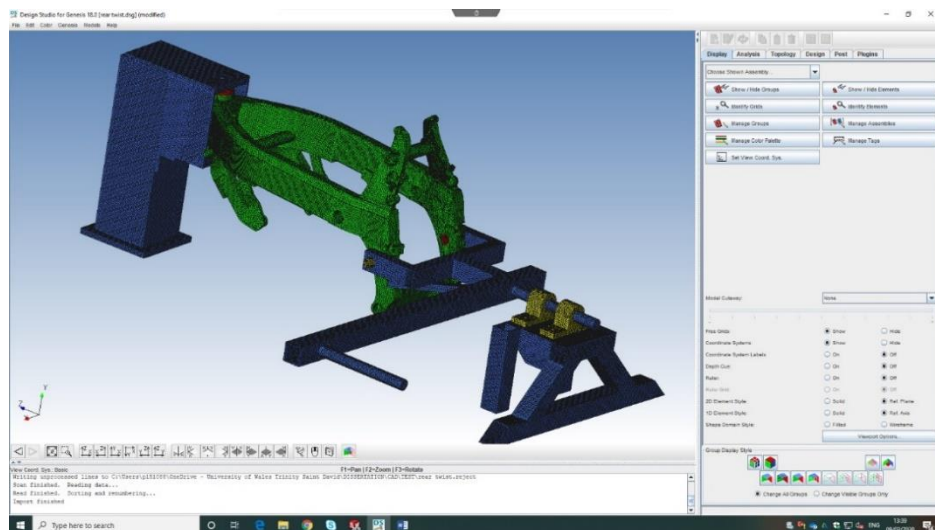


Figure 38 Rear twist with Design Studio

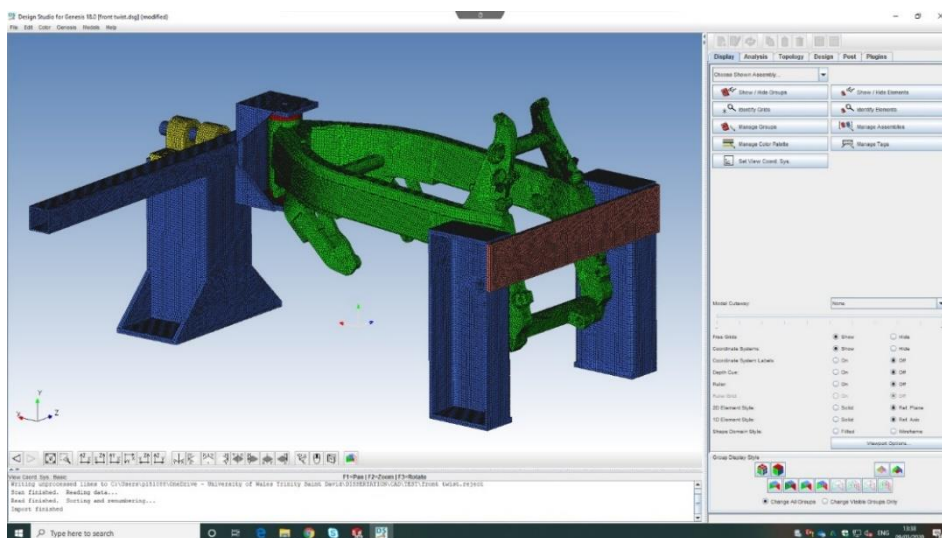


Figure 39 Front twist with Design Studio

- The main issue was represented by the lack of computational power of the author's personal device which:
 - I Made necessary the simplification of the geometries, shapes and features of the chassis.

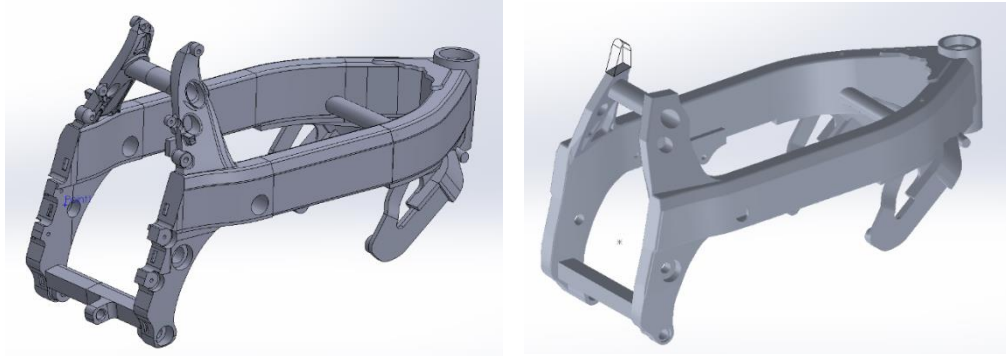


Figure 40 Ideal representation and simplified version

- II did not allow to run simulations including the jig that would have allowed the validation of either the experiment and the FEA, furthermore analysing its robustness and therefore optimize it where necessary.

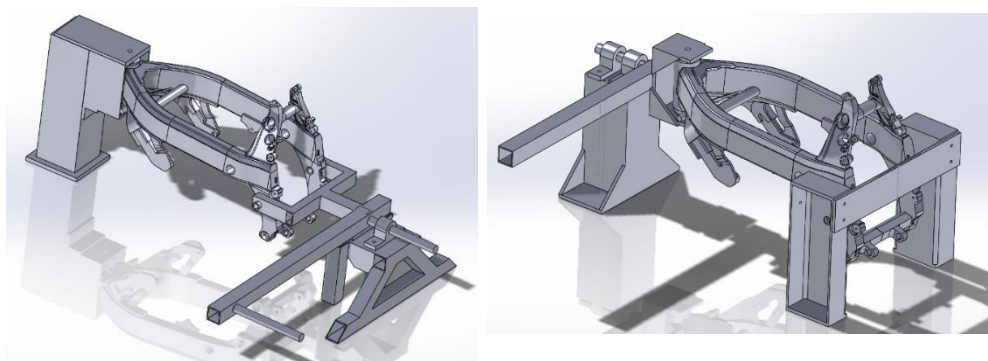


Figure 41 FEA comprehensive of the Jig

- III Did not allow to include into the simulation the extra-robustness of the engine within the frame.
- IV Did not allow to perform *mesh optimization* upon specific areas of the frame, for instance further reducing the element's dimension.
- V made difficult to perform a *mesh analysis*. In fact, fundamental for the accuracy of an FEA's result is the *mesh quality*: since it is dealt with a 3D model, solid tetrahedral elements were generated. However, it was not possible control their *aspect ratio*, their *skewness* nor their *Jacobian* relations. The first one is a ratio



Figure 42 Solid Tetrahedral mesh elements

between the longest and the shortest edges of the elements, identifying how close the elements are to a perfect tetrahedral shape. Intuitively, by definition, ratios close to 1 identifies a perfect shape on the contrary values closer to 0 represents a bad mesh quality. Moreover, the skewness is defined as the minimum angle between the vector connecting each node to its opposite mid-side and the vector connecting the mid-sides points of the sides adjacent the node. It is therefore a value that identifies how much the element is deformed, ideal value is 0° . Finally, the Jacobian value, that ranges between $-1 < x < 1$ where 1 is the ideal value, represents how perfect the element is related to its shape.

It must be acknowledged that author's "lack of computational power" identifies the incapability of its device to complete the command in acceptable time frames or either accomplishing it entirely without crashing.

- The closure of the university's facilities did not allow the repetition of the torsional tests with the application of strain gauges upon the frame.
- The time plan, *Appendix A*, shows that the development of the project was on schedule. In fact, thanks to the involvement of fellow students interested in reviving the motorcycle by the end of first semester, the vehicle was running again, allowing the project to have the dynamic test as the time plan shows.



Figure 43 Vinicius Dreher, fellow student, starting the motorcycle after weeks of commitment- 18th December

Furthermore, it was already scheduled and booked, on the 16th March, a first track-test to check the correct operation of the motorcycle. It would have followed, after the

application of the strain gauges upon the frame, a second test to gather data from the stresses and deflection of the chassis while on track.

- From the acquired data on track it would have then been possible analyse and calculate the actual forces acting upon the chassis, during the different transients, i.e. breaking, cornering, accelerating, and then replicate the experiment and the simulation with realistic force/torque values involved.

In fact, frame's stiffness is engineered to match the forces generated and expected to be experienced (Bradley, 2003).

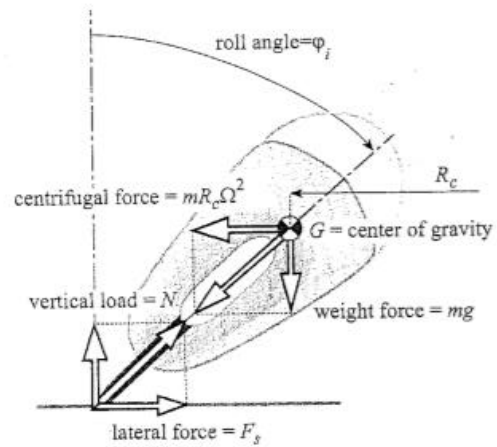


Figure 44 Forces during cornering (Cossalter, 2006)

9 Conclusion

In conclusion, this dissertation grounds a methodology to conduct a structural analysis of a motorcycle's chassis or similarly for the development of a new or optimized proposal.

It is understood that the identification of a stiffness' compromise has always been a main concern for chassis designers and that strength does not affect the handling behavior of the vehicle but identifies the loading capabilities of it. On the other hand, stiffness affects how the motorcycle will perform under the various conditions that are faced in the different scenarios.

To conduct a similar research/development, it is understood the importance that a physical test has over the validation of another key step, the numerical analysis (FEA) or vice versa.

It is in fact highlighted the importance of mesh quality for the accuracy of the results as well the methodology used for the physical tests.

The literature reviewed throughout the process has showed that similar analysis often lacks either the physical tests or the FE simulations, proving that there is not a single correct value to identify a frame's stiffness.

Furthermore, several parameters affect the performance of a motorcycle chassis therefore, the overall stiffness is a result of multiple sources that must be taken into account, e.g. engine mounting points, subframe attachments, dynamic geometries etc.

EXPERIMENT OVERALL RESULTS engine IN [N/°]					FEA OVERALL RESULTS engine IN [N/°]				
	DL 1	DL 2	DL 3	DL 4		DL 1	DL 2	DL 3	DL 4
K rear twist	3447,40	4120,06	6256,39	2681,31	K rear twist				
K front twist	10671,74	11704,49			K front twist				
K avg	7059,57	7912,27			K avg				
K total	7485,92		4468,85		K total				

EXPERIMENT OVERALL RESULTS w/o engine [N/°]					FEA OVERALL RESULTS w/o engine [N/°]				
	DL 1	DL 2	DL 3	DL 4		DL 1	DL 2	DL 3	DL 4
K rear twist	3760,07	4573,06	7356,66	2226,36	K rear twist	84531,42	84531,42	7350,56	7867,49
K front twist	7185,17	9454,17			K front twist	19547,89	19547,89		
K avg	5472,62	7013,62			K avg	52039,66	52039,66		
K total	6243,12		4791,51		K total	52039,66		7609,02	

Table 15 Stiffness Results

Moreover, it is proved the importance of data correctness especially for setting the simulations. In fact, the results accuracy is proportional to it, i.e. "GIGO, garbage in, garbage out".

Table 15, in fact, highlights this concept by showing how the stiffness values achieved throughout the simulations are overpredicting; in fact, it was demonstrated throughout the report the influence of certain assumptions over others.

Additionally, *table 15* shows some missing data due the pandemic developed during the first part of year 2020, however proving that wherever there is a risk, there are also new capabilities and opportunities (Maylor, 1996) verifiable in the rearrangement of the time plan and tasks. In fact, the author was able to fulfill his interest over material structural analysis anyhow.

To conclude, the dissertation was not intended as an expression of author's ability to 3D model nor run FEA simulation nor manufacturing skills; however, it shows the paramount steps, analysis and theoretical foundations that are core for the development of the subject. Thus, the relevance of the above script relies in the belief that a similar process, if conducted in the industry, would count a team of engineers where maximum result's accuracy is achieved throughout single specialized expertise.

10 References

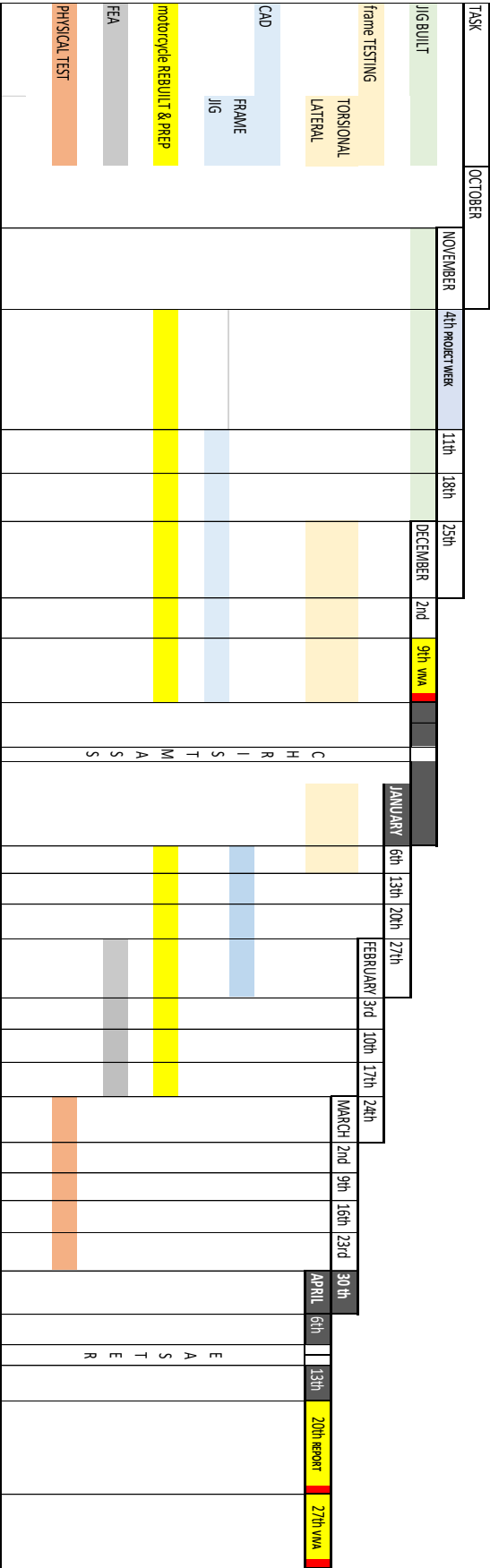
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12 Appendix A



13 Appendix B

- 1/10 searching for Paul to plan the project, he is not in
 10/10 Paul is stuck on the highway
 11/10 Paul is not in due to the rough day, yesterday
 15/10 meeting with Paul, searching the frames available
 18/10 meeting with OWEN WILLIAM to discuss the project-
 STATIC ANALYSIS of CHASSIS TORSIONAL, LATERAL & LONGITUDINAL STIFFNESS, built of a jig for so, dynamic
 analysis on track with straw caucers
 22/10 MEETING WITH JOHN KINSELA to understand the
 availability of straw caucers & acquisition module
 22/10 MEETING WITH ANDREW HARRISON TO EXPLAIN THE
 PROJECT, UNDERSTANDING THE MOTORCYCLE'S (FRAME)
 AVAILABILITY (HONDA V4, APRILIA RS 250, YAMAHA R6,
 DUCATI PANIGALE)
 DISCUSSING PROBLEM OF A RUNNING BIKE,
 19/10 MEETING WITH OWEN TO DISCUSS EMAIL TO
 FORWARDED TO COMPANIES → (ONTO THE OTHER PAGE)
 22/10 MEETING WITH OWEN TO DISCUSS A SOLUTION
 PROPOSED BY A COMPANY (CHASSIS FACTORY)
 THAT HAD REPLIED TO MY REQUEST
 26/10 RESEARCH FOR PROJECT
 28/10 RESEARCH FOR PROJECT

- ⑧ MEETING WITH ANDREW TUBBOT & DAVE COOPER
22/10 TO UNDERSTAND THE POSSIBILITY OF ~~THE~~ SCANNING THE FRAME. BEEN TOLD TO COME BACK THURSDAY FOR AN ANSWER FROM THE PERSON WHO CAN DO IT.
- ⑧ 24/10 ANDREW TUBBOT & DAVE COOPER NOT IN OFFICE
- 28/10 ~~NO ANSWERS FROM COMPANY RE: SCANNING~~
~~CONTACTED~~ PHONE CALL WITH AUTHOR JOHN BRADLEY OF ~~THE~~ ^{THE} RACING MOTORCYCLE VOL 1-2-3 TO DISCUSS THE PROJECT & HAVE INFO ABOUT IT.
- 29/10 DUE TO NO ANSWER FROM COMPANIES, IT WILL BE USED THE HONDA V4 CHASSIS.
- MEETING WITH PAUL: "THERE ARE NO BLADES, I'VE ORDERED THEM 6 WEEKS AGO WE CAN START BUILDING BUT IT WILL BE A SLOW PROCESS DUE TO LACK OF INSTRUMENTS. IF THERE ARE THINGS YOU CAN DO IN THE MEAN TIME, DO THEM AND LET'S HOPE THE BLADES ARRIVES."
 - MEETING WITH ANDREW TUBBOT THAT HAD CONFIRMED THE POSSIBILITY TO SEND THE CHASSIS* TO SCAN IT: "JUST ADVISE ME WHEN"

- 29/10 DUE TO ~~NO~~ LACK OF MATERIAL/TOOLS
DECIDED TO SEND THE CHASSIS TO SCAN
SO TO START THE CAD OF IT.
- MEETING WITH ANDREW HARRISON IN THE
WORKSHOP ~~TO~~ WITH STUDENTS FROM 1ST
YEAR THAT WILL REPAIR THE NON-RUNNING
HONDA V4 THAT WILL BE USED TO
PERFORM IN SECOND SEMESTER, DYNAMIC
TESTS ON TRACK.
- MEETING WITH ANDREW TIBBOT TO CONFIRM THE
SCAN - HE SAID HE WILL BE EMAILING THE PERSON,
~~THE~~ THE SCAN WILL TAKE ONLY 1 1/2 HOUR. HE WILL
LET ME KNOW WHEN.
- 30/10 ORGANISING BUILDING OF THE JIG WITH PAUL.
SKETCHES OF SOLUTIONS
- 4/11 } MACHINING OF SWINGARM PIVOT SPACERS
5/11 } ~~HEADSTOCK SPACERS~~
6/11 } PROJECT WEEK HEADSTOCK SPACERS
7/11 } + 1 HEADSTOCK SPACER
8/11 } FRONT CLAMP ASSEMBLY ALIGN-
- 11/11 ^{Monday} FRONT CLAMP FIXING
- 15/11 ^{Monday} SIDE ^{HOLDING} PILLARS + TOP & BOTTOM CAPS
CUTTING OF NEW FRONT & LATERAL PIVOT

18/11	SAWING FRONT & LATERAL PIVOT
19/11	SIDE PIVOT HOLES FOR CLAMPING NUTS
20/11	3D SCAN + <u>WELDING</u> OF FRONT CLAMP WELDING FRONT POST
27/11	RE-SIZING LATERAL CAMBLES OF FRONT CLAMP DUE TO DISTORTION FROM WELDING ASSEMBLING <u>ENGINE</u> FIXING ENGINE IN FRAME
28/11	FRONT CLAMP FOR TORSIONAL TEST (FROM WORKSHOP CLOSED)
2/12	TAKING MEASUREMENTS FOR CAD WELDING OF FRONT CLAMP ASKED ANDY TIBOT FOR INFO ON 3D SCAN POINT CLOUD ASKED ANDY TIBOT FOR NEW SPACER
3/12	MEETING WITH KEVIN
4/12	REAR CLAMPING POINTS ASK TIBOT FOR SPACER FOR REAR SHAFT
6/12	SPACERS, REAR SHAFT THREADED HOLES ASSEMBLE SIG FOR PRESENTATION, UPDATE KIT
7/12	3D SCAN (OBJ FORMAT)
9/12	MACHINE SHOP CLOSED - ASKING HOW TO USE 3D SCAN
10/12	" "

- ~~12/12~~ ~~12/12~~ JIG ACCOMPLISH
- 9/12 STARTING POWER POINT FOR VIVA
- 12/12 VIVA
- 13/12 FIRST TEST WITH THE JIG, NECESSARY A NEW SPACER FOR REAR HOLDING ROD WOBBLING WITHIN THE THREADS.
- 16/12 ASK A. TROTT TECHNICIAN NEW SPACER NOT POSSIBLE PRIOR CHRISTMAS - AVAILABLE FROM THE 6TH JANUARY
- 10/1 SPACER NOT READY
- 31/1 SPACER READY
JIG 100% DONE
- 3/2 REAR TWIST
- 4/2 FRONT TWIST
- ~~12/2 MEETING WITH A. HARRISON~~
- ~~12/2 MEETING WITH A. HARRISON~~
- 11/2 ~~STRONGER~~ FINISHING CAD MODEL
- 12/2 OPTIMISING REAR PART
- 18/2 TRYING TO OPTIMISE FRONT PART (AROUND HEADSTOCK)
- 20/2 MEETING WT A. HARRISON to show 3D MODEL
- 21/2 MEETING WITH KELVIN - IDENTIFIED MESH TOOL in SOLIDWORKS BUT MODEL NEED REFINEMENTS - REFINEMENT OF THE MODEL
- 22/2 " " " " difficulty to replicate lower part of the headstock.

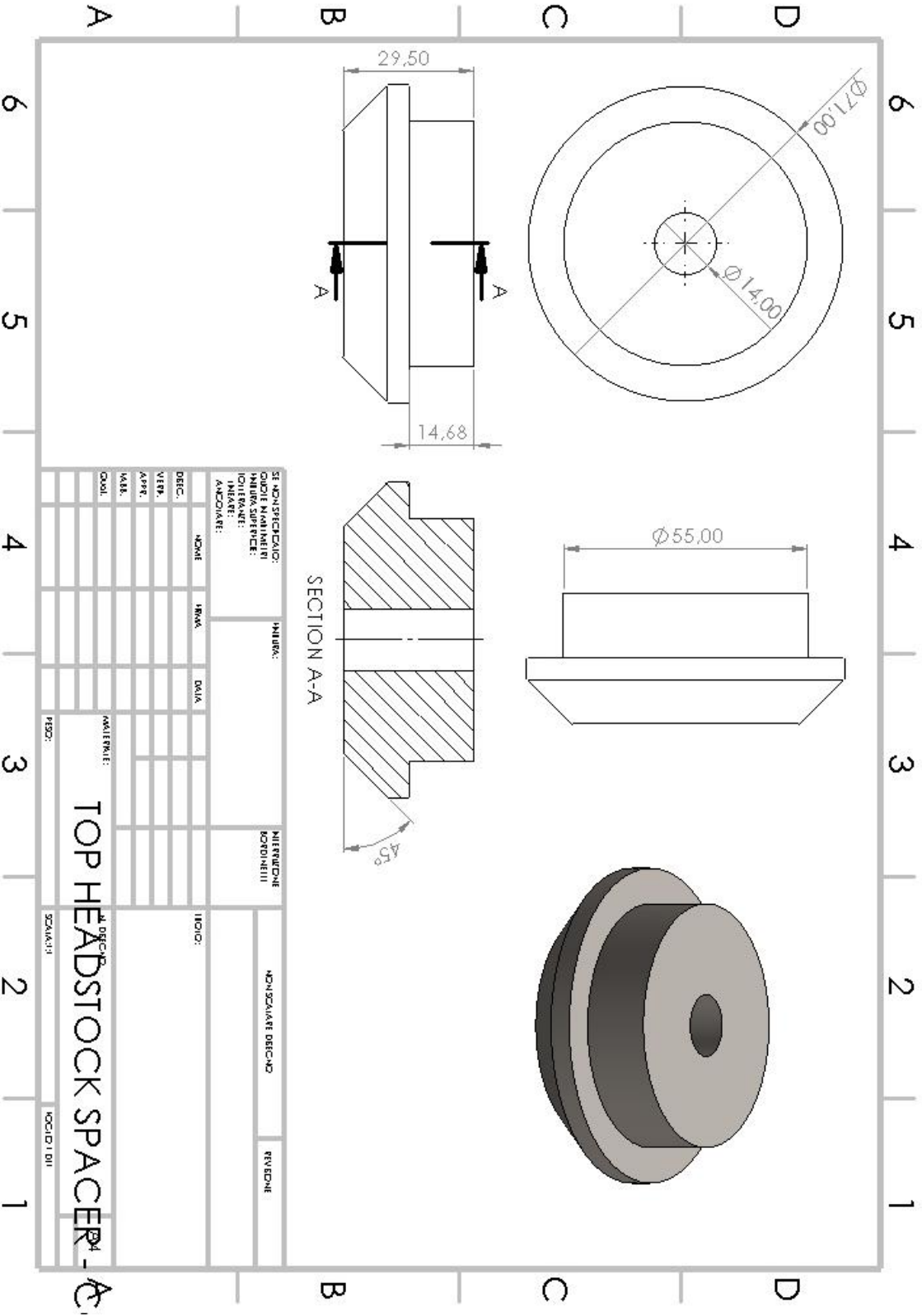
24/2	meeting with A. TIBBOTT & D. COOPER to ask for help suggested a new software to deal with surfaces (FUSION)
25/2	optimising CAD model
26/2	" " "
27/2	meeting with A. HARRISON for help with 3D model problems NOT Able to use surfaces Need to find a ^{ALTERNATIVE} way to accomplish that section - - MESH PROBLEM WITH REAR FRAME PART -
03/03	meeting with KELVIN LAKE organising action plan create mesh of all components
06/03	meeting with KELVIN LAKE first approach to DESIGN STUDIO
7/03	organizing all meshes and trying to generate a single mesh with all components SIG + Remes
11/03	meeting with Kelvin ~~tried~~ to merge surfaces Better if import the model with all surfaces already ~~connected~~ connected
12/03	NEW MODEL ALL CONNECTED MEETING WITH KELVIN LAKE for FEM simulation SIMULATION GONE WRONG ~~surfaces~~ don't remain attached BETTER TO CREATE FROM ZERO the model
13/03	MEETING WITH ANDREW HARRISON: HIGHLIGHT THE EFFICIENT PROCESS instead of exploiting ~~problems~~ ^{MISTAKES in the APPROACH}

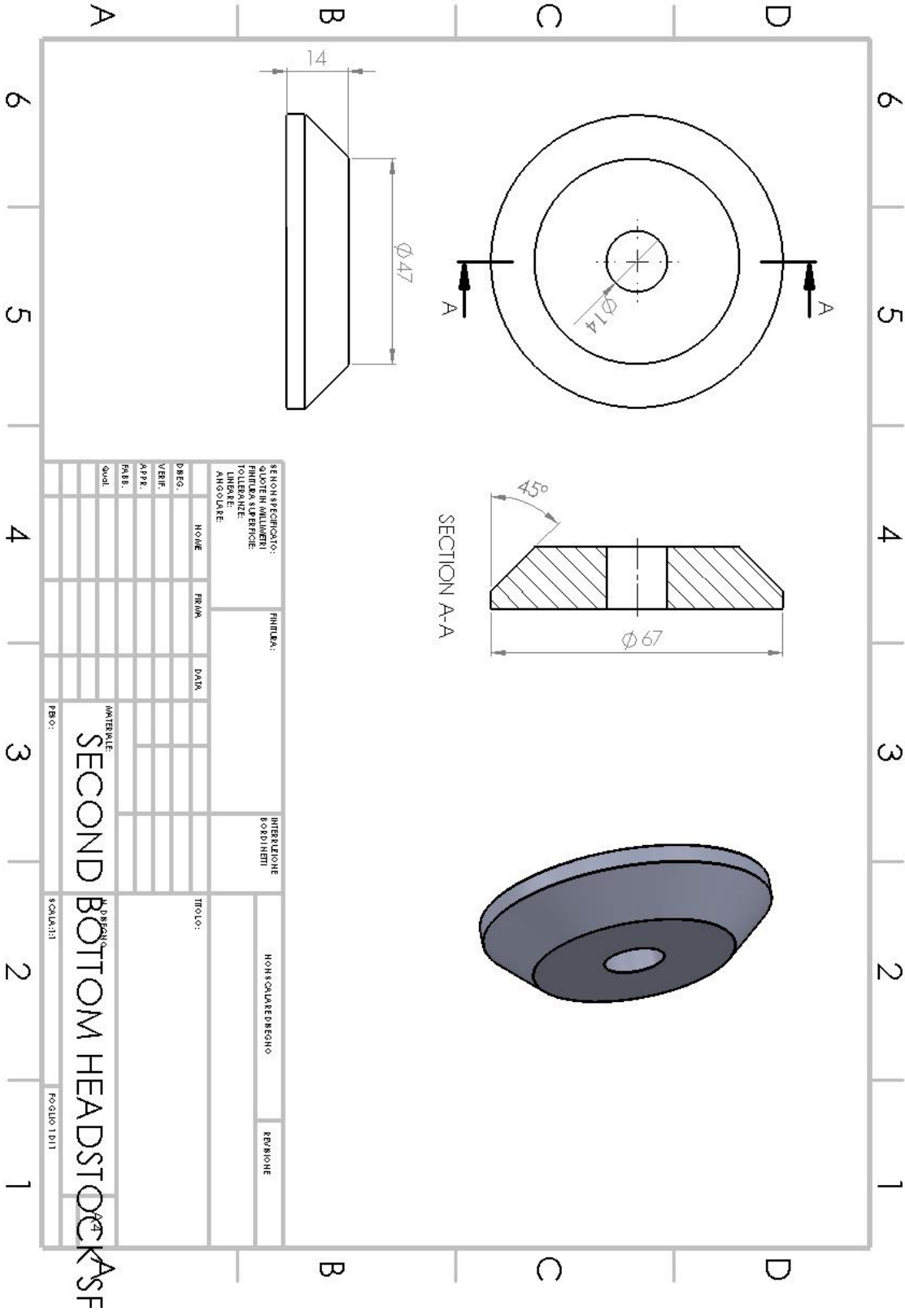
14/3	~~REBUILT~~ REBUILT ENTIRE REAR FRAME PART - STILL HAVING PROBLEMS WITH MESH - TEXTED HARRISON A. to ask for help
15/3	MODEL MESHED ~~REBUILT~~ HARRISON. A. REPLIED ME EARLY IN THE MORNING WITH A MESHED AND SIMULATION RUN
16/3	~~REBUILT~~ MEETING WITH HARRISON A. TO UNDERSTAND ~~REBUILT~~ HOW TO ACHIEVE A MESH IN SOLIDWORKS
16/3	[STOP FACE TO FACE LESSONS]
22/3	VIDEO CALL WITH JANUARIO A. TO ASK TIPS FOR SIMULATIONS ONTO SOLIDWORKS.
23/3	UPDATING LAKE K Lecturer ABOUT THE IMPOSSIBILITY TO USE DESIGN STUDIO SOFTWARE DUE TO UNIVERSITY LOCK DOWN
↑ + 2 WEEKS EXTENSION REPORT DEADLINE 21/5 ↓ 30/3 ~~REBUILT~~ FIXED REAR GEOMETRY AND FEATURES OF THE FRAME - ADDED THE SHELL FEATURE FOR THE ONLY REAR SECTION.	
31/3	RUN A ~~REBUILT~~ THIRD SIMULATION TO UNDERSTAND THE PROCESS - ORGANIZED ASSEMBLY TO EXPLOIT THE CORRECT COORDINATE SYSTEM
2/4	MEETING ON ~~REBUILT~~ SKYPE WITH COURSE MATE JANUARIO A. TO ACQUIRE DEEPER KNOWLEDGE OVER SIMULATIONS
	MEETING WITH HARRISON A. TO UPDATE THE ~~REBUILT~~ DEVELOPMENT PLANNING OF SIMULATIONS
3/4	EMAILED TIBBOT. A. TO ASK FOR WEIGHT MEASUREMENTS OF THE V4 ENGINE

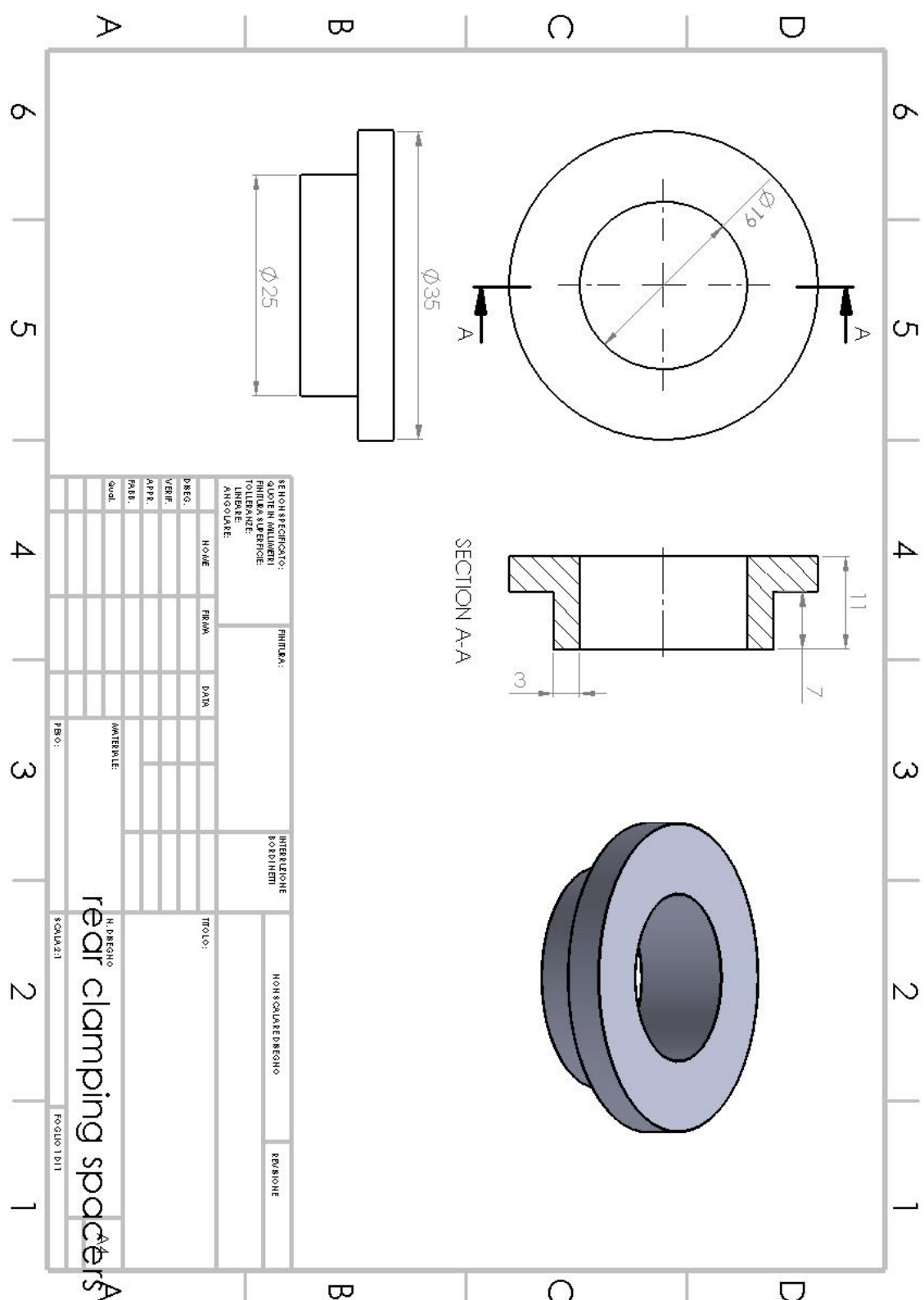
20/4 UPDATING LECTURER HARRISON.A
UPON PROBLEMS WITH COMPUTATIONAL
POWER AND OVERALL PROGRESS
OF THE DISSERTATION

22/4 MEETING UPON MICROSOFT TEAM
WITH LECTURER LAKE K.
TO DISCUSS THE CORRECTNESS OF
FEA RESULTS
QUERIES UPON STRAIN ENERGY
GRAPHS

14 Appendix C

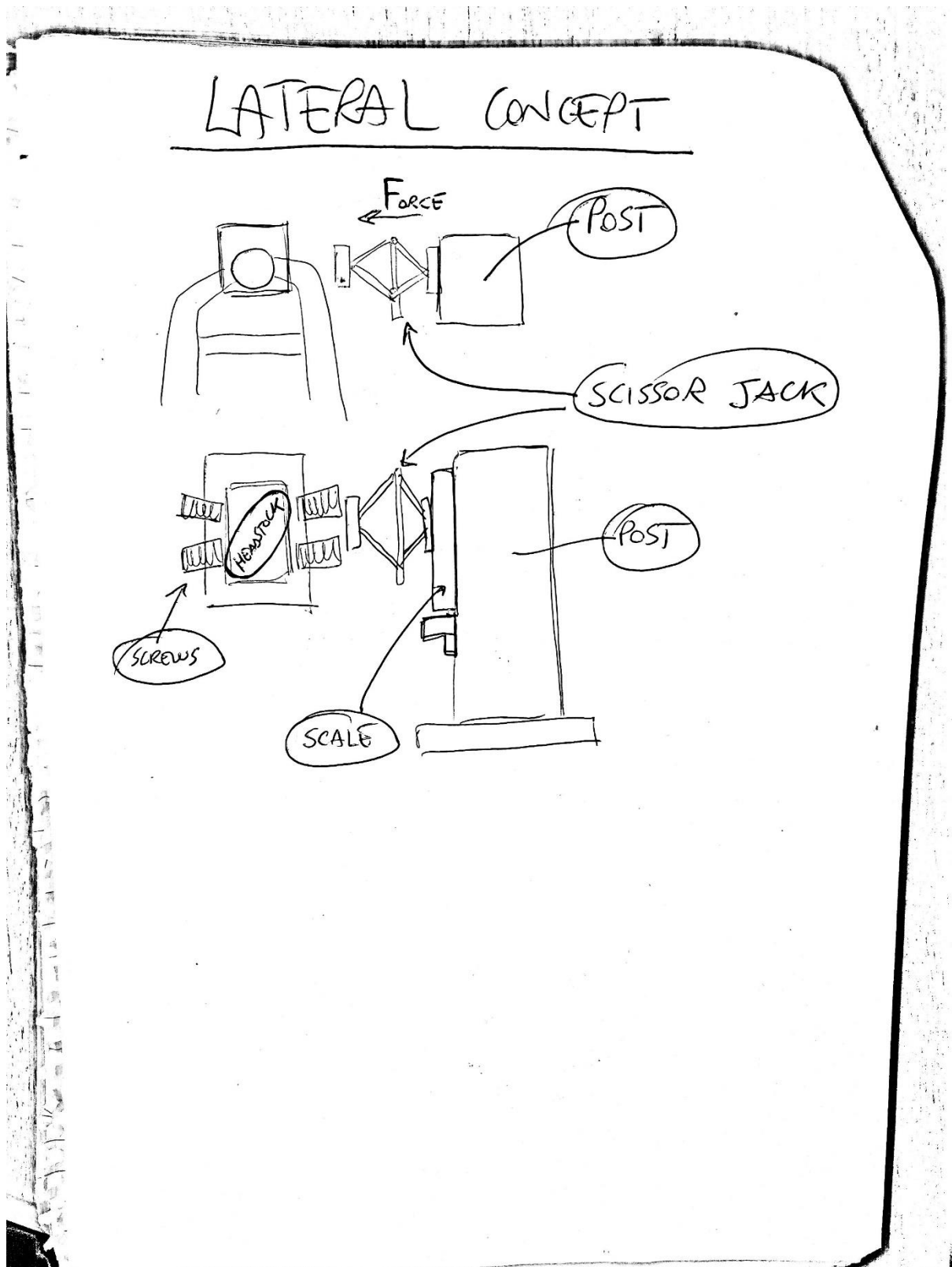




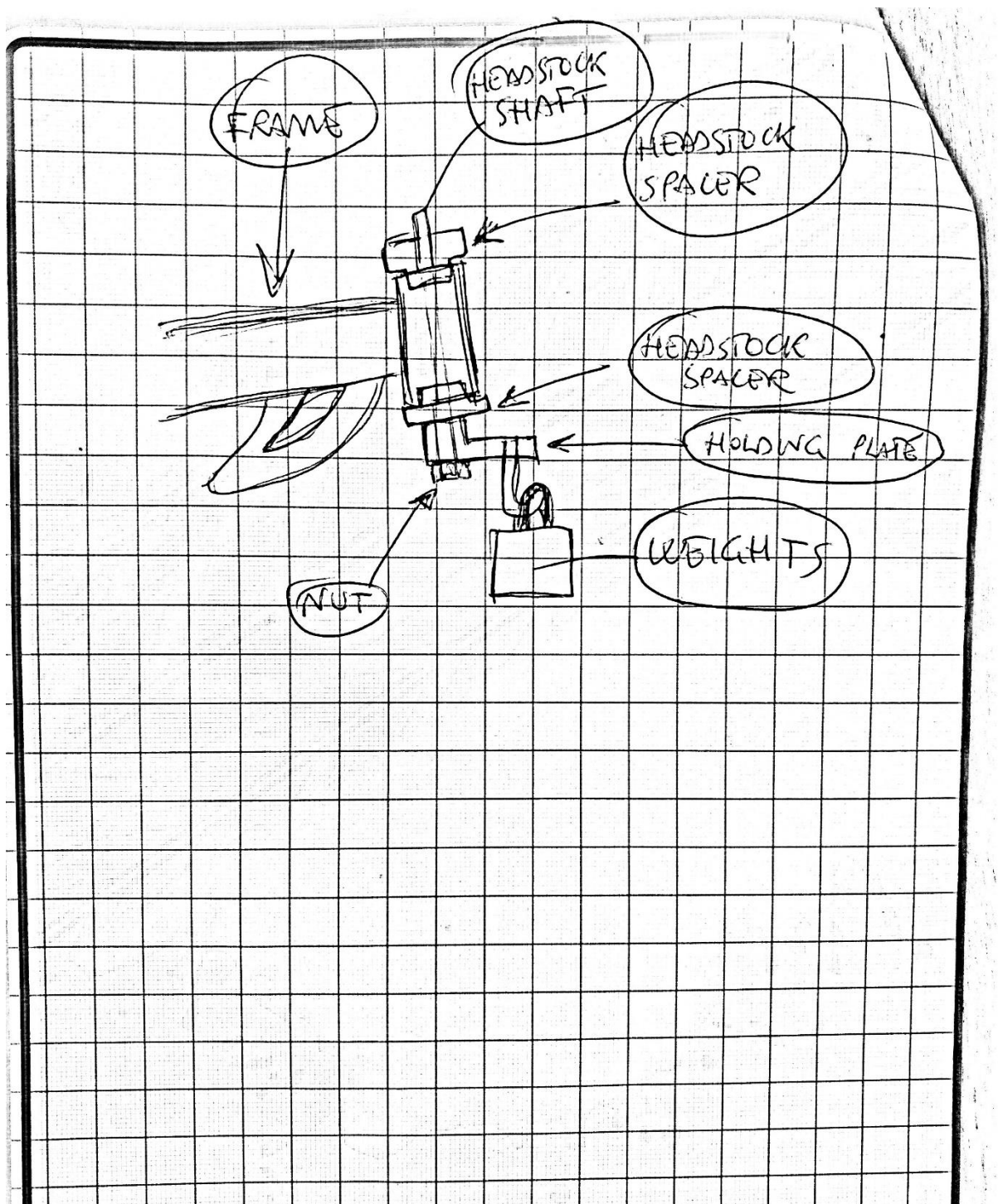


15 Appendix D

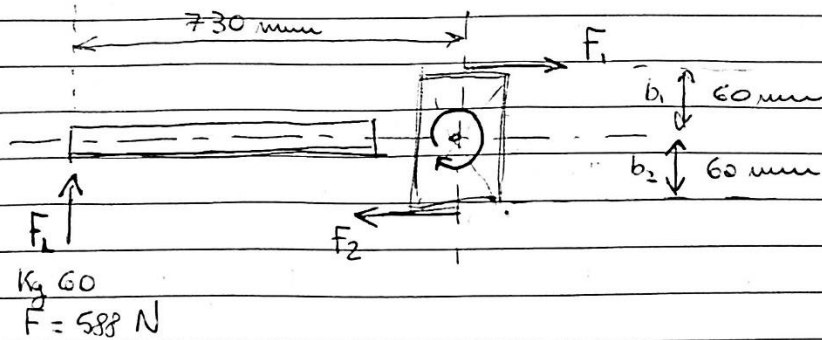
Lateral experiment concept:



Longitudinal experiment concept:



16 Appendix E

FRONT TWIST

$$F_1 = F_2 = F$$

$$T_{\text{single}} = F \cdot b$$

$$588 \cdot 0,73 = 429\ Nm$$

at the center
plane

$$T_{\text{Total}} = T_1 + T_2$$

$$\curvearrow \quad \curvearrow$$

$$F_1 \cdot 0,06 + F_2 \cdot 0,06$$

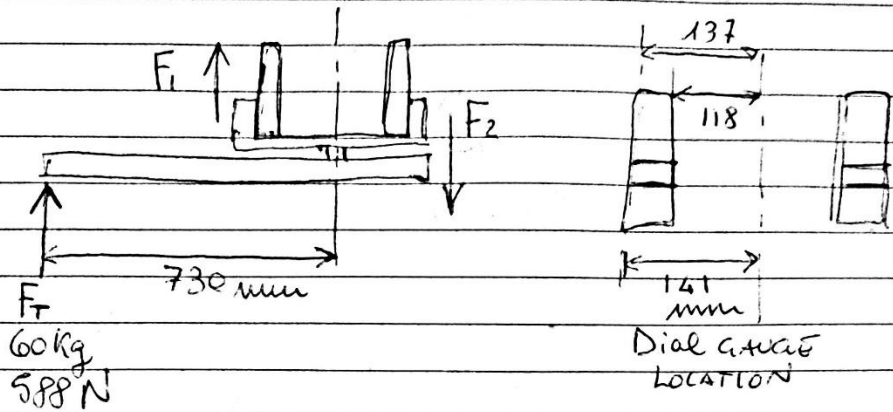
$$\therefore F_1 = F_2 = F$$

$$F \cdot 0,06 + F \cdot 0,06$$

$$2 F \cdot 0,06$$

$$T_{\text{Total}} = 2 \cdot F \cdot 0,06$$

REAR TWIST



$$M = b \cdot F = 588 \cdot 0,73 = 429 \text{ Nm}$$

at the center plane

$$F_1 = F_2 = F$$

$$b = \text{AVERAGE } 137 \text{ \& } 118 \text{ mm}$$

$$M_T = F_1 \cdot b + F_2 \cdot b$$

$$2 F b$$

$$F = \frac{M_T}{2 b}$$

