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DESIGN PROCESS OF A PROTOTYPE MOTORCYCLE SWINGARM

MAJOR PROJECT

Dissertation submitted to the University of Wales Trinity Saint David
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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.

ABSTRACT

This work presents the design process followed by the author to develop a prototype swingarm for the Yamaha R6 of the University of Wales Trinity Saint David 's motorcycle racing team, to race at the Isle of Man TT racing circuit.

Initially, the original swingarm was evaluated through experimental tests for its lateral and torsional stiffness and for natural frequency values. Then, in order to study the strength of the swingarm, the load cases were identified and calculated resorting to real data acquired by the racing team at the racing circuit in 2019.

Following a market study and considering the manufacturing constraints available to the author, a concept design was proposed for the new swingarm. With this initial design a Finite Element model was developed to perform strength, stiffness and vibration analyses. A safety measure was defined by the author in which the resultant stress values from the Finite Element Analysis should not exceed the material's fatigue strength.

After performing an iterative optimisation process on the new swingarm design, all the strength, stiffness and vibration constraints were respected, and the objectives met. The swingarm stiffness was successfully tuned in the final design, with the lateral and the torsional stiffness values varying within the reference values. The natural frequencies resulting from the finite element modal analysis on the final design proved to be within an acceptable percentage of error when compared to the experimental vibration test performed to the original swingarm.

The project culminates with a new optimised swingarm design, which respects the safety and performance characteristics of the original one, also achieving a lighter result with accurate considerations to be manufactured.

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1

INTRODUCTION



1. INTRODUCTION

1.1. AIMS AND OBJECTIVES

Within the scope of the module of Major Project of the third year, level 6 of the Motorcycle Engineering Degree at the University of Wales Trinity Saint David, this work intends to provide a detailed exploration of the design process of a prototype motorcycle swingarm, developing guidelines that will enable the adoption of the same design process to the development of other major structural chassis components in a motorcycle.

This work will lead to the final objective of designing a motorcycle swingarm for the University's racing team motorcycle, a Yamaha R6, to race in the Isle of Man TT, considering the identification of the load cases during a race and respecting the chosen constraints (available volume, suspension geometry, stiffness and natural frequency values of the original swingarm), while accomplishing a new and lighter swingarm design.

1.2. MOTIVATION

The motivation behind the project was underpinned by the author's passion for motorcycles and racing, as well as by the engineering challenge comprised by designing a component that will constantly perform at the motorcycle's limits of traction and performance.

The project will also allow the author to improve skills and divide his workflow between diverse software products: 2D-Data Recording; Matlab; GOM Inspect and Solidworks, as well as develop close contact with different experiments and manufacturing processes, such as stiffness jigs, LDV experiment, CNC machining and welding.

Besides the hard skills detailed above, soft skills will also be implemented and further developed, due to the direct contact to various lecturers, workshop technicians and industry experts during the project, allied to restrict time management and organisational skills, that will benefit the author once applying for jobs in the industry.

1.3. METHODOLOGY

For a better understanding of the topics, this dissertation is divided into progressive stages, that will guide the reader through the sequential processes defined by the author.

The initial stage – *Evaluation of Initial Design* - consists of the original swingarm being experimentally tested, in order to acquire performance constraints, such as lateral and torsional stiffness and natural frequency values. Within the following stage – *Load Cases* -, the maximum load cases in a racing scenario are defined, employing real data into motorcycle dynamic models and empirical formulas. Lastly, on *Finite Element Analysis*, once the material and the fatigue criteria are defined, a CAD model of the new design is developed, allowing to perform all the necessary numerical analysis until the constraints values for performance and safety are reached.

Success will be measured by meeting all the defined constraints and objectives in the Finite Element Analysis environment and, also, by being able to validate the numerical results against experimental results obtained from testing the new designed swingarm. Achieving an admissible correlation between the results will represent a powerful tool when designing new swingarms with different stiffness characteristics.

1.4. TIMING PLAN

When defining the timing plan for the work to be carried out, two different streams of work were highlighted. The first one, taking place in the first term of the academic year, comprises the research, definition of load cases, safety and performance parameters of the swingarm, leading to a first design concept for the new swingarm. It is important to have in consideration that compromises have been made in certain aspects of the work (which are further explained throughout the report), in order to assure the conclusion of this first stream of work within the available time frame. The second and final stream of the project focuses on the performance optimisation of the swingarm design, while guaranteeing that all the design features are designed in accordance with the manufacturing processes available to the author (see *Chapter 12.1. Timing Plan* of the *Appendix* for a detailed Gantt Chart of the planning).

2

RESEARCH AND LITERATURE SURVEY



2. RESEARCH AND LITERATURE SURVEY

A motorcycle swingarm is one of the most important structural components in a motorcycle, being responsible for the rear suspension mechanism, performance and safety of the motorcycle while in motion.

The main purpose of the swingarm consists of connecting the frame and engine to the rear wheel of the motorcycle and it is characterized by three connection areas:

- a pivot point, where the swingarm attaches to the frame or engine, allowing a quasi-vertical motion of the rear wheel;
- a rear area, where the swingarm connects to the rear wheel, compelling a perfect alignment with the chain drive and between the rear and front wheels of the motorcycle;
- a suspension mounting point, where the swingarm connects to the rear suspension directly or to a suspension linkage system that will then operate the rear suspension.

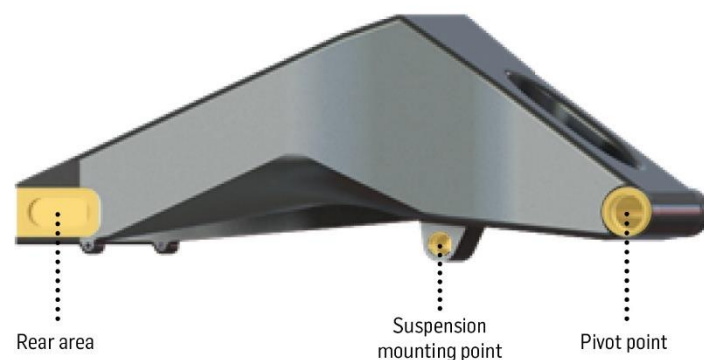


Figure 1. Motorcycle Swingarm with the three connection areas highlighted

The strength of the swingarm is characterised by the ability of the swingarm to cope with all the load cases in a certain riding scenario. All the different load cases - acceleration, braking, cornering and suspension - are well described in *Cossalter, 2016*, along with all motorcycle dynamic models and empirical formulas used to obtain the load values. However, due to the lack of input data and limited examples presented, it was not possible to use those results as the load cases reference, to design the new swingarm.

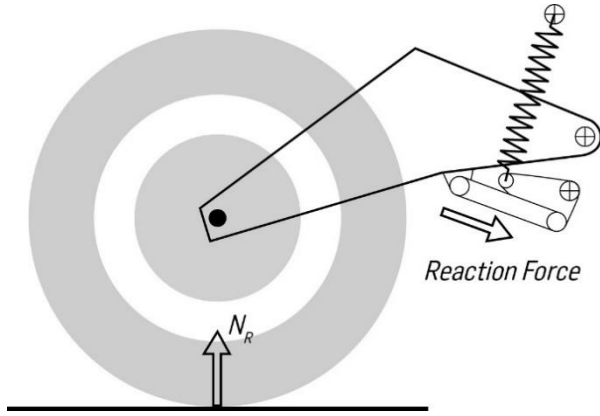


Figure 2. Suspension forces

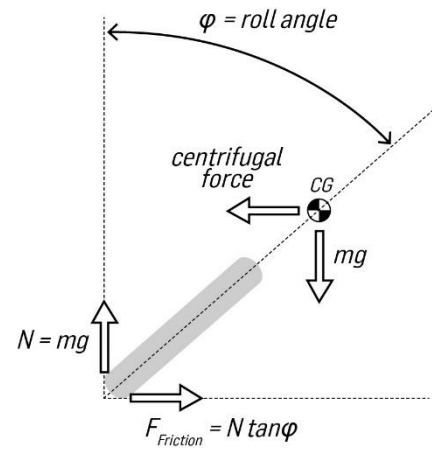


Figure 3. Cornering forces

What is stated above motivated the author to acquire real data, from the scenario in which the new swingarm is meant to perform, in order to feed them into the motorcycle dynamic models and empirical formulas - consequently obtaining more accurate approximations for the maximum load values acting upon the swingarm.

Besides the safety concerns previously exposed, the stiffness of the swingarm is required to be tuned to improve the stability, response and performance of the swingarm. The majority of the load cases are applied in areas with an offset in relation to the centre of the swingarm (chain drive, brake disc, etc.), which produce twisting moments that will harm the wheel alignment and geometry of the motorcycle, resulting in poor performance (torsional stiffness). When cornering, the motion of the rear suspension is not aligned with the surface irregularities, being the swingarm responsible for offering some compliance and flex in order to potentiate traction and comfort to the rider (lateral stiffness).

The torsional and lateral stiffness values vary in a range of values depending on the weight, power and purpose of the motorcycle. These range of values well tested and tuned by the manufacturers, yet there is no published information, thus pointing the author to a physical test being the only solution to retrieve those values from a swingarm.

From the literature survey carried out, some recent technical paper, were found, concerning the stiffness properties of various motorcycle swingarms. All the tests undertaken were performed following the same principle presented in *Cossalter, 2006*.

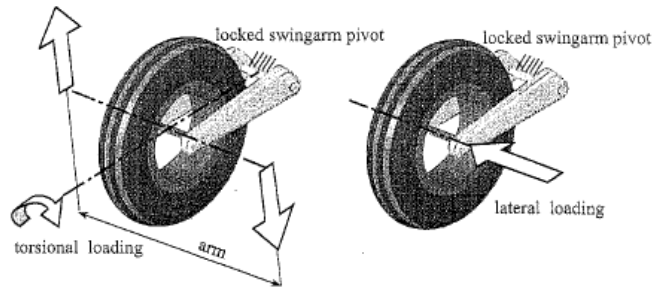


Figure 4. Loading condition to evaluate Torsional and Lateral Stiffness
Source: Cossalter, 2006

A discrepancy of values between technical papers was observed – as can be checked below on *Table 1* - due to different experiment set up and deficient measures in the experimental testing, caused by not stiff enough jigs and different force applications.

Even though the stiffness values examined could not be used as reference, the work developed by Papadakis, 2011, represents a useful contribution when comparing stiffness properties of swingarms from different types of motorcycles.

Table 1. Stiffness comparison from different references

	(Papadakis, 2011)				(Risitano, 2012)			(Cossalter, 2006)
	CRF 450	TZ 250	CBR600 F2	CBR600 RR	WSBK 2008	WSBK 2008	WSBK 2008	Sportbike 1000cc
Lateral Stiffness (N/mm)	157.53	188.8	337.67	357.63	-	-	-	800-1600
Torsional Stiffness (Nm/°)	318.82	754.2	545	987.48	666	930	1300	1000-2000

Besides physically loading the swingarm to acquire stiffness values, a modal analysis could be performed to achieve stiffness relations without the need to load a single component. Therefore, the natural frequencies and frequency modes of the original swingarm had to be obtained to allow for the design of the new swingarm with the same performance and safety concerns. The natural frequencies of the swingarm could be defined as the frequencies in which the swingarm is prone to vibrate if submitted to a disruption (disturbance), and are defined by:

$$f = \frac{\sqrt{\frac{k}{m}}}{2\pi}$$

Consequently, these frequency values represent a reference of the relation between the overall stiffness of the swingarm and the respective mass, as well as the frequency values to aim the new design at, in order to avoid the instability modes of the part.

During the literature survey, the data found concerning the vibration topic was limited, mainly due to the cost of the equipment required to perform an experimental modal analysis (Laser Doppler Vibrometer). It were then found, natural frequency values concerning a motorcycle swingarm with similar weight, power and performance pretentions as the one analysed by the author, as observed below in the table.

Table 2. Natural frequencies comparison from different references

Frequency Modes	(Risitano, 2012) WSBK 2008 (Hz)	(Pagano, 2017) Ducati 899 (Hz)
I	115	169
II	261	337
III	304	422
IV	579	501

In terms of design, modern swingarms, have iterated into two different types of designs – double sided swingarms and single side swingarms. In a double sided swingarm, due to its almost symmetrical design, there is an even distribution of the stress suffered by the swingarm, making the design and simulation process quicker and more efficient, consequently reducing the final cost. A single sided swingarm adds more complexity to the design and simulation process, due to its asymmetrical nature, but has the advantages of providing an easier and quicker rear wheel removal and a fixed wheel alignment from the manufacturer, while allowing the exhaust to be tucked closer to the rear wheel, improving the aerodynamic performance of the motorcycle.



Figure 5. Example of double sided swingarm



Figure 6. Example of single sided swingarm

In MotoGP - the most technically advanced motorcycle racing series - where cost and time are less of a constraint and performance is the ultimate objective, the double sided swingarm design is the set-up opted by all the racing teams.



Figure 7. Suzuki GSX RR MotoGP motorcycle

The manufacturing method to be used in this project was determined by the manufacturing processes available in the University's workshop, which implied that the new swingarm would be built from billets of aluminium CNC machined.

The CNC (computer numerical control) machining process is frequently adopted for small productions or for prototyping parts. It allows to progress from a CAD file to a real part in a short period of time. Although the cost per unit is relatively high. Other current manufacturing processes are used to produce swingarms, such as casting, hydroforming or cold pressing, requiring an investment in hardware and tooling in order to reduce the cost per unit that will only be profitable in mass production parts.

3

EVALUATION OF ORIGINAL DESIGN



3. EVALUATION OF ORIGINAL DESIGN

A description of the work carried out by the author, will be presented in the following sub-chapters in a logical order, that will enable the reader to follow the steps involved in the design process a motorcycle swingarm.

Once the initial swingarm design and the performance environment are assessed, it will be possible to define the physical and performance constraints for the new prototype swingarm and, in addition, to trace engineering objectives.

The identification of all the load cases during the defined performance environment will follow, as well as the decision about the material and manufacturing concerns and several Finite Element Analysis (FEA) to assure that all the constraints and objectives of the project are met.

3.1. MOTORCYCLE SPECIFICATIONS

The motorcycle chosen by the author for the design of the new prototype swingarm is the Yamaha R6 from 2009 – the current motorcycle of the University's racing team. Subsequently, the specifications of the racing motorcycle were laid out on the following table, in which the values were either measured by the author, supplied by the manufacturer or assumed. Some values may differ from what is publicly presented by the manufacturer, due to the current set-up of the motorcycle for road racing.



Figure 8. Yamaha R6 2009

Table 3. UWTSD's Yamaha R6 2009 specifications

YAMAHA R6 2009 SPECS – Racing Team UWTSD		
GEOMETRY	VALUE	UNITS
Rake	24	°
Trail	98.87	mm
Wheelbase	1395	mm
Swingarm angle	-12.7° to 0.09°	°
Cog Bias	52/48	%
Cog Y	530	mm
WEIGHT		
Dry weight	166	Kg
Wet weight	188	Kg
Rider weight	80	Kg
Total Mass	2626.4	N
FRONT SUSPENSION		
Spring Rate	9.5	N/mm
Damping coefficient	2.045	Ns/mm
Wheel travel	120	mm
REAR SUSPENSION		
Spring Rate	85	N/mm
Damping coefficient	7.8	Ns/mm
Suspension travel	60	mm
Link Ratio	1.88 or 2.05	
TYRES		
Friction coefficient	1.3	
Front tyre width	120	mm
Rear tyre width	180	mm
Front tyre diameter	599.8	mm
Rear tyre diameter	629.8	mm
GEARING		
Front sprocket	16	
Rear sprocket	45	
Primary Ratio	2.073	
Gear 1 Ratio	2.583	
Gear 2 Ratio	2.000	
Gear 3 Ratio	1.670	
Gear 4 Ratio	1.444	
Gear 5 Ratio	1.286	
Gear 6 Ratio	1.150	

The values on the table above, allied to real data recorded from the motorcycle on a racing environment will be used on *Chapter 4. Load Cases*, to feed motorcycle dynamic models on Matlab and empirical formulas in order to calculate maximum suspension, acceleration, braking and cornering forces acting upon the swingarm.

3.2. PERFORMANCE ENVIRONMENT

The racing environment in which the motorcycle will perform and for which the swingarm is designed for is the Isle of Man TT circuit (IOM TT). It is an annual motorcycle road race run in the Isle of Man, in which the circuit is composed of 37.73 miles of the island's public roads.

This event, also known as Tourist Trophy, is undoubtedly the most demanding racing circuit for a motorcycle, mainly due to the maximum and average speeds achieved and the nature of the circuit - public roads. Consequently, the perfect balance between performance and safety is required, to assure that all the motorcycle's components are well engineered to constantly perform on the limit.

With the data acquired by the University's racing team at the IOM TT 2019, the author was able to export and process the relevant information into Matlab and obtain a general analysis of a lap during different racing scenarios, such as acceleration, braking and cornering, shown in the following figure.

```
ISLE OF MAN - GENERAL ANALYSIS OF ONE LAP

ACCELERATION
Percentage of time on the Throttle: 78.31%
Percentage of time using the Throttle with 90% TPS or more: 33.22%

BRAKING
Percentage of time using the Front Brake: 19.07%

SUSPENSION
N° of cycles from rear suspension: 6754
Rear suspension travel(mm) while accelerating:
- Average: 18.69; - Max: 54.49; - Min: -0.11
Rear suspension travel(mm) while braking:
- Average: 10.49; - Max: 45.92; - Min: -0.11

CORNERING
Velocities (km/h) for banking angles between 0 and 20(53.99%):
- Average: 211.67; - Max: 285.50; - Min: 29.10
Velocities (km/h) for banking angles between 20 and 35(24.87%):
- Average: 195.70; - Max: 280.70; - Min: 29.60
Velocities (km/h) for banking angles between 35 and 50(18.12%):
- Average: 177.20; - Max: 269.10; - Min: 29.60
Velocities (km/h) for banking angles between 50 and 61(3.01%):
- Average: 156.66; - Max: 249.60; - Min: 39.00
```

Figure 9. General analysis of one lap at the IOM TT circuit from UWTSD's racing team

3.3. STIFFNESS EXPERIMENT

For the 2nd year project, the author performed a set of stiffness experiments to the original swingarm. The objective was that through physical experiments to obtain the torsional and lateral stiffness values of the original swingarm, so that they could be used as constraints when designing the new optimized design.

The experiment was then replicated in a numerical simulation environment (FEA), in order to validate the results from the software package used, and/or to identify the corresponding error when comparing them with the results of the physical experiments.

Unfortunately, the jig built for the experiment was proven not to be stiff enough, which led to the impossibility of validating the results obtained.

Consequently, it was decided to proceed with the design of the new prototype swingarm using stiffness values from previous technical papers as reference. Once the new swingarm would be built, it would be tested in an outsourced proven stiffness jig, allowing the validation of all the simulations performed.



Figure 10. Stiffness experiment to the original swingarm

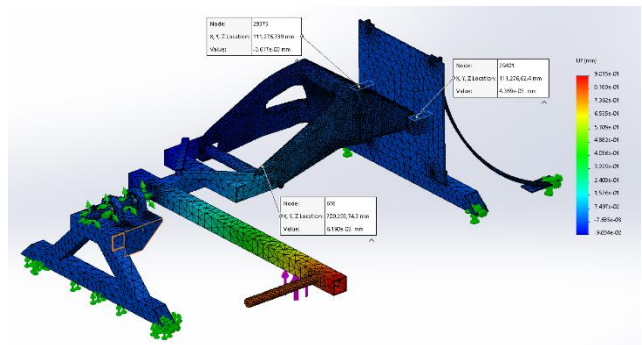


Figure 11. Stiffness FEA to the original swingarm

3.4. VIBRATION EXPERIMENT

As mentioned previously, the aim of the author is to design a swingarm with the same performance and safety concerns as the original swingarm. To do so, a modal analysis employing a Laser Doppler Vibrometer (LDV) on the original swingarm was performed.

A modal analysis uses the overall stiffness and mass of a structural part to define the frequency values in which the swingarm will naturally vibrate. These frequency values are denominated as natural frequencies and the deformed shapes of the swingarm at these frequencies are denominated as modes of vibration.

It is of vital importance that the natural frequencies of the studied swingarm do not equal the frequencies expected to be emitted by the motorcycle in its performance environment (from engine rpm, road irregularities or from other motorcycle parts), to avoid instability modes and structural damage.

ALDV experiment allows to measure the vibration velocity and displacement at fixed targets in the swingarm using the Doppler Effect. The advantage of this experiment against others of the same purpose, is that it allows vibration measurements in areas of difficult access, while there is no need to load the component.



Figure 12. LDV experiment to the original swingarm

With the proposed set-up, minimal errors are expected due to the complexity of creating an ideal LDV experiment. Those errors are related with the impact promoted to the swingarm not being a perfect impulse (infinitely small in duration) and to the unviability of creating an experiment with the swingarm free in space.

The experiment was then performed to various points in different surface areas, as demonstrated in the images below:

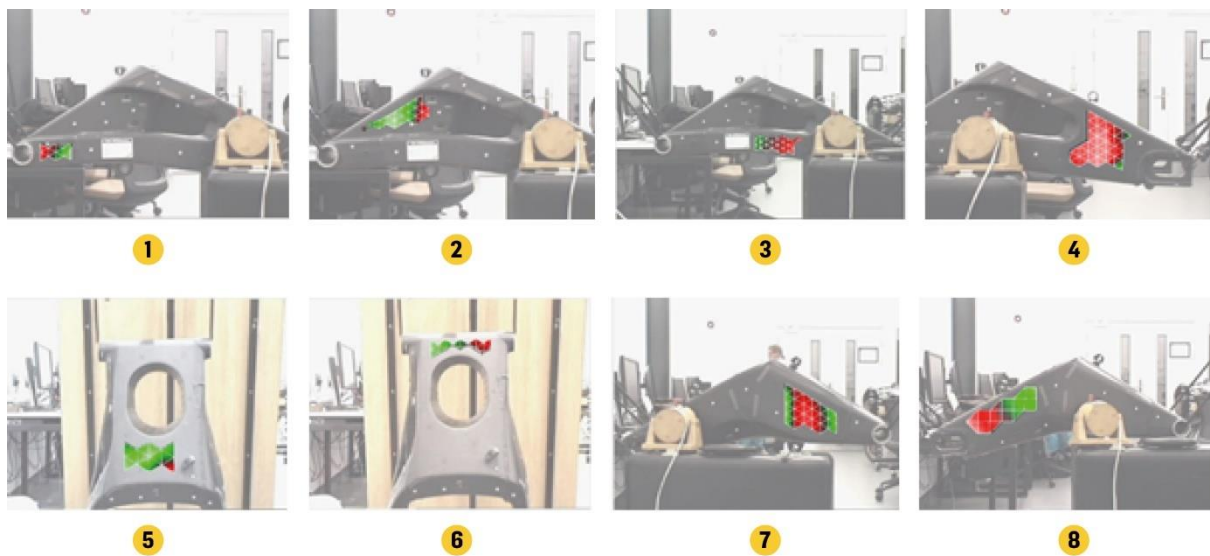


Figure 13. Surface areas tested in the LDV experiment

The natural frequencies measured for each surface were the following:

Table 4. LDV experiment results - natural frequencies

	1 st Mode (Hz)	2 nd Mode (Hz)	3 rd Mode (Hz)
1 - Left Side	115.625	462.5	
2 - Left Side	115.625		
3 - Left Side	115.625	462.5	
4 - Left Side	112.5	306.5	456.25
5 - Top View	100	315.625	468.75
6 - Top View	112.5	315.625	468.75
7 - Right Side	115.625	462.5	940.625
8 - Right Side	115.625	462.5	

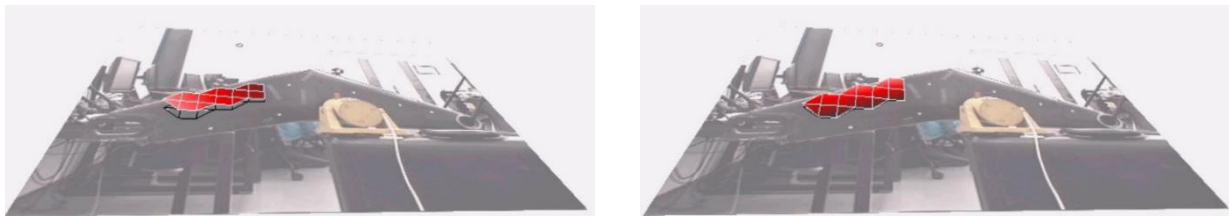


Figure 14. 1st and 2nd Modes of Vibrations of original swingarm from LDV experiment

In the figure above, an example of the 1st and 2nd Modes of Vibration (mode shapes) for the 8 - Right Side is presented.

Later in the work, on *Chapter 6.4. Modal Analysis*, all the Modes of Vibration resulting from the experiment will be compared against the FEA results from the new swingarm design.

Considering that the natural frequencies of the swingarm should not equal the frequencies expected from the engine, the range of frequencies imposed by the engine rpm should be calculated in order to undertake an analysis. It is known that the range of rpm in the motorcycle being studied varies from 1500 rpm (idle) to a maximum 14500 rpm, which correspond to 50 Hz and 483,3 Hz, respectively.

As concluded from the previous calculation, the natural frequencies from the 1st and 2nd mode of vibration of the swingarm are within the minimum and maximum frequencies imposed by the engine rpm. Even though the matching frequency values lead to instability modes and structural damage to the swingarm, the author is aware that all the other components of the rear assembly (wheel spindle, rim, tyre, etc.) will introduce added mass and different stiffness, consequently changing the frequency response.

Nevertheless, it is important to ensure that the natural frequencies of the new swingarm design equal the ones from the original swingarm to maintain the same performance and safety standards.

4

LOAD CASES



4. LOAD CASES

In this chapter the author identifies and calculates an approximation of the loads suffered by the swingarm during a lap in its performance scenario.

Real data - acquired through sensors in the racing motorcycle during a lap performed by the UWTSD's Yamaha R6 at the IOM TT 2019 circuit - is employed in motorcycle dynamic models and empirical formulas mainly sourced from *Cossalter, 2006*, and in motion simulations developed by the author in order to obtain the load values that are later used in the FEA simulations.

4.1. SUSPENSION FORCE

As analysed in *Chapter 3.2. Performance Environment*, the rear suspension of the motorcycle performs 6754 cycles during one lap of the circuit. Each cycle is characterized by a compression movement of the rear suspension followed by a rebound movement.

To ensure that the new swingarm design has the strength to cope with the loads going through the suspension, the maximum value within those suspension cycles needs to be identified and analysed.

Once the limit working scenario of the rear suspension is identified, the maximum reaction force at the suspension mount is calculated and consequently the maximum normal force at the rear wheel that generates that reaction.

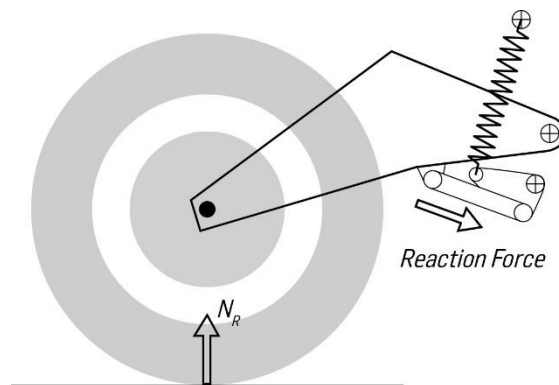


Figure 15. Load case - suspension force

Therefore, so as to calculate loads in the swingarm from the rear suspension data, the complete design of the rear suspension mechanism with its correct geometry is required. This includes all the moving parts

connected to the rear suspension - rear shock and linkage system - respecting all the geometrical relations between these parts and the swingarm.

To do so, the approach consisted of firstly measuring all the components and relations between them. A reference measure in an image was used to scale and record all the other measures and relations needed.

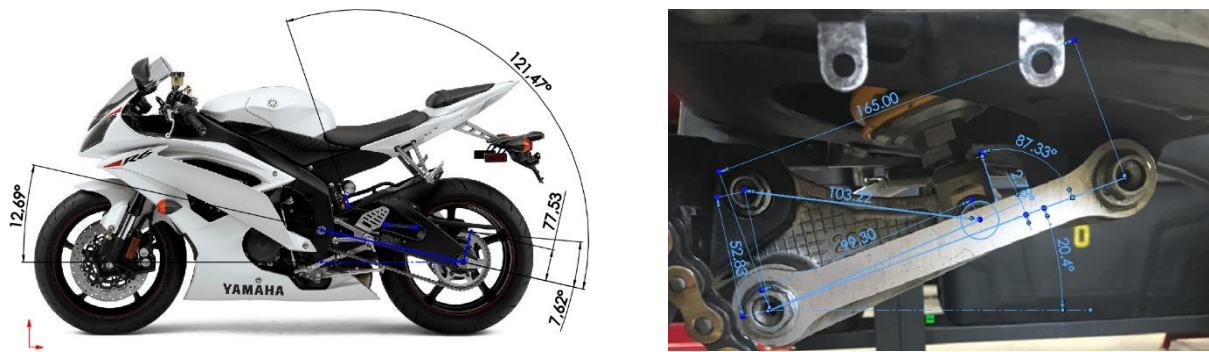


Figure 16. Yamaha R6 2009 - rear suspension geometry analysis

Subsequently, all the components were designed and correctly assembled in a CAD software - Solidworks - allowing the author to perform a motion study in order to evaluate the link ratios obtained, thus validating the work completed so far.

Through the manufacturer's motorcycle specifications, it was known that the rear wheel vertical travel and the rear suspension travel was of 120 mm and 60 mm respectively, implicating that a link ratio near 2 was expected.

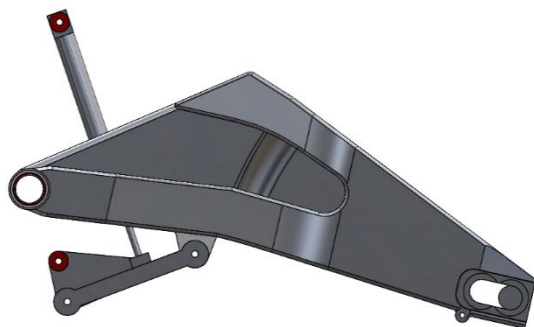


Figure 17. CAD of rear suspension assembly

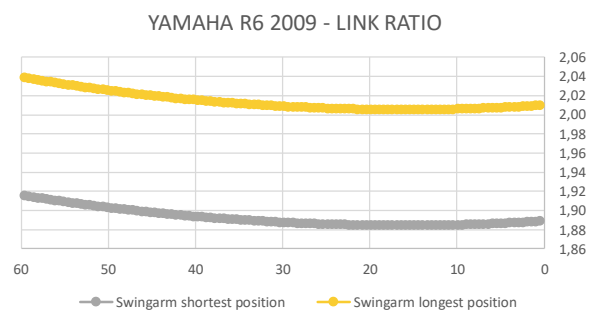


Figure 18. Rear suspension link ratio

As it can be observed above in *Figure 18*, the link ratio of the swingarm from the longest position of the rear wheel to the shortest ranges from 2 to 1,90 respectively. Also, by analysing the trend of the link ratio, an almost linear, slightly progressive behaviour of the rear suspension movement could be noticed.

With the previous results, the author was able to validate the work developed so far, enabling further developments to be made in order to get the suspension reaction force and rear normal force.

Thus, it is relevant to define the limit situations in which the rear suspension performs. To do so, motion studies were developed to analyse two distinct situations:

- Resultant maximum suspension reaction force and rear normal force when using the maximum suspension acceleration recorded to feed the motion study (using real data collected at the IOM TT);
- Resultant maximum suspension reaction force and rear normal force from a hypothetical limit scenario of the motorcycle going over a bump at its maximum velocity;

4.1.1. SUSPENSION REACTION FORCES USING REAL DATA

Having the rear suspension geometry in a simulation environment, the author proceeded to identify the maximum velocity and acceleration observed in the rear shock of the motorcycle during a lap at the Isle of Man TT.

These values will be used to feed the rear suspension motion study developed, in order to achieve the intended resultant maximum suspension reaction force and rear normal force.

After a cautious analysis (see *Chapter 12.2. Suspension Reaction Forces Using Real Data* of the *Appendix*) to why the maximum velocity and acceleration values collected from data should not be used as a reference, it was then decided to export all suspension data to Matlab in order to better process and analyse the velocity and acceleration distribution over a lap at the Isle of Man TT.

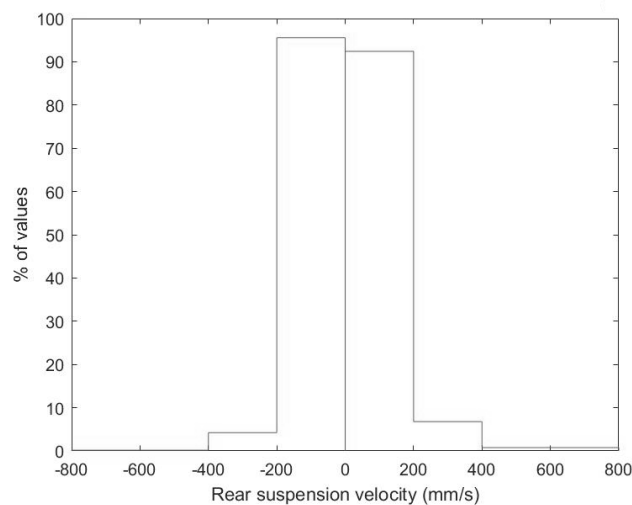


Figure 19. Distribution of rear shock velocity in a lap during the IOM TT 2019

Table 5. Rear shock velocity analysis in a lap during the IOM TT 2019

REBOUND			COMPRESSION		
Velocity Ranges (mm/s)	Nº of values	Percentage	Velocity Ranges (mm/s)	Nº of values	Percentage
[-200 : 0[	115481	95.58 %	]0 : 200]	95098	92.42 %
[-400 : -200[	5121	4.24 %	]200 : 400]	6970	6.77 %
[-800 : -400[	216	0.18 %	]400 : 800]	811	0.79 %
]inf : -800[	0	0 %	]800 : inf[	15	0.015 %

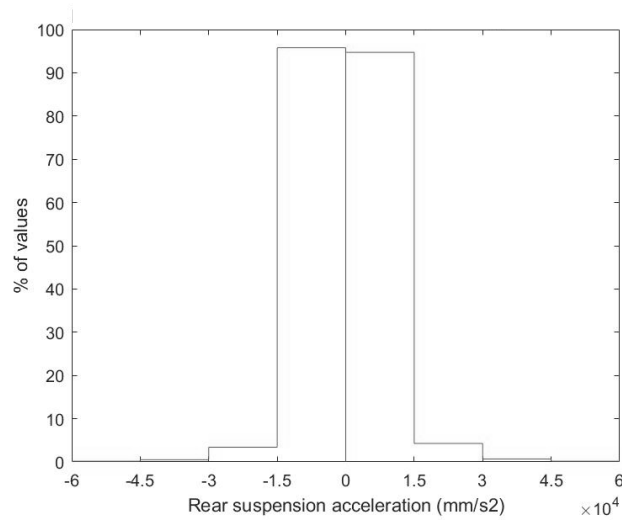


Figure 20. Distribution of rear shock acceleration in a lap during the IOM TT 2019

Table 6. Rear shock acceleration analysis in a lap during the IOM TT 2019

REBOUND			COMPRESSION		
Acceleration Ranges (mm/s^2)	Nº of values	Percentage	Acceleration Ranges (mm/s^2)	Nº of values	Percentage
[-15000 : 0[	112620	95.80 %	]0 : 15000]	104738	94.73 %
[-30000 : -15000[	4014	3.41 %	]15000 : 30000]	4721	4.27 %
[-45000 : -30000[	592	0.50 %	]30000 : 45000]	756	0.68 %
[-60000 : -45000[	193	0.17 %	]45000 : 60000]	197	0.18 %
]inf : -60000[	144	0.12 %	]60000 : inf[	154	0.14 %

Even though the ranges with the maximum values for velocity and acceleration represent a low relative percentage, in absolute terms there is still a great amount of values within them.

Having this fact in consideration, the velocity and acceleration values selected to be used in the motion study are the following:

Table 7. Rear shock max/min velocity and acceleration values

REAR SUSPENSION IOM 2019	
MAX / MIN Velocity	MAX / MIN Acceleration
400 mm/s	60000 mm/s^2

At this point, the motion study could be developed with the calculated suspension constraints in order to obtain the maximum suspension reaction force and maximum rear normal force.

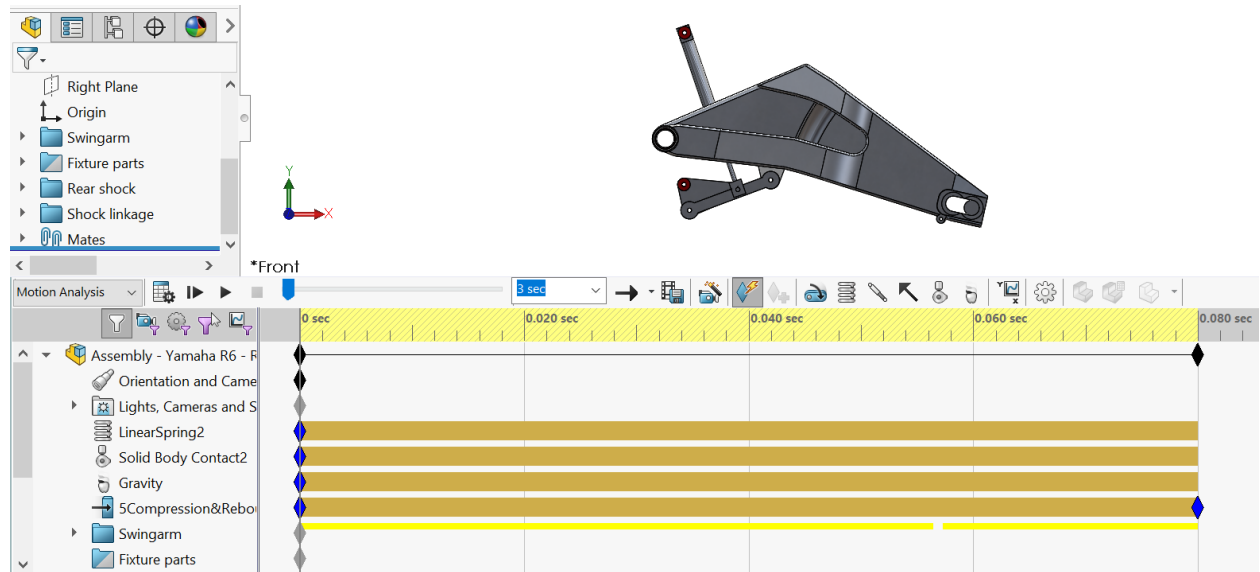


Figure 21. Rear suspension motion study

The study specifications and parameters were set as follows:

- The rear suspension was set at its full compressed position to assure the full travel would be analysed;
- The suspension specifications were calculated as shown below:
 - Rear suspension spring used in the motorcycle;
 - $k_R = 85 \text{ N/mm}$
 - Rear suspension damping coefficient calculations:
 - $K'_R = \frac{k_R}{LR^2} = \frac{85}{2^2} = 21.25 \text{ N/mm}$
 - $c'_c = c_c \times Z = 2\sqrt{K'_R \cdot m_{rear\ sprung}} \times Z$
 - $c'_c = 2\sqrt{21250 \times 91,7} \times 0,7 = 1954 \text{ Ns/m}$
 - $c_c = c'_c \times LR^2 = 1954 \times 2^2 = 7.817 \text{ Ns/mm}$
- Gravity was applied, as all the parts were given the correct weight in order to obtain the most accurate result possible;
- A motor was applied at the rear shock to replicate the maximum acceleration observed in the real data – 60,000 mm/s²;

With the current set-up, it was possible to plot the results of the suspension acceleration for validation purposes, the rear wheel normal force and the x and y components of the reaction force at the suspension mount in the swingarm.

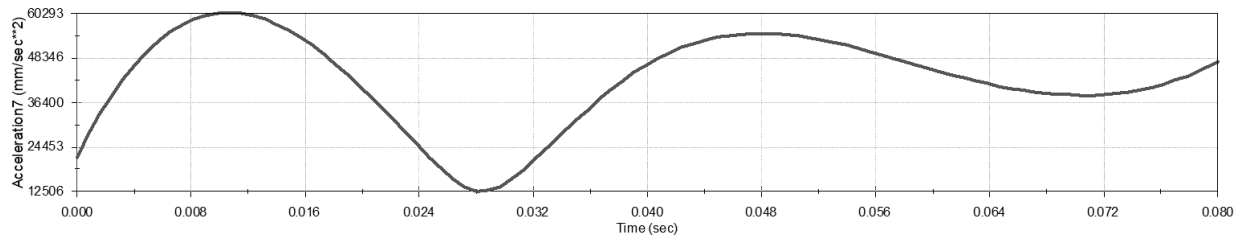


Figure 22. Motion study using real data - rear suspension acceleration

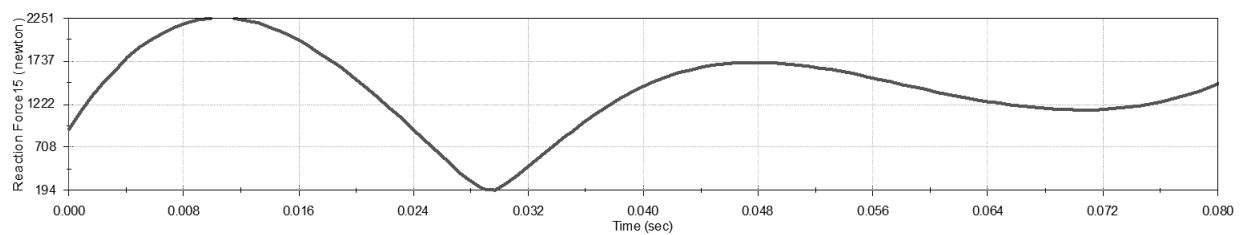


Figure 23. Motion study using real data - rear normal force

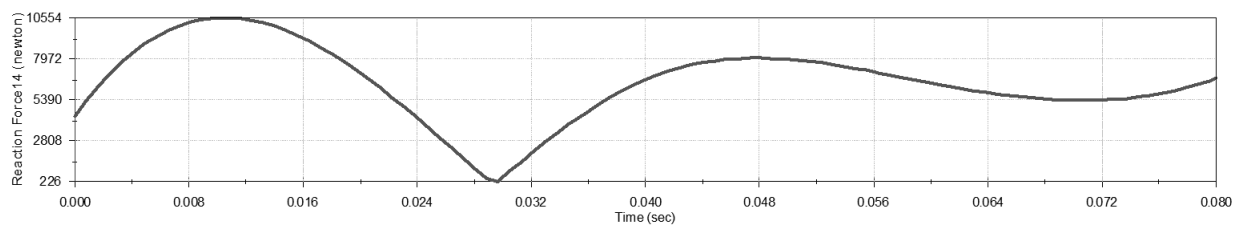


Figure 24. Motion study using real data - rear suspension mount reaction force magnitude

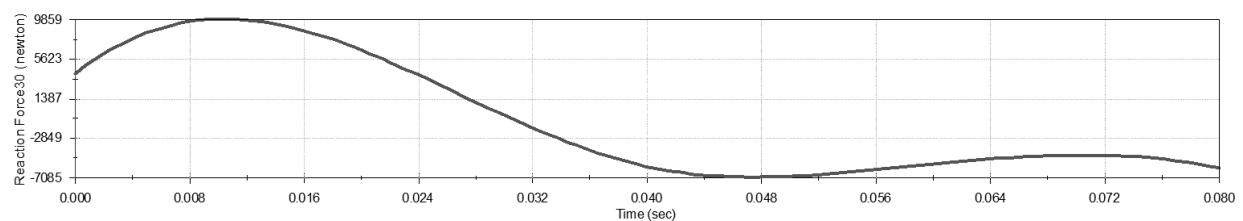


Figure 25. Motion study using real data - rear suspension mount reaction force y axis

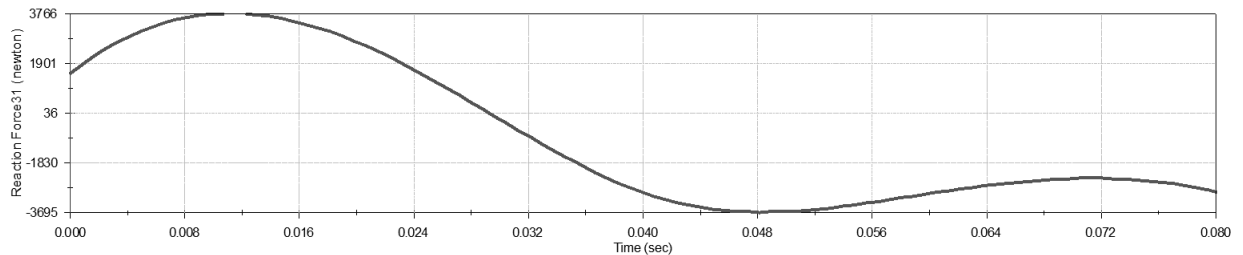


Figure 26. Motion study using real data - rear suspension mount reaction force x axis

From the analysis of *Figure 22*, it was observed that the same critical scenario acquired from real data of maximum suspension acceleration was possible to be created in a simulation scenario. This allowed to identify the normal rear force value that caused the maximum suspension acceleration recorded from data and, consequently, to identify the suspension reaction force components in the swingarm suspension mount.

In short, the suspension acceleration of $60,000 \text{ mm/s}^2$ recorded from data, resulted from a rear normal force applied at the rear tyre of approximately 2250 N.

Motivated by the extreme mechanical demands from the swingarm during a lap at the Isle of Man and the safety concerns over the new swingarm design, it was decided to perform a simulation over a maximum theoretical scenario.

In this maximum theoretical scenario, it is assumed that the rear suspension was already fully compressed (total mass of motorcycle + rider over the rear tyre) when the maximum acceleration value occurred. This is accomplished by setting a pre-load of 2626 N (*Figure 27*) plus applying a rear normal force of 2250 N over a period in time. The swingarm will end up experiencing an overall rear normal force of 4876 N.

For this maximum theoretical scenario, the resultant reaction force and components at the suspension mount are the following:

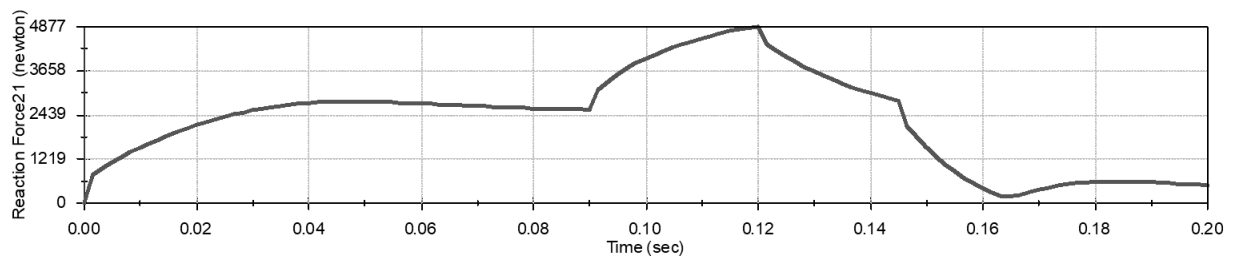


Figure 27. Motion study for maximum scenario - rear normal force

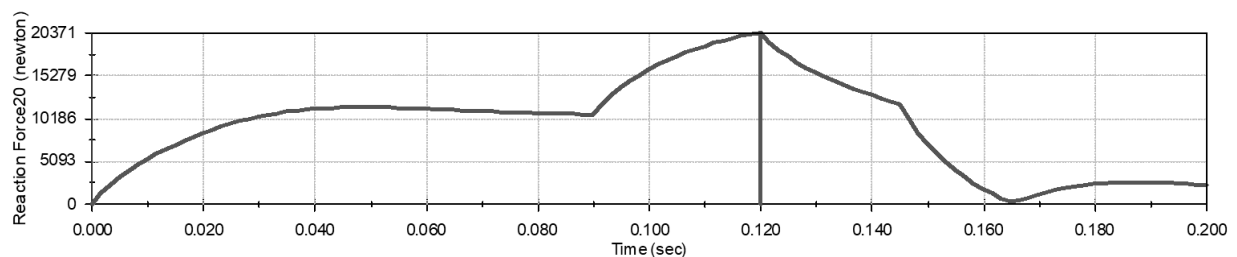


Figure 28. Motion study for maximum scenario - rear suspension mount reaction force magnitude

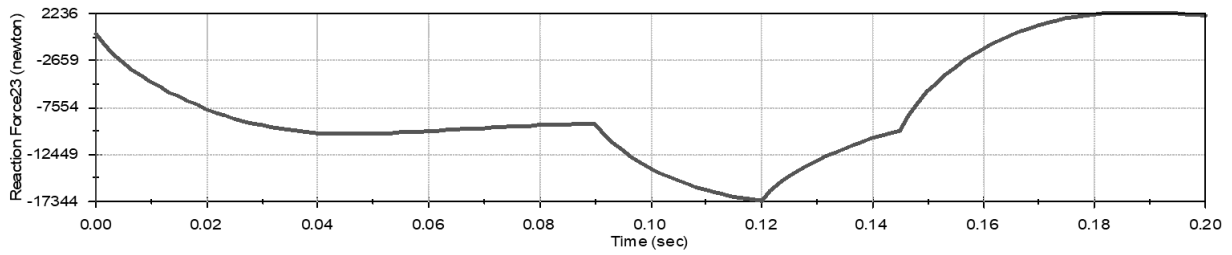


Figure 29. Motion study for maximum scenario - rear suspension mount reaction force y axis

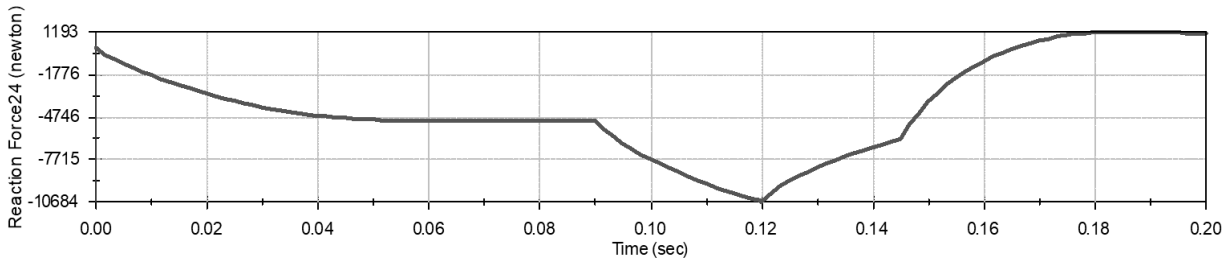


Figure 30. Motion study for maximum scenario - rear suspension mount reaction force x axis

From the analysis of the plots above, it was defined that the suspension load case to be used, when performing FE analysis to the new swingarm design, is the following:

$$Normal\ Force_{Rear\ Wheel} = 4876N$$

$$X\ Reaction\ Force_{Suspension\ Mount} = 17344N$$

$$Y\ Reaction\ Force_{Suspension\ Mount} = 10684N$$

4.1.2. SUSPENSION REACTION FROM GOING OVER A BUMP

A motion simulation of the rear wheel going over a bump at the motorcycle's maximum velocity was performed, in order to create in some way a validation process to the maximum rear normal force calculated in Chapter 4.1.1. *Suspension Reaction Forces Using Real Data.*

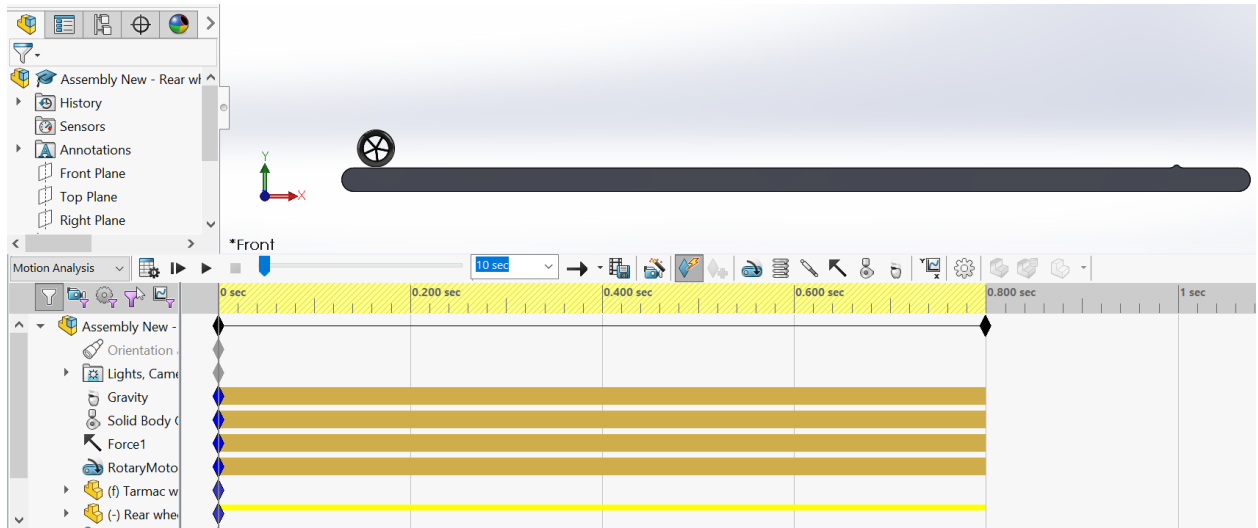


Figure 31. Motion study going over a bump

The study specifications and parameters were set as follows:

- Gravity was applied, as all the parts were given the correct weight to obtain the most accurate result possible;
- A solid body contact was defined between the tyre and the tarmac with a coefficient of friction equal to 1 - the maximum allowed by the software;
- A vertical force was applied to the rear wheel with a value equal to 2462 N, corresponding to the total mass of the motorcycle + rider, minus the weight of the rear wheel assembly;
- Then, a rotary motor was applied to the rear wheel with a value corresponding to the maximum real linear velocity of the motorcycle = 78.75 m/s (283,5 Kph).

The resultant reaction force between the tyre and the bump was then copied as the rear normal force to the simulation represented in *Chapter 4.1.1. Suspension Reaction Forces Using Real Data*. This allowed the calculation of the swingarm suspension mount reaction force and consequently the validation against the results obtained from *Chapter 4.1.1*.

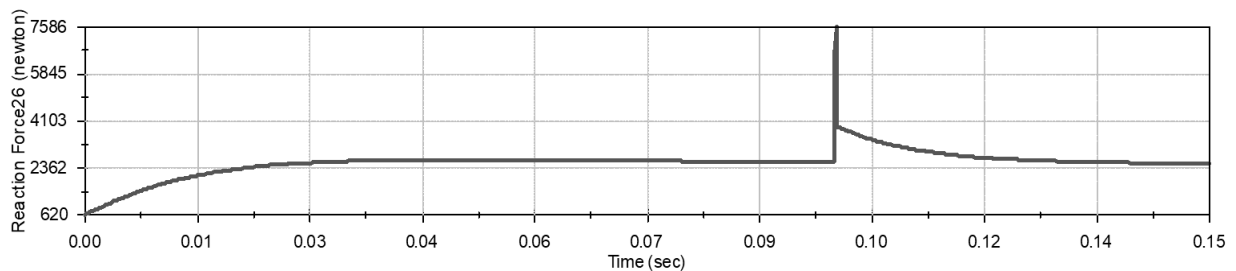


Figure 32. Motion study going over a bump - tyre contact force

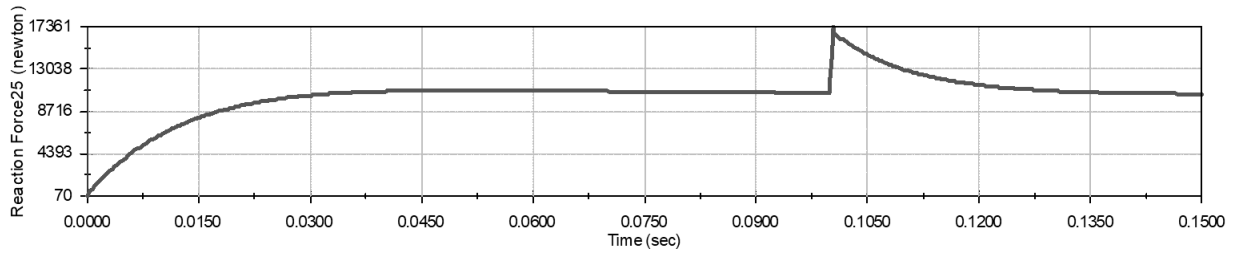


Figure 33. Motion study going over a bump - rear normal force

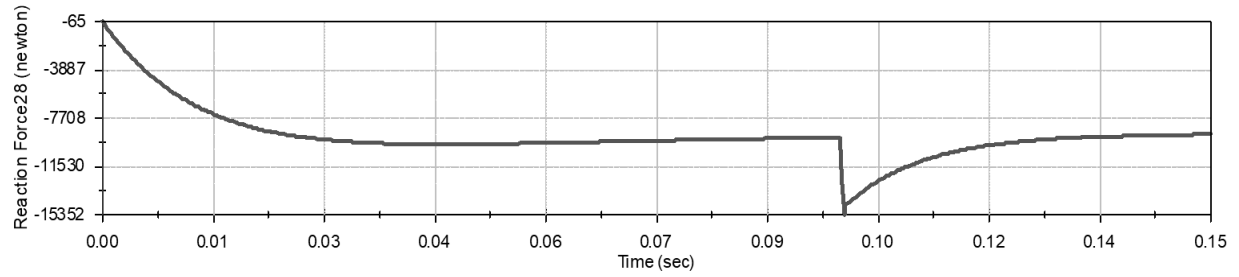


Figure 34. Motion study going over a bump - suspension mount reaction force y axis

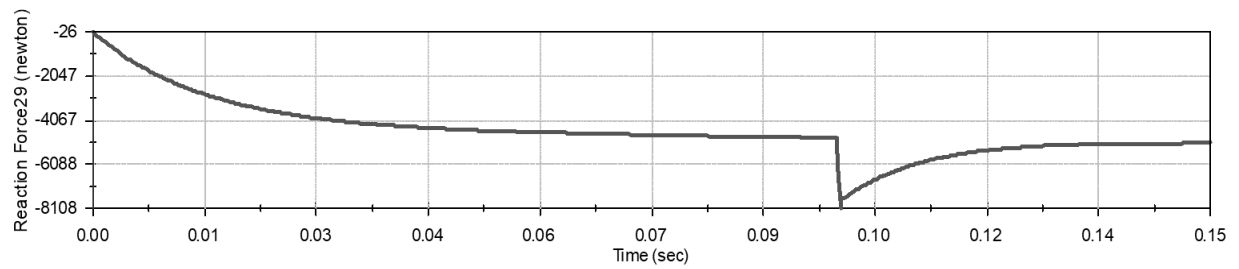


Figure 35. Motion study going over a bump - suspension mount reaction force x axis

When comparing the results above against the results from *Chapter 4.1.1*, it can be concluded that the values presented in the maximum theoretical scenario created by the author were realistic. This validation is reached once the results from the simulation above, a possible real scenario where the motorcycle goes over a bump at its maximum velocity, are similar to the maximum theoretical value from *Chapter 4.1.1*. The maximum resultant reaction force at the suspension mount was 20371 N, when using the suspension real data, versus 17361 N, resulting from the simulation of the motorcycle going over a bump.

4.2. DRIVING FORCE

During a lap at the Isle of Man TT, with a duration of approximately 18 minutes, the driver spends almost 80% of the lap demanding power from the engine (throttle on), in which 33% of those 80%, the rider is requesting for the full power available of motorcycle (as it was observed in *Chapter 3.2. Performance Environment*).

With this recurrent power demand over a lap, it is relevant to identify the maximum driving force pulling the motorcycle forwards in order to assure the correct performance and safety concerns of the new swingarm design.

The power from the engine is transmitted through a chain to the rear sprocket at the rear wheel. Since the rear sprocket has an offset in relation to the wheel's centreline, it will cause a twisting moment on the swingarm, possibly resulting in poor dynamic performance of the motorcycle if the driving force is not well considered in the simulation phase of the new swingarm.

These considerations will promote the continuous alignment of the wheels, improving straight line and corner performance, as well as the powertrain's efficiency (chain), assuring the correct performance of the author's swingarm and its reliability.

In order to identify the maximum driving force and the force values observed during the lap at different velocities, the author developed a simulation of the maximum possible acceleration of the current motorcycle from 60 kph to its maximum velocity.

The simulation used the motorcycle geometry, power curve, primary and final gear ratios as input and the gears were changed in order to optimise the acceleration. As a result, through the use of motorcycle dynamic formulas (Cossalter, 2006), it was possible to calculate the maximum acceleration values for different limiting situations, such as wheelie, traction and power limited (see *Chapter 12.3. Driving Force* of the *Appendix* for the motorcycle dynamic formulas and Matlab code).

GENERAL ACCELERATION SIMULATION ANALYSIS

```
The time to accelerate from 60.00 km/h to 283.51 km/h is 20.76s!  
Maximum acceleration : 8.40 m/s2 - power limited || 9.73 m/s2 - wheelie limited || 7.26 m/s2 - traction limited  
Maximum forces : 2529.01 N - driving force || 2966.34 N - power limited  
Maximum normal forces : 1260.67 N - front normal || 2299.10 N - rear normal  
Minimum normal forces : 309.13 N - front normal || 1365.73 N - rear normal
```

Figure 36. Yamaha R6 - General acceleration simulation analysis

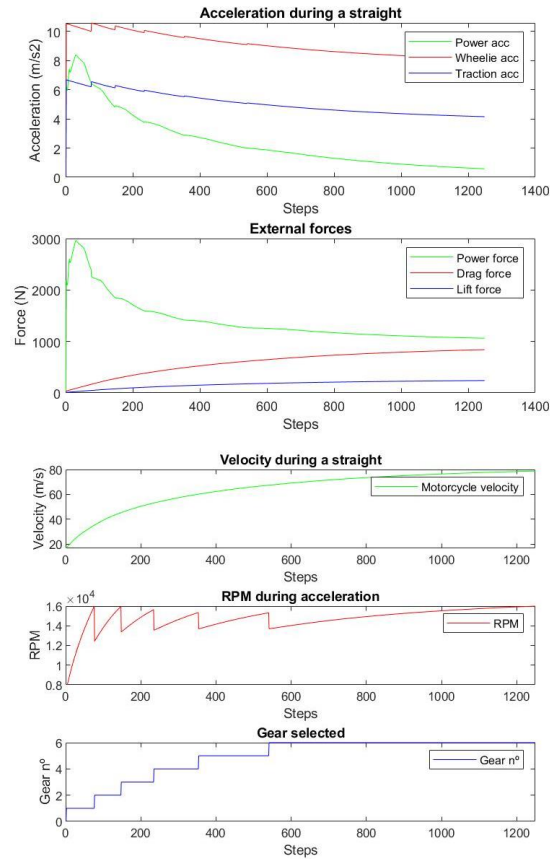


Figure 37. Yamaha R6 - acceleration simulation analysis

As the maximum acceleration and driving force values can never be superior, in a real scenario, to the maximum theoretical ones calculated from the motorcycle power and gear ratios, it was decided that the power limited acceleration would be used to calculate the maximum driving force applied to the rear wheel. Finally, for simulation purposes, the calculation of the maximum force in the chain direction was required.

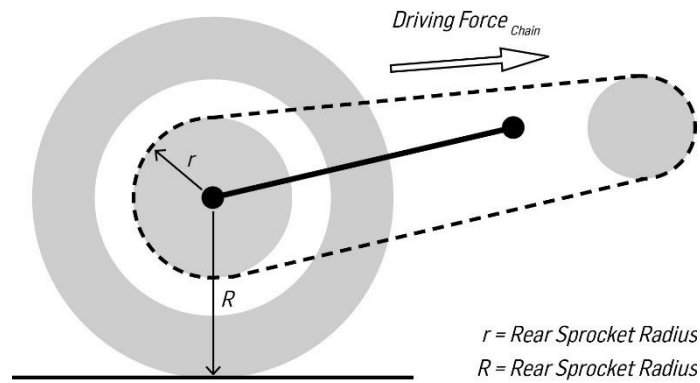


Figure 38. Free body diagram - driving force

$$Driving\ Force_{chain} = \frac{Torque_{wheel}}{Rear\ Sprocket\ Radius} (N)$$

Where

- $Torque_{wheel} = Driving\ Force_{wheel} \cdot Wheel\ Radius = 2966.34 * 0.315 (Nm)$
- $Rear\ Sprocket\ Radius = 0.1175 (m)$

$$Driving\ Force_{chain} = 7952.32 (N)$$

4.3. BRAKING FORCE

The maximum braking force that could be achieved on the rear wheel is only dependent of the maximum frictional force generated between the tyre and the tarmac.

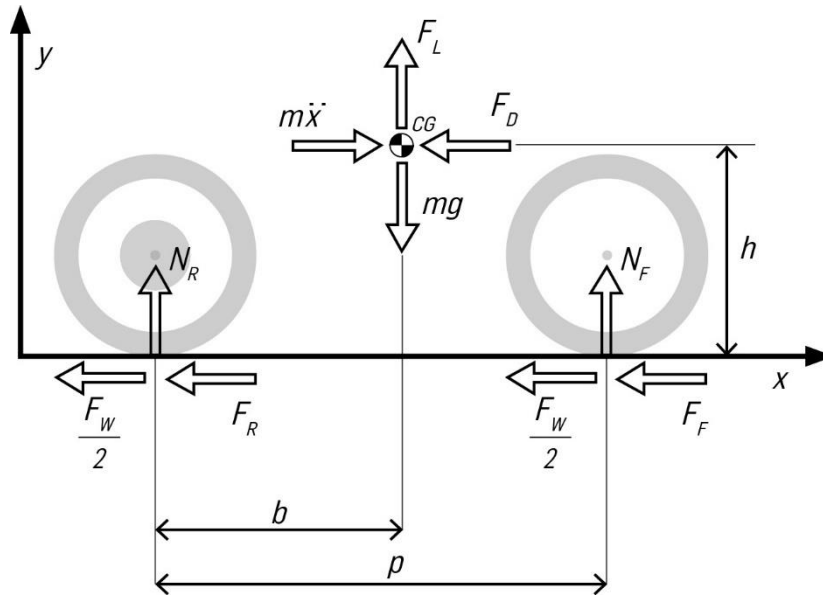


Figure 39. Free body diagram - braking force

$$F_{Rear\ Braking} = \mu N_R$$

According to *Brembo, 2019*, an average peak deceleration per lap was 1.16 G in a specific MotoGP circuit. As MotoGP is a much more extreme segment than the one being analysed, the same deceleration reference value was assumed for safety.

Assuming an extreme of having all of the mass over the rear wheel under braking and because $F = ma$,

$$F_{Rear\ Braking} = \mu N_R = ma$$

$$\mu = a = 1.16$$

Additionally, it was needed to calculate the maximum normal rear force experienced during a lap at the Isle of Man TT using the results obtained in *Chapter 3.2. Performance Environment* for the rear suspension position under braking.

$$N_R = K'_R \cdot (x_{suspension} \cdot LR)$$

$$N_{R\ mean} = 446\ (N)$$

$$N_{R\ maximum} = 1952\ (N)$$

Where,

- $K'_R = 21.25\ (N/mm)$ (from *Chapter 4.1.1. Suspension Reaction Forces Using Real Data*)
- $x_{suspension\ mean} = 10.49\ (mm)$
- $x_{suspension\ maximum} = 45.92\ (mm)$
- $LR = 2$

The rear braking force will then be:

$$F_{Rear\ Braking} = \mu N_R$$

$$F_{Rear\ Braking\ mean} = 517\ (N)$$

$$F_{Rear\ Braking\ maximum} = 2264\ (N)$$

Finally, for simulation purposes, the calculation of the maximum torque applied to the rear wheel under braking was required.

$$Torque_{Braking\ Rear\ Wheel} = F_{Rear\ Braking\ maximum} \cdot Wheel\ Radius$$

Where

- $Wheel\ Radius = 0.315\ (m)$

$$Torque_{Braking\ Rear\ Wheel} = 713\ (Nm)$$

4.4. CORNERING FORCE

Centrifugal force and gyroscopic effects allow a motorcycle to corner. Being cornering one of the most difficult dynamic situations to define in a racing circuit, the author chose to only analyse the velocities of the motorcycle over maximum values of lean angle.

The Isle of Man TT represents a very particular circuit, where motorcycles achieve very high velocities even during maximum lean angles. Even though a reduction in the maximum velocity with the increased lean angle can be observed, it is impressive how close to their top speed the motorcycles are to the lean angle presented – as it can be observed in the figure below.

```

CORNERING
Velocities (km/h) for banking angles between 50 and 52(1.21%):
- Average: 157.16; - Max: 249.60; - Min: 39.00
Velocities (km/h) for banking angles between 52 and 54(0.90%):
- Average: 160.17; - Max: 247.40; - Min: 61.30
Velocities (km/h) for banking angles between 54 and 56(0.56%):
- Average: 159.38; - Max: 246.80; - Min: 62.00
Velocities (km/h) for banking angles between 56 and 58(0.21%):
- Average: 142.43; - Max: 202.70; - Min: 79.70
Velocities (km/h) for banking angles between 58 and 61(0.13%):
- Average: 142.43; - Max: 207.20; - Min: 80.20

```

Figure 40. Yamaha R6 at IOM TT 2019- cornering analysis

Making use of the velocity and lean angle values acquired from data, it was possible to calculate the forces applied to the motorcycle centre of gravity during a corner.

The free body diagram of a motorcycle in a corner and the formulas used to perform the calculations are attached in the *Chapter 12.4. Cornering Force* of the *Appendix*.

CORNERING FORCES ANALYSIS

Real Angle	Ideal Angle	Corner Radius	Velocity	Force Centrif	Lateral G-Force	Force Normal	Force Resultant
50	43.49	517	249.60	2491.19	0.9	2626	3619.94
52	45.27	477	247.40	2651.69	1.0	2626	3732.21
54	47.07	446	246.80	2823.21	1.1	2626	3855.97
56	48.87	283	202.70	3007.32	1.1	2626	3992.74
58	50.67	277	207.20	3205.90	1.2	2626	4144.37

Figure 41. Yamaha R6 at IOM TT 2019 - cornering forces analysis

For the FE analysis purposes, the values to perform the simulation are the maximum centrifugal and normal force calculated in *Figure 41*– Centrifugal and Normal Force of 3205.9 N and 2626 N respectively.

4.5. SAFETY FACTOR

When designing a component, it is common for engineers to employ a safety factor over the load cases calculated or identified. This provides extra confidence in the results against undetermined and uncertain conditions.

Safety factors are occasionally dictated by legislation, however when that is not the case, distinct methods from different authors could be employed to calculate the safety factor or it could even be defined by design experience.

The Pugsley method was the method chosen by the author to define the safety factor to use in the design phase, as it is the one that is commonly adopted for mechanical engineering projects. As referenced in *Schmid, et al, 2014*, the Pugsley method can be unconservative and might predict low safety factors for real applications, reason why it should only be used as a guideline.

As presented below, the calculations involved in the Pugsley method are simple, but encompass concerns over several parameters that might be out of the engineer/designer control or skill level during the analysis and manufacturing stage. The confidence of the engineer is evaluated regarding the quality of the material, manufacturing operations, the load cases defined, the accuracy of the FE analysis and of the experimental data, as well as the possible danger to personnel and economic impact.

Characteristic ^a		B			
A	C	vg	g	f	p
vg	vg	1.1	1.3	1.5	1.7
	g	1.2	1.45	1.7	1.95
	f	1.3	1.6	1.9	2.2
	p	1.4	1.75	2.1	2.45
g	vg	1.3	1.55	1.8	2.05
	g	1.45	1.75	2.05	2.35
	f	1.6	1.95	2.3	2.65
	p	1.75	2.15	2.55	2.95
f	vg	1.5	1.8	2.1	2.4
	g	1.7	2.05	2.4	2.75
	f	1.9	2.3	2.7	3.1
	p	2.1	2.55	3.0	3.45
p	vg	1.7	2.15	2.4	2.75
	g	1.95	2.35	2.75	3.15
	f	2.2	2.65	3.1	3.55
	p	2.45	2.95	3.45	3.95

^a vg = very good, g = good, f = fair, and p = poor.
A = quality of materials, workmanship, maintenance, and inspection.
B = control over load applied to part.
C = accuracy of stress analysis, experimental data or experience with similar parts.

Characteristic E ^d	D		
	ns	s	vs
ns	1.0	1.2	1.4
s	1.0	1.3	1.5
vs	1.2	1.4	1.6

^d vs = very serious, s = serious, and ns = not serious
D = danger to personnel
E = economic impact

Figure 42. Pugsley method parameters.

Source: *Schmid, et al., 2014*

If, for the purpose of this exercise:

- A = g;
- B = vg;
- C = g;
- D = vs ;
- E = ns;

$$n_{safety} = \frac{\sigma_{allowed}}{\sigma_{design}} = n_{sx(A,B,C)} * n_{sy(D,E)}$$

$$n_{safety} = 1.45 * 1.4 = 2.03$$

5

CONCEPT DESIGN



5. CONCEPT DESIGN

Supported by all the work developed in the previous chapters, it was possible to confidently define the constraints related to the design of the new swingarm, which will assure the safe operation of the swingarm in its performance environment.

The milestone was then to design a new lighter swingarm, a factor that will benefit acceleration and suspension performance due to the reduction in unsprung mass, considering the manufacturing processes available to the author. This target will have to be met while respecting the original swingarm geometry, achieving the same stiffness and natural frequency values as the original swingarm and safely withstanding all the load cases identified and calculated.

Following the constraints and objectives already defined, a market study will be developed over swingarms from motorcycles of the same segment of the one being studied – sportbike - and mainly over prototype racing bikes. This will provide enough information about design features, materials and manufacturing processes, allowing the author to propose an initial design of the new swingarm.

5.1. MARKET STUDY

A market study was conducted over sportbike swingarms, with a focus on prototype racing motorcycles, as the engineering challenges faced to design them were certainly similar to the ones the author has to consider.

As production motorcycles usually resort to casting as the manufacturing process for the swingarm and having in consideration that this manufacturing process was not available to the author, the choice was to centre the analysis on smaller scale racing motorcycles. Also, a high cost in tooling needed for the casting process, which makes it unviable for prototypes, even though it enables to produce swingarms over large quantities while keeping a good balance between performance and final cost.

Several images and specifications were gathered on sportbike swingarms from the study conducted, in order to perform an in-depth analysis to the following details:

- Design features
 - Chain adjustment mechanism;
 - Suspension mount design;
- Overall geometry
 - Shape of the swingarm;
 - Number of parts to build the swingarm;
- Materials
 - Which material;
 - Heat-treatment;
- Manufacturing processes
 - Needs to match the manufacturing process available;
- Connections
 - Type of connection (welds, bolts, etc.);
 - Location of the connections;
 - Specifications (example: bolt diameter, weld width, etc.);

An extract of the analysis conducted is shown below:



Figure 43. Examples of prototype swingarms

Source: DR Moto, Kalex and Suter

The swingarms belonging to the highest performance racing motorcycles in the industry are commonly built out of several machined aluminium billets and then welded together, resulting in a unit type shell piece. This type of construction offers the best stiffness to weight ratio, due to the high second moment of area from the hollowed rectangular sections. Unfortunately, this was not an option for the current project because of the difficulty in CNC machining the side panels that feature small thicknesses and lack of support when machining.

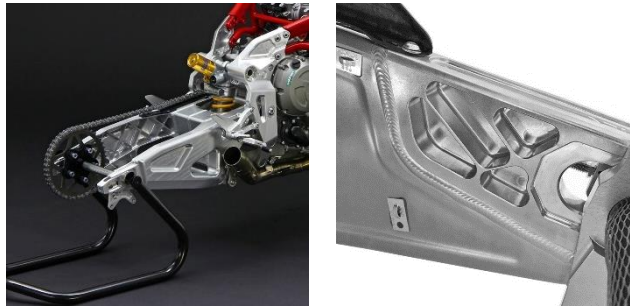


Figure 44. Examples of swingarm inner pockets
Source: Bimoto and Suter

The solution adopted consists in machining pocket sections on the inner side of the arms of the swingarm, ending up with a sort of a “U” shape section instead of a rectangular shape section, thus impacting the swingarm with slight gain in mass (*Figure 44*).

Some solutions regarding the way to design the chain tensioner mechanism, the location of the welds of the rear parts and the way to couple the side arms of the swingarm with its middle body can be observed below.



Figure 45. Examples of swingarm features and details
Source: Suter, Honda and DR Moto

Additionally, carbon and hybrid swingarms (composed by carbon and aluminium) are getting more common at the highest level of motorcycle racing – MotoGP - but due to time constraints related to the complexity of the design, simulation and optimization of a composite structure, it was decided to leave these structures out of the author's analysis.

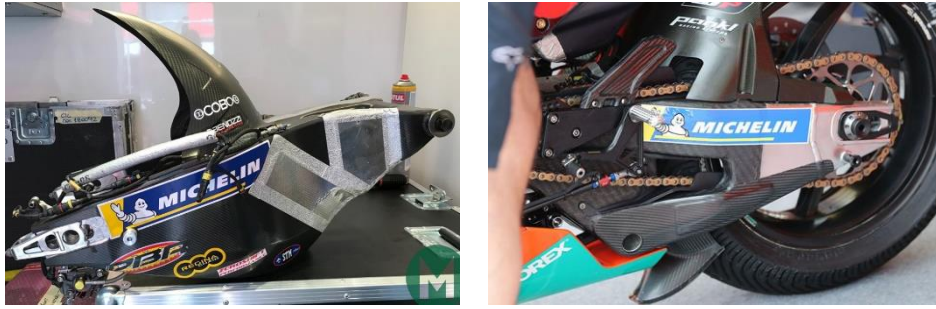


Figure 46. Examples of MotoGP swingarms
Source: Ducati and KTM

5.2. MATERIAL ANALYSIS

From the research conducted, it was concluded that the non-carbon swingarms belonging to the highest performance motorcycles are manufactured out of aluminium. It was then carried out a comparison of the different grades of aluminium, highlighting the grades with the best mechanical properties and welding characteristics.

Table 8. Comparison of different grades of aluminium

Aluminium Grade	Yield Strength (MPa)	Ultimate Strength (Mpa)	Fatigue Strength (MPa)	Young Modulus (GPa)	Poisson Ratio	Shear Strength	Bearing Yield Strength (MPa)	Machinability	Weldability	Filler
6061-O	55.2	124	62.1	68.9	0.33	82.7	103	Good	Good	4043 or 5356
6061-T4, T451	145	241	96.5	68.9	0.33	165		Excellent	Good	4043 or 5356
6061-T6, T651	276	310	96.5	68.9	0.33	207	386	Excellent	Good (YS reduced to 124Mpa and US to 186 Mpa)	4043 or 5356
5083-O	145	290	150	70.3	0.33	172	248	Average	Excellent	5356 or 5183
5083-H116, H321	228	317	159	70.3	0.33	190		Average	Excellent	5356 or 5183

Analysing the table above, the best suited aluminium grade for the manufacturing process of the swingarm is the 5083–H116. It presents the highest ultimate and fatigue strength and the second highest yield strength that grants the material a more ductile behaviour, with a bigger plastic deformation area that will work as a safety feature, as the swingarm will be more likely to bend before cracking.

However, the 5083 – H116 is a marine grade aluminium which is hard to get access to, so the focus of the analysis fell over the 5083 – O and the 6061 – T6 aluminium.

The 5083 – O aluminium presents a superior fatigue strength and the fact that it keeps its mechanical properties after welding as an advantage against the other option. As a drawback, it presents a much lower yield strength and represents a challenge for the machinist because of its softness characteristics.

Even though the 6061 – T6 presents better mechanical properties and better machining characteristics, due to being a heat-treated material, it will lose part of its mechanical properties after welding. This requires extra work during the design and FE analysis stage, in order to optimize the welds location in the lower stress areas (*Corporation, s.d.*).

The author found the best choice to use the 6061 – T6 aluminium because of its almost twice superior yield strength in the overall swingarm and, also, because once the author is responsible for the design and simulation of the swingarm, the location of the welds in lower stress areas will be addressed. Also, the easier machining characteristic of the 6061 – T6 versus the 5083 – O will help reducing the risk related to outsourced work (external factors).

The author defined that, as a safety measure, the maximum stress calculated in the FE analysis should be inferior to the fatigue strength of the chosen material when applying the load cases previously calculated.

Fatigue strength is defined as the stress value, which a material will withstand before suffering a structural damage of any nature after an infinite number of stress cycles (*Stephens, et al., 2001*).

The fatigue performance of a material is frequently characterized by a S-N curve – Fatigue Curve. This curve will provide information about fatigue life (cycles to failure) of a material, in function of the cyclic stress value.

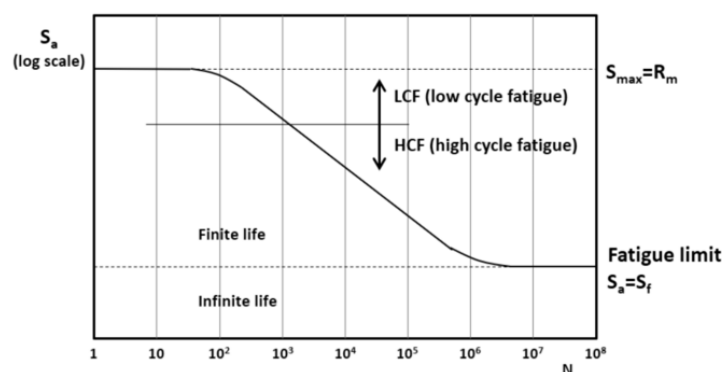


Figure 47. Fatigue curve
Source: *Stephens, 2001*

For the material selected to build the swingarm, 6061 – T6 aluminium, a study was found regarding the investigation of its Fatigue Curve, so it could be used in the construction of a welded nuclear vessel (Yarh, 1993).

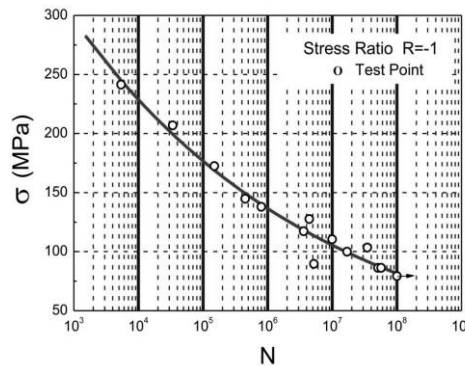


Figure 48. Fatigue curve - 6061 - T6 aluminium
Source: Yarh, 1993

Summing up, the chosen material was the 6061 – T6 aluminium, the swingarm welds will be located in lower stress areas and, during the FE analysis, when submitting the new designed swingarm to the load cases, a maximum stress inferior to the material's fatigue strength – 96.5 MPa – should be expected.

5.3. FIRST DESIGN PROPOSAL

Following the market study conducted on *Chapter 5.1. Market Study*, the author was then prepared to move towards designing the new swingarm.

All the details and specifications resulting from the research conducted to the highest performance racing and prototype motorcycle swingarms were taken in consideration, as well as some additional design constraints, described below:

- The original swingarm geometry had to be kept. This will imply keeping the same location and dimensions of the original pivot point, suspension mounting point and rear wheel area. Thus, the suspension and motorcycle dynamic performance are kept the same, making it possible to use the original swingarm spacers, retainers, pivot bearings, suspension bearings and chain adjusters;



Figure 49. Location of the original pivot point, suspension mounting point and rear wheel area

- The machining process limitations had to be considered in an early phase of the design proposal to avoid further changes in a future and more complex phase of the design. This implies knowing the following parameters:
 - CNC usable working area = 400x300 mm;
 - Number of axis available = 3 axis;
 - Minimum diameter of the milling cutter = 6 mm;
 - Maximum depth the milling cutter can go = 60 mm;



Figure 50. UWTSD's CNC machine
Source: Mazak

Considering the previous design constraints, a first design proposal could be drafted, which would be able to directly replace the original swingarm and keep most of the original components from the swingarm assembly (spacers, bearings, chain adjuster, retainers and suspension linkage), while comprising all the correct considerations to be machinable through the manufacturing processes available to the author. Allowing for these considerations prior to the first design proposal will benefit the author in the following stages of the project, positively limiting the operations that are feasible to be performed in order to optimise the swingarm during the FE analysis.

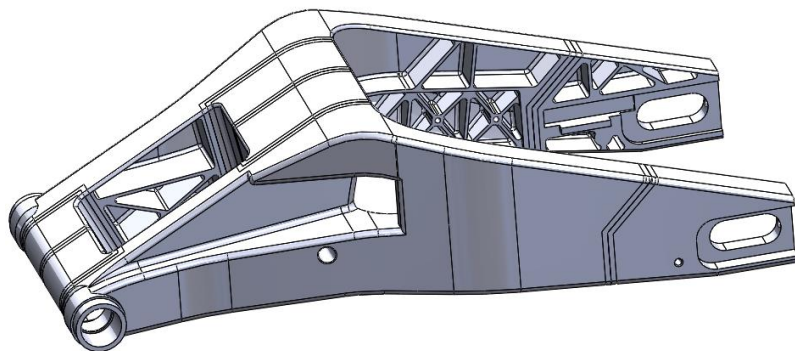


Figure 51. First design proposal

From the first proposal, *Figure 51*, the following details can be highlighted:

- The overall thickness adopted was 3 mm. It could be further tuned to cope with the stresses observed in the swingarm;
- The swingarm was initially planned to be machined in 7 different parts and then welded together. The dimensions of the different parts were defined having the manufacturing limitations in consideration;

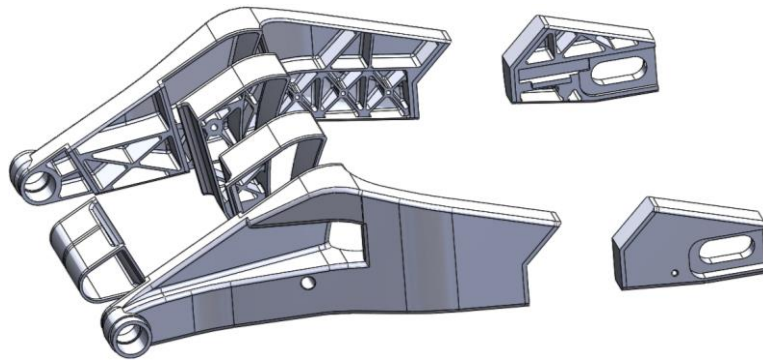


Figure 52. First design proposal - exploded view

- The machined pockets were located on the inner side of the swingarm. This keeps more material on the outer part of the swingarm, consequently increasing the moment of inertia and improving the overall stiffness to weight ratio;
- The final mass of the first design proposal (6 kg) was tuned to be close to the original swingarm mass (6.5 Kg);
- Initially, the author aimed for the new swingarm to offer lateral and torsional stiffness tunability. The threaded holes on the inner side of the swingarm were planned to allow longitudinal connections improving lateral stiffness while the transversal connection in the middle body of the swingarm intended to improve the torsional stiffness. Later in the project, it was proven that the added weight and complexity would not result in significant stiffness changes;

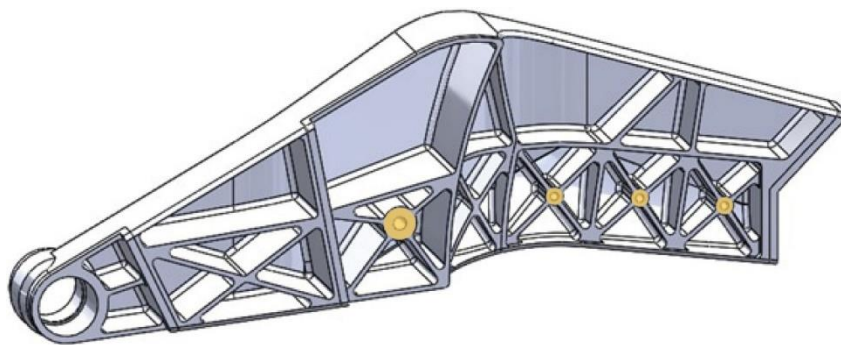


Figure 53. First design proposal - stiffness tuning features

- The middle body of the swingarm was reinforced on the inner side to help the suspension mount cope with the suspension reaction forces. After the FE analysis was performed, it was proven that the reinforcement would have to be relocated to a different position;

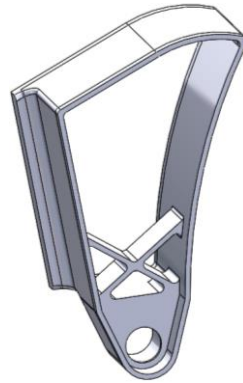


Figure 54. First design proposal - suspension mount detail

6

FINITE ELEMENT ANALYSIS



6. FINITE ELEMENT ANALYSIS

At this stage, as a CAD model of the new swingarm design was already developed and its material defined, all the necessary numerical analyses could be performed in order to optimise the swingarm design until the performance and safety objectives are reached.

Firstly, a strength analysis will be performed to the new swingarm. This analysis will reproduce all the load cases calculated on *Chapter 4. Load Cases* in a FE environment. Once the simulations are performed, the objective will consist in having all the stress values below the fatigue strength of the chosen material. This will ensure that the new swingarm is structurally safe when performing in real critical scenarios at the Isle of Man TT circuit.

Secondly, a stiffness analysis will be carried out to ensure the best dynamic performance of the new swingarm. The new design will then be assessed to its lateral and torsional performance and minor changes will be operated, if needed, in order to validate the results against reference stiffness values.

Lastly, a modal analysis will be performed to the new swingarm and its results can then be compared against the results obtained from the physical test performed to the original swingarm (experiment explained in *Chapter 3.4. Vibration Experiment*). This validation will ensure that the new design is able to avoid instability modes and potential structural damage.

6.1. MESH

When performing Finite Element Analysis, it is crucial to define a sensible mesh in the swingarm in order to get accurate results. Geometries as complex as a swingarm, a mesh represents the division of the swingarm in smaller elements with a known geometry to ease the computing capacity required. The elements are connected with each other through nodes and, depending on the application and objective, the number of elements can be tuned for more accurate results or for a reduction of the calculation time.

To perform a quality mesh in Solidworks Simulation, there are certain parameters that are recommended to have control over, such as:

- Element size is recommended to be 2 to 3 times smaller than the smallest thickness of the part. This will allow to have an element in compression in the smallest thickness of the part, other in the neutral axis and another in extension;

- Aspect Ratio ideally < 10 , preferentially over the critical areas;
- Jacobian < 40 ;
- Smooth stress gradients are recommended. The opposite would indicate substantial changes of stress between elements. Thus, a smooth transition from coarse to fine mesh will result in more accurate results.
- In areas of higher stress values, a finer mesh should be applied. To then validate the accuracy of the results obtained, a mesh convergence should be performed.

Study name	Mesh (-Default-)
Mesh type	Solid Mesh
Mesher Used	Curvature-based mesh
Jacobian points	4 points
Mesh Control	Defined
Max Element Size	20 mm
Min Element Size	5 mm
Mesh quality	High
Total nodes	1477122
Total elements	921234
Maximum Aspect Ratio	1 066.3
Percentage of elements with Aspect Ratio < 3	89.5
Percentage of elements with Aspect Ratio > 10	0.319
% of distorted elements (Jacobian)	0
Remesh failed parts with incompatible mesh	Off
Time to complete mesh(hh:mm:ss)	00:01:19
Computer name	

Figure 55. Mesh details from Solidworks

The mesh specifications that allowed the author to obtain good overall results in the simulations, considering the accuracy of the results and the time to perform them, were the following:

- Type of mesh: Curvature – based mesh;
- Overall minimum size of 1st order element was 5 mm;
- In areas of higher stress, it was used elements of a constant size equal to 0.1 mm;

The mesh specifications above resulted in fine approximation of the real model and were achieved after performing a convergence analysis, resulting in a total of 921234 elements and 1477122 nodes, with a meshing time of 01min19sec.

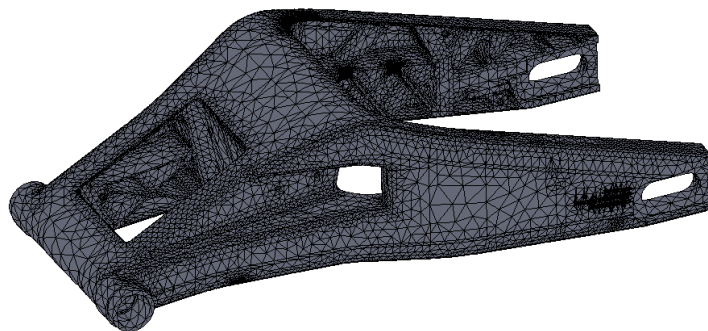


Figure 56. Swingarm mesh preview

6.2. STRENGTH ANALYSIS

6.2.1. SUSPENSION

In the following static simulation, the stress distribution on the new swingarm design when applying the maximum rear normal force and the maximum suspension reaction force will be analysed. This simulation will test the equivalent to the maximum real limit scenario experienced by the suspension in the IOM TT.

The swingarm will be tested in the two possible limit positions for the suspension – with the suspension fully compressed and fully extended – and with the rear wheel positioned in the shortest and longest possible set-up because the results will be affected by the forces location and direction. Only the most critical set-up is presented below.

6.2.1.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
- Position:
 - Swingarm set in the fully compressed position (- 0.9° with horizontal) with rear spacer in the longest set-up;
- Connections:
 - Bonded connection between parts;
- Fixtures:
 - The front bearing surfaces in the pivot point were set as fixed. As the external forces will be in equilibrium, no moments will be generated around the pivot;
- External Loads:
 - Remote Load
 - ***Normal Force_{Rear Wheel}*** = 4876 N applied on the rear wheel contact point with the ground;

- Bearing Load

- $X \text{ Reaction Force}_{\text{Suspension Mount}} = 17344 \text{ N}$ applied on the suspension mount bearing surface;
- $Y \text{ Reaction Force}_{\text{Suspension Mount}} = 10684 \text{ N}$ applied on the suspension mount bearing surface;

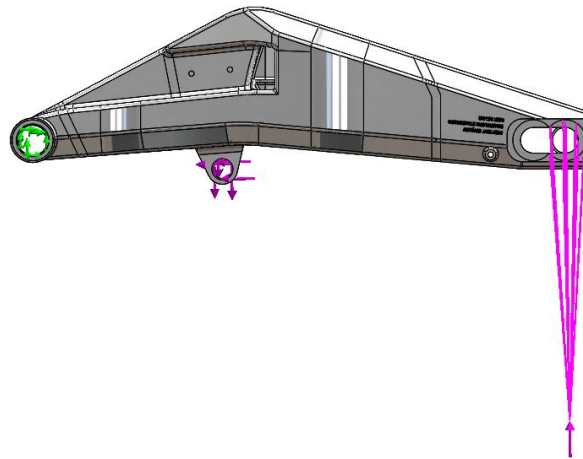


Figure 57. FEA suspension force set-up

6.2.1.2. RESULTS

- Objective:
 - Maximum Stress $\leq 96.5 \text{ MPa}$, which is the fatigue strength of the chosen material;

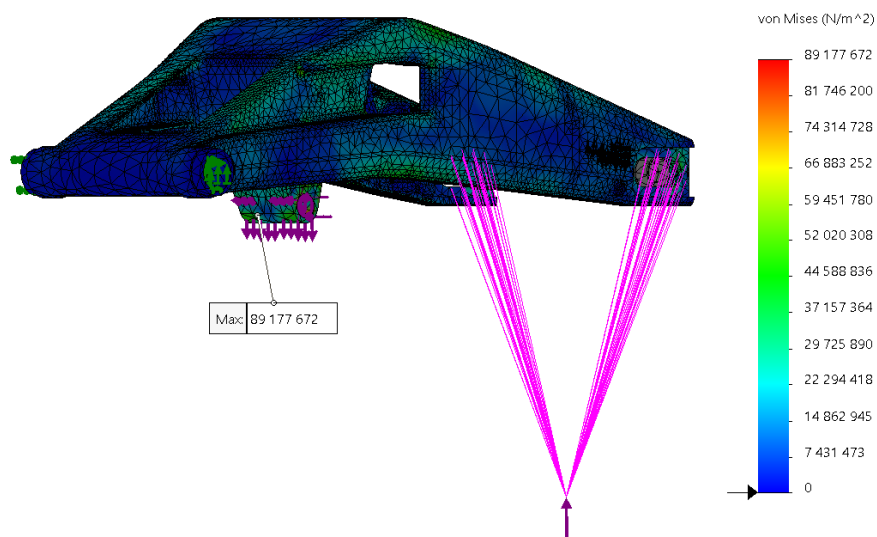


Figure 58. FEA suspension force stress results

As it can be observed from *Figure 58* the objective for this simulation was met, overall stress suffered by the swingarm inferior to the materials' fatigue strength. This result means that the swingarm could be infinitely subjected to the calculated maximum rear normal force, with no structural damage caused by fatigue.

Despite the successful results obtained with all the design changes operated in the swingarm, there are still some concerning stress concentration areas that should be addressed once the swingarm is built.

The level of concern is dependent on the accuracy of the Finite Elements Analysis results versus the results obtained once the real swingarm is built and tested.

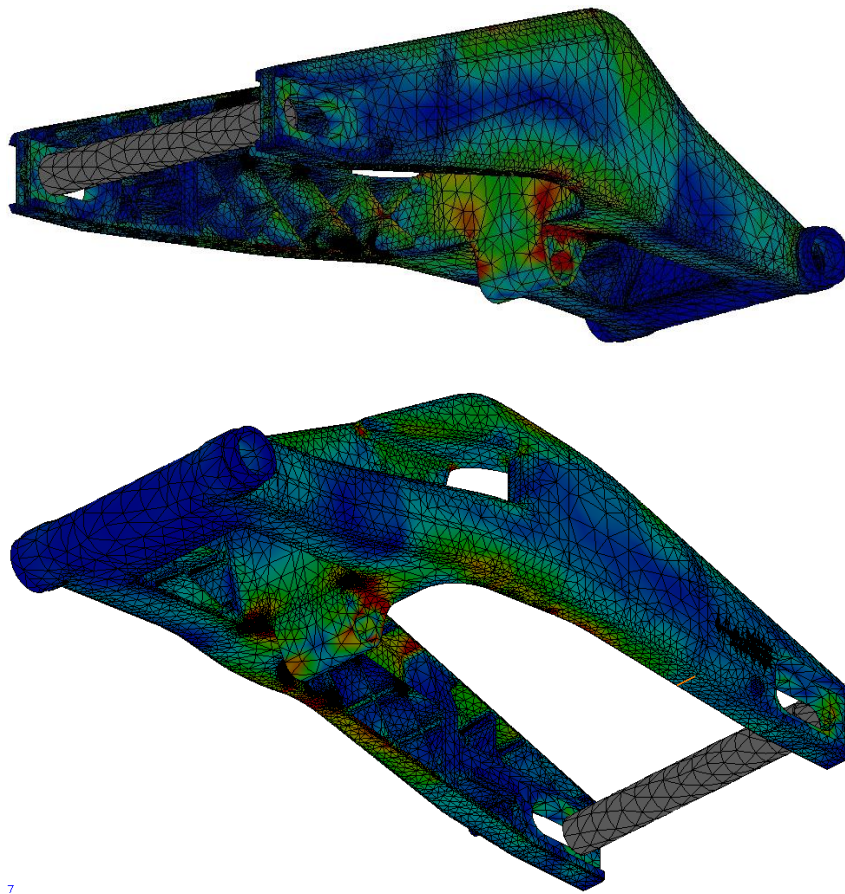


Figure 59. FEA suspension force stress concentration areas
(red highlights stress above 50 MPa)

6.2.2. DRIVING

In the following static simulation, the stress distribution on the new swingarm design when applying the maximum driving force and rear normal force under full acceleration will be analysed. This simulation will test the stress over the swingarm when the rider demands the maximum power output from the engine that will then be transmitted from the drive chain to the rear sprocket.

The swingarm will be tested in the two possible limit positions for the suspension, with the suspension fully compressed and fully extended, and with the rear wheel positioned in the shortest and longest possible set-up because the results will be affected by the forces location and direction. Only the most critical set-up is presented below.

6.2.2.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
- Position:
 - Swingarm set in the fully extended position (- 12.7° with horizontal) with rear spacer in the longest set-up;
- Connections:
 - Bonded connection between parts;
- Fixtures:
 - The front bearing surfaces in the pivot point were set as fixed. As the external forces will be in equilibrium, no moments will be generated around the pivot;
- External Loads:
 - Remote Load
 - ***Driving Force_{Chain}*** = 7952.32 N applied on top of the rear sprocket in the chain direction;
 - ***Normal Force_{Rear Wheel}*** = 2626 N applied on the rear wheel contact point with the ground;
 - Bearing Load
 - ***X Reaction Force_{Suspension Mount}*** = 9859 N applied on the suspension mount bearing surface;
 - ***Y Reaction Force_{Suspension Mount}*** = 3766 N applied on the suspension mount bearing surface;

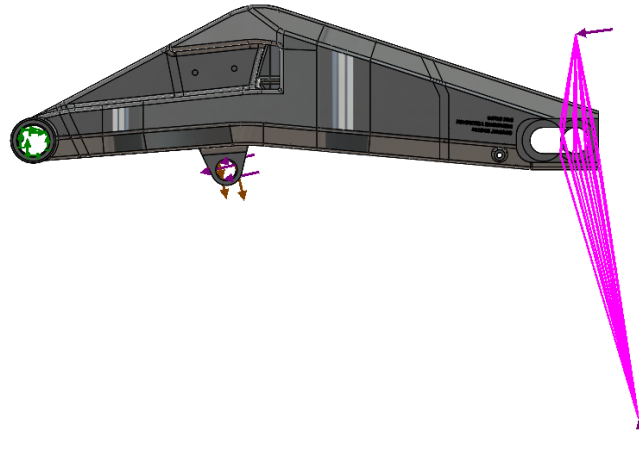


Figure 60. FEA driving force set-up

6.2.2.2. RESULTS

- Objective:
 - Maximum Stress ≤ 96.5 MPa, which is the fatigue strength of the chosen material;

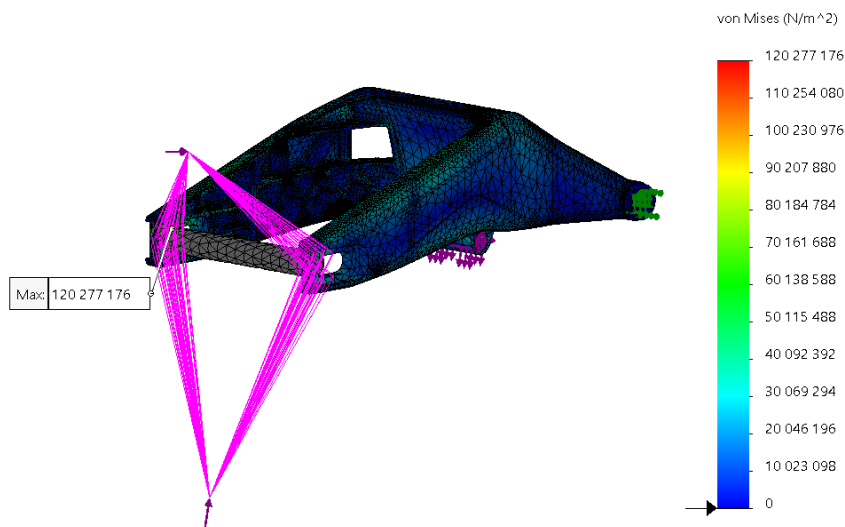


Figure 61. FEA driving force stress results

Even though the values observed in *Figure 61* might indicate that results from the simulation were not successful, a more carefully evaluation of the stress distribution should be performed. The high stress values observed in the rear area of the swingarm could be caused by the driving force and rear normal force being applied directly in the swingarm surface. In a real scenario these forces will have been damped by the rear wheel, rear tyre, rear sprocket, chain and rear rubber dampers before even being transferred to the swingarm.

Besides the already justified higher stress values in the rear area, the overall stress suffered by the swingarm was considerably below the materials' fatigue strength. This result means that the swingarm could be infinitely subjected to the calculated maximum driving force, with no structural damage caused by fatigue.

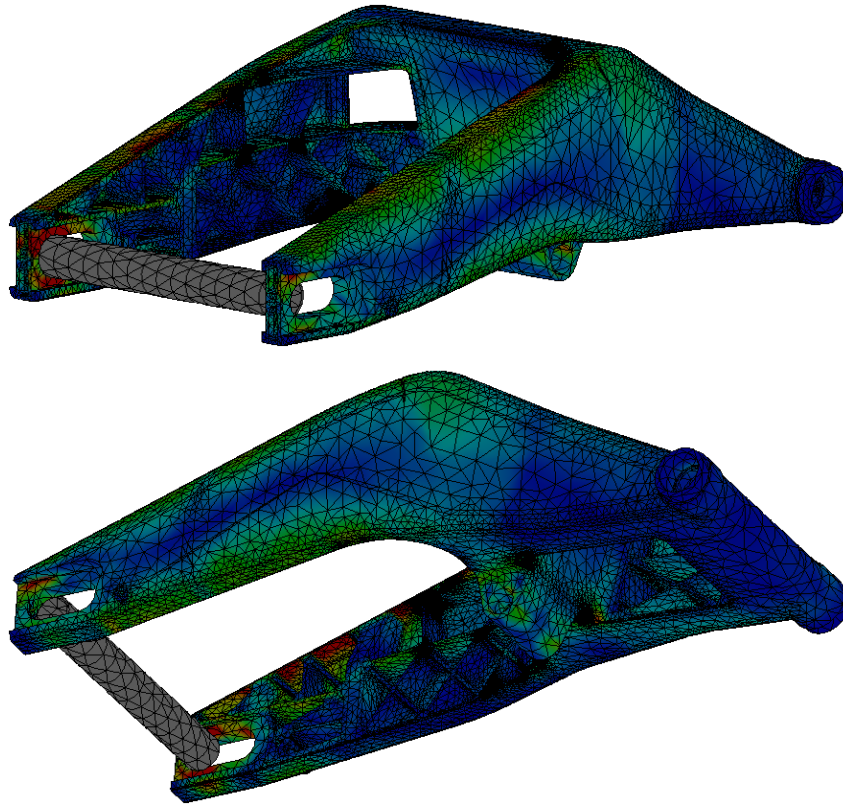


Figure 62. FEA driving force stress concentration areas
(red highlights stress above 50 MPa)

6.2.3. BRAKING

In the following static simulation, the stress distribution on the new swingarm design when applying the maximum rear braking force will be analysed. This simulation will test the equivalent to the maximum rear braking scenario experienced by the motorcycle in the IOM TT.

The braking torque calculated in *Chapter 4.3. Braking Force* was applied on the rear caliper support to investigate the stress distribution where this support seats on the swingarm. Once again, the most critical set up was tested and analysed, with the rear wheel positioned in the longest set-up available.

6.2.3.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
 - Rear caliper support set as a rigid part;
- Position:
 - Swingarm set in the fully extended position (-12.7° with horizontal) with rear spacer in the longest set-up;
- Connections:
 - Bonded connection between parts;
- Fixtures:
 - Front swingarm pivot set as a fixed hinge;
 - Suspension mount bearing surfaces set as a fixed hinge;
- External Loads:
 - Remote Load
 - $Torque_{Braking\ Rear\ Wheel} = 713\ Nm$ applied in the rear calliper support, more precisely in the bolts' sections where the rear caliper mounts;

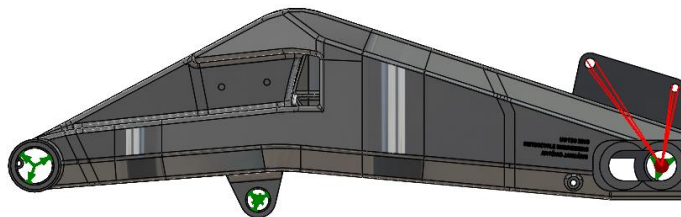


Figure 63. FEA braking force set-up

6.2.3.2. RESULTS

- Objective:
 - Maximum Stress $\leq 96.5\ MPa$, which is the fatigue strength of the chosen material;

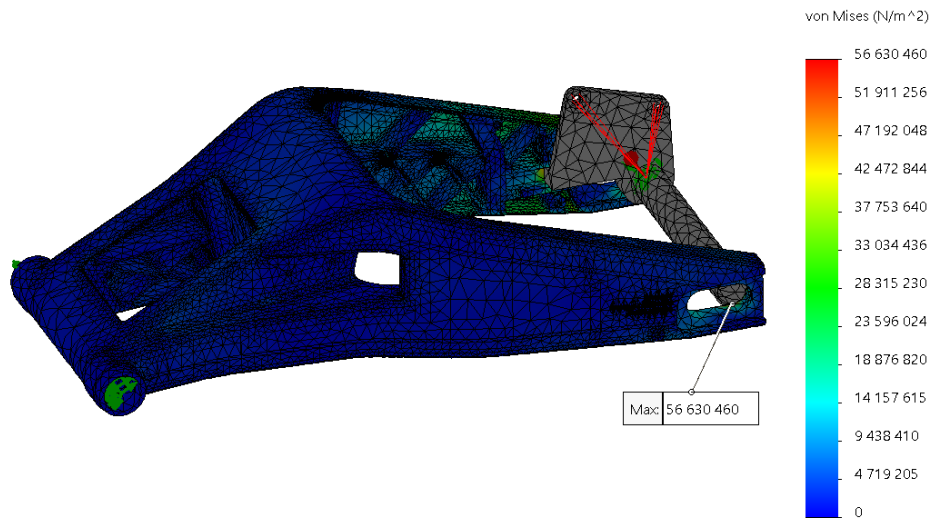


Figure 64. FEA braking force stress results

As it can be observed from *Figure 64*, the objective for this simulation was met – overall stress suffered by the swingarm during the braking simulation was inferior to the materials' fatigue strength. This result means that the calculated maximum rear braking torque could be infinitely subjected to the swingarm with no structural damage caused by fatigue.

This was an expected result, once the features in the new swingarm design that support all the braking forces were surveyed from the original swingarm from the manufacturer. Transferring these features not only assures the correct fit of the rear caliper support in the swingarm but also that the structural and fatigue considerations from the manufacturer were kept.

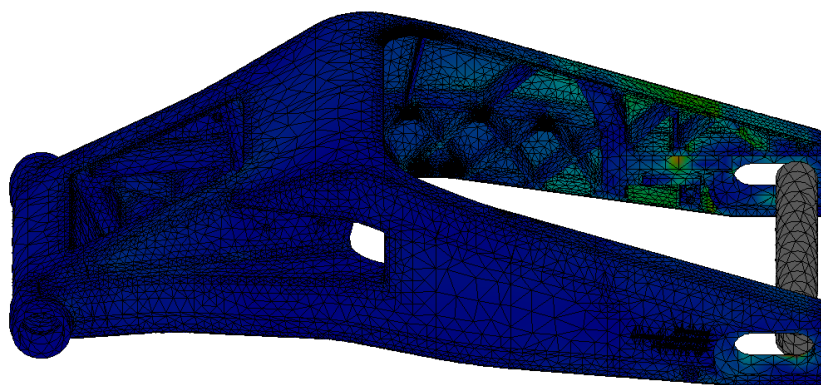


Figure 65. FEA braking force stress concentration areas
(red highlights stress above 50 MPa)

6.2.4. CORNERING

In the following static simulation, the stress distribution on the new swingarm design when applying the maximum cornering forces – Centrifugal force and Normal force – will be analysed. This simulation will test the equivalent to the maximum real limit scenario experienced by the swingarm during a corner at the IOM TT. This racing circuit is very particular when compared to GP circuits, because velocities over 200 kph can be achieved at the motorcycle's maximum lean angles.

The swingarm will be tested in the fully compressed and extended position, and with the rear wheel positioned in the shortest and longest possible set-up, because the results will be affected by the location and direction of the forces. Also, due to the slight asymmetric geometry of the swingarm, it is relevant to simulate corners to the right, as well as to the left. The most critical set up is presented below.

6.2.4.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
- Position:
 - Swingarm set in the fully extended position (-12.7° with horizontal) with rear spacer in the longest set-up;
- Connections:
 - Bonded connection between parts;
- Fixtures:
 - Front swingarm pivot set as a fixed hinge;
 - Suspension mount bearing surfaces set as a fixed hinge;
- External Loads:
 - Remote Load
 - ***Normal Force*** $_{Rear\ Wheel} = 2626N$ applied on the rear wheel contact point with the ground;
 - ***Centrifugal Force*** $_{Rear\ Wheel} = 3206N$ applied on the rear wheel contact point with the ground;

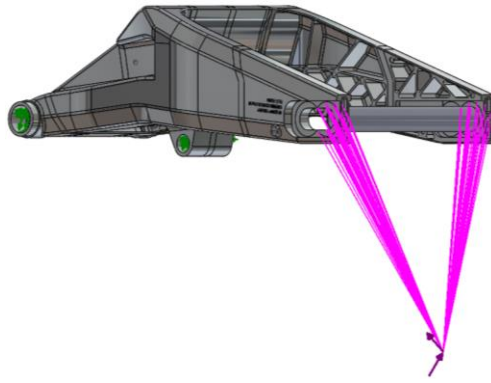


Figure 66. FEA cornering forces set-up

6.2.4.2. RESULTS

- Objective:
 - Maximum Stress ≤ 96.5 MPa, which is the fatigue strength of the chosen material;

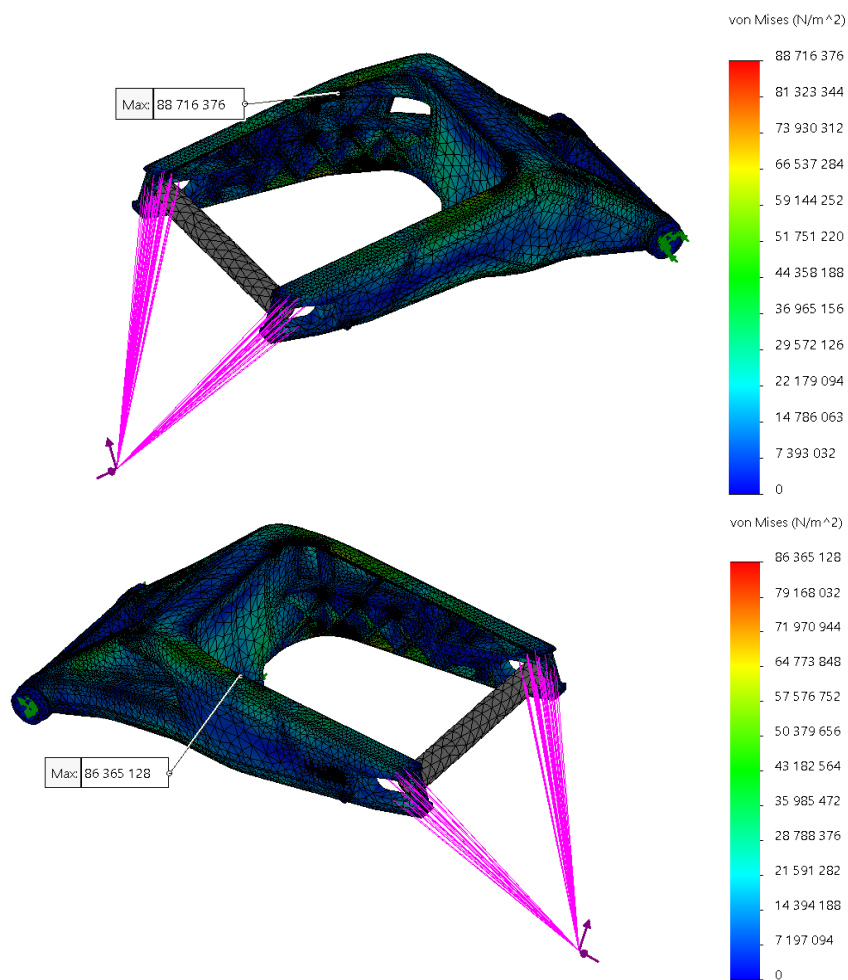


Figure 67. FEA cornering forces stress results (right and left cornering)

As it can be observed from the figures above, the objective for this simulation was met – overall stress suffered by the swingarm when cornering to the right or left was inferior to the materials' fatigue strength. This result means that the swingarm could be infinitely subjected to the calculated maximum cornering forces in *Chapter 4.4. Cornering Force*, with no structural damage caused by fatigue.

Despite the successful results obtained with all the design changes operated in the swingarm, there are still some concerning stress concentration areas that should be addressed once the swingarm is built.

The level of concern is dependent on the accuracy of the Finite Elements Analysis results versus the results obtained once the real swingarm is built and tested.

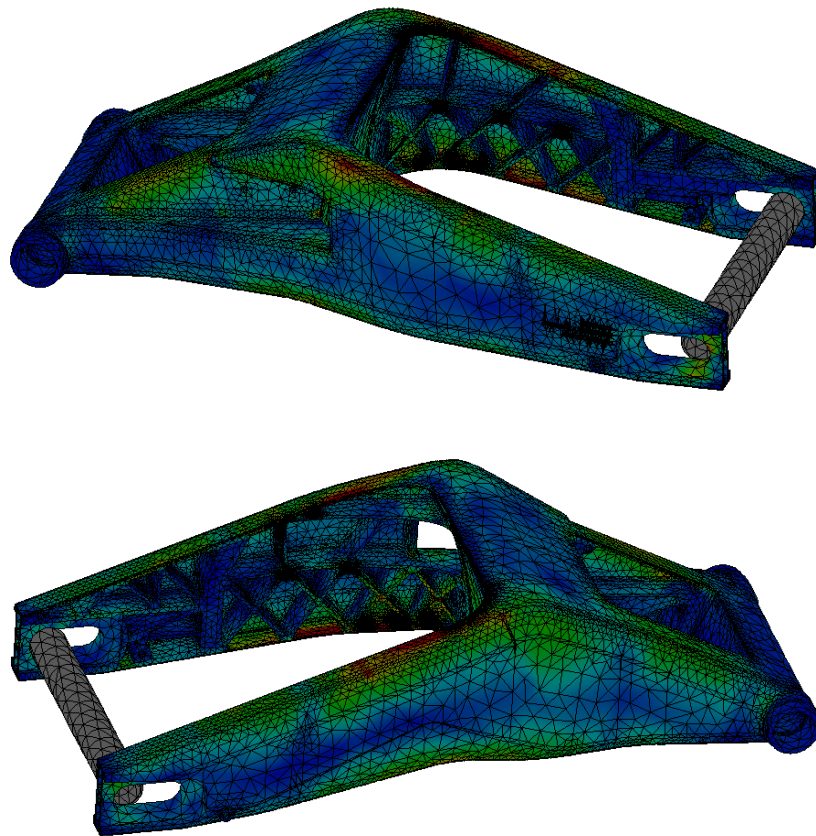


Figure 68. FEA cornering forces stress concentration areas
(red highlights stress above 50 MPa)

6.3. STIFFNESS ANALYSIS

6.3.1. LATERAL STIFFNESS

In the following static simulation, the displacement changes of the swingarm when applying a certain constant lateral force will be analysed. This simulation will attest the lateral stiffness values of the new swingarm design.

Complementarily to what was analysed in the previous chapter, *6.2. Strength Analysis*, where the safety concerns of the new swingarm were addressed – the current chapter will analyse the swingarm characteristics that would improve its stability, response and dynamic performance.

The swingarm will then be tested for the rear wheel positioned in the shortest and longest possible set-ups, in order to obtain the range of values where the lateral stiffness could vary.

6.3.1.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
- Position:
 - Swingarm set up with rear spacer in the shortest and longest positions;
- Connections:
 - Bonded connection between parts;
- Fixtures:
 - Front swingarm pivot set as a fixed;
- External Loads:
 - Force
 - ***Lateral Force*** = 1000 N applied on the outer surface of the rear area of the swingarm;

- Lateral Stiffness Calculations:
 - $\delta = (DY1 + DY2 + DY3)/3$
 - $Lateral\ Bending = \frac{F}{\delta} [N/mm]$

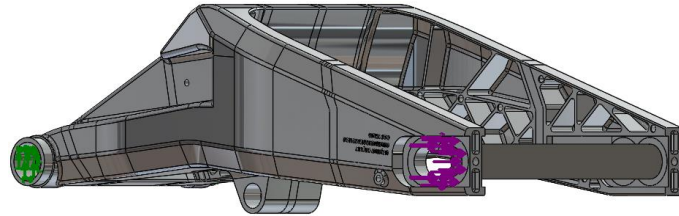


Figure 69. FEA lateral stiffness set-up

6.3.1.2. RESULTS

- Objective:
 - $800\ N/mm \leq Lateral\ Stiffness \leq 1600\ N/mm$;

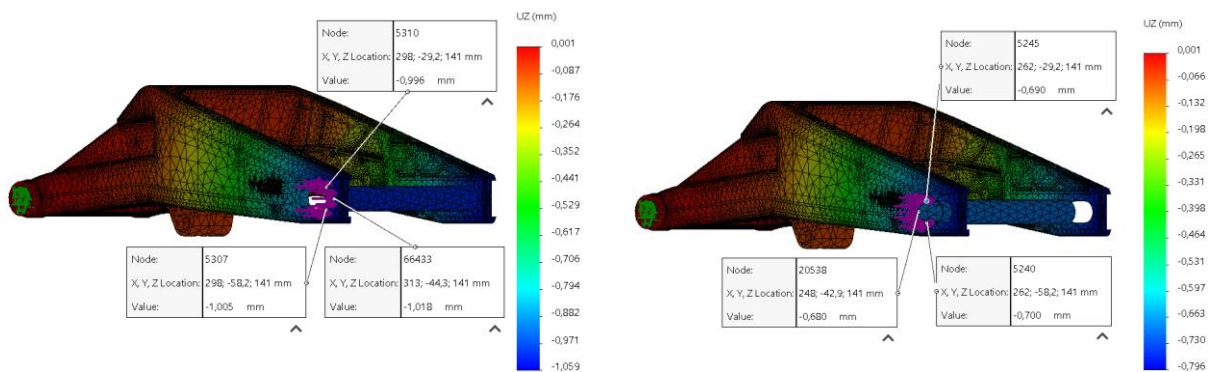


Figure 70. FEA lateral stiffness displacement results

Table 9. Lateral stiffness results

	Lateral Stiffness Longest Set Up	Lateral Stiffness Longest Set Up	Units
DY1	0.996	0.690	mm
DY2	1.018	0.680	mm
DY3	1.005	0.700	mm
δ	1.006	0.690	mm
Lateral Stiffness	994	1449	N/mm
Vittore Cossalter's References	800-1600	800-1600	N/mm

To perform the lateral stiffness simulations, the same set-up as the one used in Vittore Cossalter's book was followed, reason why similar results to the reference ones were expected.

As it was observed from the results in *Table 9*, the lateral stiffness objectives were met in the final swingarm design. Complementarily to the successful results, the author was also able to develop an in-depth knowledge over the design features that affect the lateral stiffness values: the longer the distance between the swingarm pivot point and rear wheel, the lower the lateral stiffness value while the stronger the inner section that connects the swingarm arms to the middle body, the bigger the lateral stiffness value.

6.3.2. TORSIONAL STIFFNESS

In the following static simulation, the angle of twist of the swingarm when applying a certain constant torque to the rear area of the swingarm will be analysed. This simulation will attest the torsional stiffness values of the new swingarm design.

Complementarily to what was analysed in the previous chapter, *6.2. Strength Analysis*, where the safety concerns of the new swingarm were addressed, the current chapter will analyse the swingarm characteristics that would improve its stability, response and dynamic performance.

Torsional stiffness is responsible to maintain the wheel alignment and geometry of the motorcycle, when twisting moments are applied on the swingarm. These twisting moments occur when the load cases are applied with an offset in relation to the centre of the swingarm (chain drive, brake disc, etc.)

The swingarm will then be tested for the rear wheel positioned in the shortest and longest possible set-ups, in order to obtain the range of values where the torsional stiffness could vary.

6.3.2.1. METHODOLOGY

- Parts:
 - Final swingarm design;
 - Rear spacer set as a rigid part;
- Position:
 - Swingarm set-up with rear spacer in the shortest and longest positions;

- Connections:
 - Bonded connection between parts;
- Fixtures:
 - Front swingarm pivot set as a fixed;
- External Loads:
 - Remote Load
 - *Twisting Moment* = 500 Nm applied on the outer surface of the rear area of the swingarm;
- Torsional Stiffness Calculations:
 - $\theta = \tan^{-1} \frac{DY1-DY2}{\text{distance between } DY}$
 - *Torsional Bending* = $(F * L)/(\theta)$ [Nm/deg]

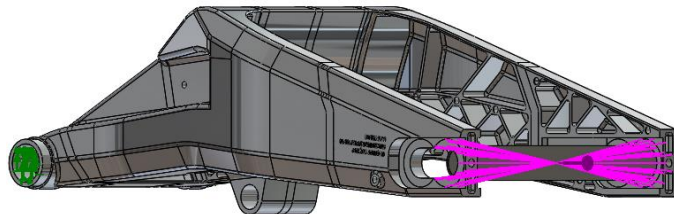


Figure 71. FEA torsional stiffness set-up

6.3.2.2. RESULTS

- Objective:
 - $1000 \text{ Nm/}^\circ \leq \text{Torsional Stiffness} \leq 2000 \text{ Nm/}^\circ$;

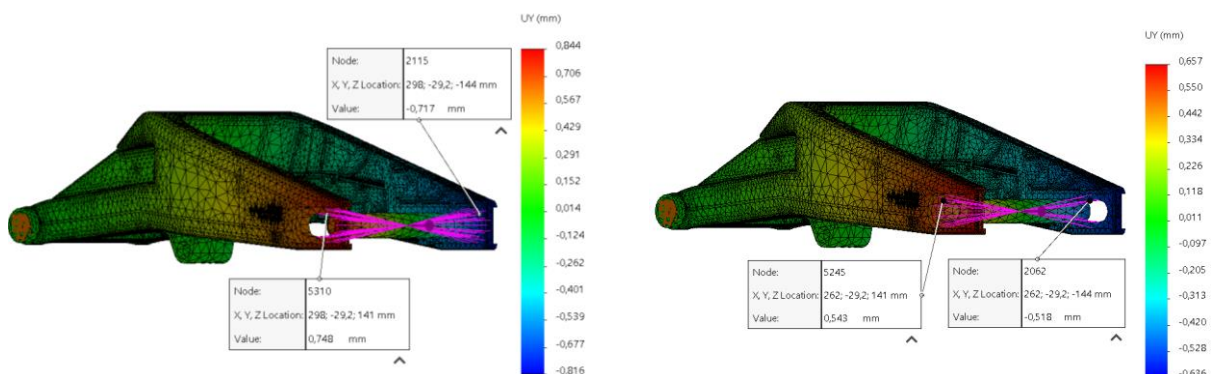


Figure 72. FEA torsional stiffness displacement results

Table 10. Torsional stiffness results

	Lateral Stiffness – Longest Set Up	Lateral Stiffness – Longest Set Up	Units
DY1	0.748	0.543	mm
DY2	-0.717	-0.518	mm
<i>Distance between DY</i>	284.5	284.5	mm
θ	0.295	0.214	°
Torsional Stiffness	1695	2340	Nm/°
Vittore Cossalter's References	1000-2000	1000-2000	Nm/°

To perform the torsional stiffness simulations, the same set-up as the one used in Vittore Cossalter's book was followed, reason why it was expected similar results to the reference ones.

As it was observed from the results in *Table 10*, the torsional stiffness objectives were partially met, as the stiffness range in the final swingarm design was slightly superior to the range of values assumed as reference.

Complementarily to the satisfactory results, the author was also able to develop an in-depth knowledge over the design features that affect the torsional stiffness values: the torsional stiffness value is related to the thickness or quantity of material in the middle body of the swingarm and in the outer regions of the swingarm arms.

6.4. MODAL ANALYSIS

In the following frequency simulation, the natural frequencies and the models of vibration of the new swingarm design will be analysed and identified. The modes of vibration will correspond to the deformed shape of the new swingarm under these frequencies.

A modal analysis uses the overall stiffness and mass of a structural part to define the frequency values in which the swingarm will naturally vibrate. Thus, to keep the same performance and safety concerns of the original swingarm, the results from this modal analysis will be compared against the results from the LDV experiment performed by the author in *Chapter 3.4. Vibration Experiment*.

6.4.1. METHODOLOGY

- Parts:
 - Final swingarm design;
- Simulation Properties:
 - Number of frequencies = 10;

6.4.2. RESULTS

- Objective:
 - Natural frequencies from simulation should match the frequency values obtained in the LDV experiment performed to the original swingarm and should not equal the frequencies expected from the motorcycle in its performance environment (from engine rpm, road irregularities or from other motorcycle parts), so as to avoid instability modes and structural damage.

Table 11. Comparison of natural frequencies

Frequency Modes (Hz)	Superbike 2008	Ducati 899	LDV – side arms	LDV – body	Final Design
1 st	115	197.46	115.625	112.5	135.07
2 nd	261	378.59	462.5	315.625	246.81
3 rd	304	480.40	940.625	468.75	429.06
4 th	579	797.18			744.33

When comparing the results from the FE modal analysis, performed to the newly designed swingarm, with the results from the LDV experiment, performed to the original swingarm, a slightly superior frequency was observed in the 1st mode of the new swingarm design and an extra mode of frequency was identified between the 1st and 2nd modes observed in the LDV experiment.

The fact that the swingarm was supported with strings in the LDV experiment should be taken in consideration since that may have dampened some of the vibration, resulting in the lower frequencies, when compared to the FE modal analysis where the swingarm was assumed in free space. These different details in the set-up of the test experiment versus the simulation will inevitably result in a solution with a percentage of error.

Table 12. Analysis of Yamaha R6 frequencies

Frequency Modes	Final Design (Hz)	Engine Speed (RPM)	Quickest suspension stroke (Hz)
1 st	135.07	8100	50
2 nd	246.81	14808	
3 rd	429.06		
4 th	744.33		

The engine speed equivalent to the 1st and 2nd frequency modes of the new swingarm design corresponds exactly to both ends of the Yamaha R6 powerband. As defined in the objectives of the current chapter, if the engine speed of the motorcycle in its performance environment is able to equal the natural frequencies of the swingarm, it could result in instability modes and structural damage to the part. However, the frequencies outputted from the engine will be filtered by several other components before reaching the swingarm, which makes it difficult for the author to perform a risk assessment based on the results observed on *Table 12*.

Nevertheless, the results achieved were positive, with the natural frequency values of the new swingarm design being similar to the values of the original one. Also, as it can be observed in *Figure 73*, the modes of vibration of the new swingarm design were similar to the ones extracted from the LDV experiment, which confidently indicates a similar overall stiffness between the two swingarms.

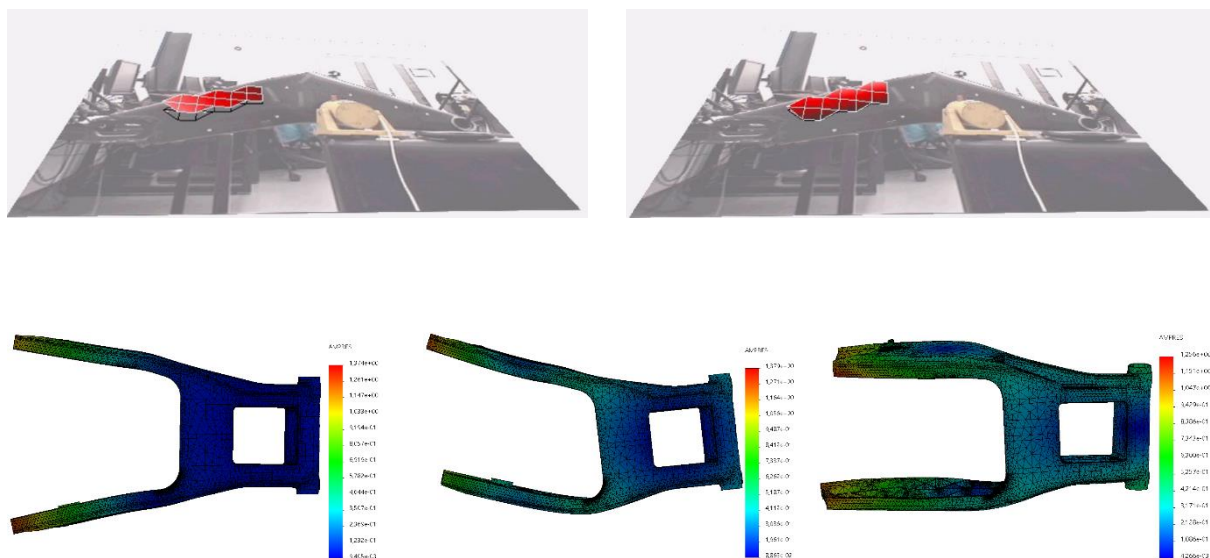


Figure 73. Comparison of modes of vibration - LDV experiments vs. FE modal analysis (LDV experiment - 1st and 2nd modes; FE modal analysis - 1st, 2nd and 3rd modes)

7

FINAL DESIGN



7. FINAL DESIGN

7.1. OVERVIEW

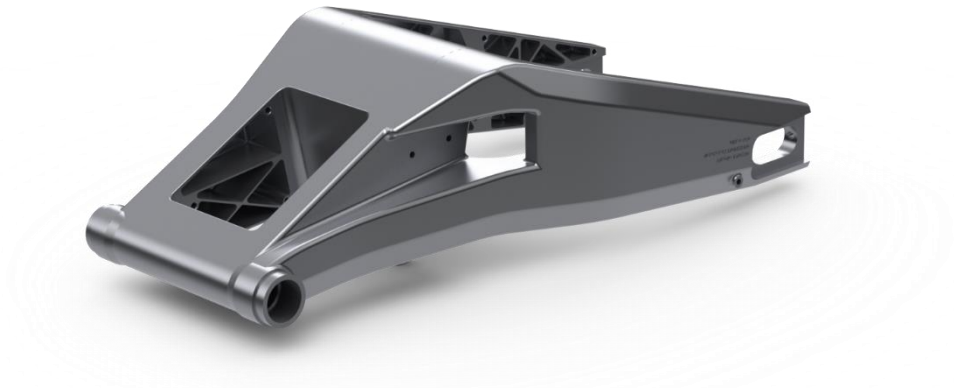


Figure 74. Final design proposal of the swingarm

After performing the Finite Element Analysis to the new swingarm design and a series of optimisation iterations in the design features of the swingarm, the strength, stiffness and vibration objectives were met while respecting all the initially defined constraints.

A final design was achieved with a mass of 6.05 Kg, an improvement when compared to the mass of the original swingarm, which was of 6.50 Kg. The table below showcases a summary of the new swingarm characteristics:

Table 13. Characteristics of final swingarm design

FINAL SWINGARM DESIGN		Units
Mass	6.05	Kg
Material	Aluminium 6061–T6	
Welding Filler	4043 or 5356	
Maximum Stress	96.5	MPa
Lateral Stiffness	994–1449	N/mm
Torsional Stiffness	1695–2340	Nm/°
Natural Frequency	1 st mode 135.07	Hz
	2 nd mode 246.81	
	3 rd mode 429.06	

7.2. MANUFACTURING CONCERNS



Figure 75. Final design proposal of the swingarm - exploded view

As it can be observed in *Figure 75*, the new design swingarm is composed of 9 different parts of CNC machined aluminium. These different parts will be connected by welds, which were strategically positioned in lower stress areas identified in *Chapter 6. Finite Element Analysis*, having the machining process limitations in considerations, as exposed in *Chapter 5.3. First Design Proposal*.

Considering that the CNC available to manufacture the author's swingarm operates in 3 axis, the different pieces of the swingarm were designed to be machined through the following steps:

- The inner surface of each one of the swingarm parts will be the first side to be machined. This will be mainly because it is where the details and pockets are located, allowing to cut deep into the billet of aluminium with a strong fixation over the volume of material;
- Threaded holes were designed into the first surface to be machined. This feature will enable to fix the part into a jig, providing a stable support when machining the outer surface of the swingarm parts. It was a crucial feature, as it will allow to machine thinner wall sections, avoiding unwanted vibrations, as well as keeping a superior quality outer finishing;



Figure 76. Final design proposal - detail of the inner pockets



Figure 77. Final design proposal of the swingarm - rear view

All the single parts will be machined from different aluminium volumes, resulting in a total of 73.4 Kg of material needed. The final machined swingarm will correspond to approximately 8% of total amount of material and the machining waste and remainder of material will be recycled, theoretically resulting in zero waste.

Detailed figures and technical drawings of each of the swingarm components can be found in *Chapter 12.5. Final Design of the Appendix*.

8

PANDEMIC IMPACT FURTHER WORK



8. PANDEMIC IMPACT – FURTHER WORK

The author was affected by the pandemic in a critical stage of the project. Unfortunately, it impacted the project in a phase with a large percentage of work to be developed by external labour, meaning that manufacturing the swingarm would no longer be viable, as the university workshop was closed.

A great effort has been made by the author to have a final swingarm design completely ready to be manufactured in the first week of February 2020. The proposed planning – detailed in *Chapter 1.4. Timing Plan* – would provide the workshop technicians with approximately 3 months of work to develop the necessary code to CNC machine the swingarm jigs, to machine prototypes to validate the code and to actually machine the swingarm parts designed by the author.

It was also planned that, once the workshop technicians had finished building the swingarm parts, the swingarm would have been sent to a professional welder to be finished (all the details were discussed and agreed with the professional welder chosen to take part in the project). Still in the welder's workshop, the author would have had access to a good quality stiffness jig, where both the swingarm designed by the author and the original swingarm could have been experimentally tested for their lateral and torsional stiffness.

This experimental stiffness test performed on the finished swingarm designed by the author would have been crucial in the outcome of this project, as it would have allowed to validate the results from the Finite Element Analysis against an experimental test. This validation would result in a percentage of error over the software package used when performing FE analysis over complex components, such as the swingarm. Also, by experimentally testing both swingarms, the author would have been able to compare the stiffness values of the new design with the original swingarm, which would have allowed to relate the rider's feedback and performance with the swingarms' stiffness differences.

9

CONCLUSIONS



9. CONCLUSIONS

The current project allowed the author to develop an in-depth understanding on how a motorcycle swingarm should perform in order to offer the best possible performance to the rider whilst retaining all the safety concerns.

A preliminary study was developed to identify all the maximum load cases acting upon the swingarm. These various limiting scenarios were identified and calculated by resorting to real data acquired at the circuit of the Isle of Man TT 2019 with the same motorcycle for which the new swingarm is intended for.

Once the first concept design of the new swingarm was established, numerical models were developed to simulate all the calculated load cases. After several iterations, successful results were achieved when the maximum stress observed was inferior to the material's fatigue strength. This safety measure imposed by the author means that the identified maximum load cases represent less than 36% of the Yield strength of the material and that, under cyclical maximum loads, the swingarm will be able to operate infinitely.

Once attested that the swingarm will safely perform on the circuit, other numerical models were developed to analyse the lateral and torsional stiffness of the swingarm. The results were once more successful, with the lateral stiffness values varying from 994 N/mm to 1440 N/mm and the torsional stiffness from 1695 Nm/° to 2340 Nm/°, depending on the rear wheel distance to the swingarm pivot point. These values are within the surveyed reference values and are responsible for the dynamic performance of the swingarm, assuring the best traction, comfort and feedback to the rider under acceleration, braking and cornering situations.

Additionally, the original swingarm was subjected to a vibration experiment, in order to identify the frequency values in which the swingarm will naturally vibrate and its modes of vibration. The same experiment was simulated on the new swingarm design by performing a FE modal analysis, which resulted in natural frequencies values with a 17% error difference. The results were considered successful and the error was assumed to be related to the mass difference between the swingarms and the difficulty in seamlessly replicating the experiment conditions in the FE model.

During the optimisation process of the new swingarm, the author was able to respect all the strength, stiffness and vibration objectives, the manufacturing constraints when operating changes in the design and still achieve a new swingarm design with 6.05 Kg, approximately 8% lighter than the original one.

Despite the know-how acquired by the author over the design and simulation process of an engineering problem that will enable better estimations, quicker progress and reduce costs, physical testing will still be essential to validate the FE models developed in order to further improve the component.

In conclusion, this project allowed the author to develop a fully comprehensive knowledge over the design process of a structural component of a motorcycle, achieving a final design of a racing prototype swingarm ready to be manufactured, unfortunately seeing it impacted by the COVID-19 pandemic in the last stage before physical completion and testing. However, the successful process followed by the author could be adopted in the future as a guideline for the design of other major structural chassis components.

10

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10. REFERENCES

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11

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11. BIBLIOGRAPHY

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12

APPENDIX



12. APPENDIX

12.1. TIMING PLAN

Table 14. Gantt chart with detailed timing plan

MAJOR PROJECT GANTT CHART																																					
TASK		1st Term																		2nd Term																	
		September					October					November				December				January				February				March				April					
		2	9	16	23	30	7	14	21	28	4	11	18	25	2	9	16	23	30	6	13	20	27	3	10	17	24	2	9	16	23	30	6	13	20	27	
Load Cases	Suspension Forces																																				
	Driving Forces																																				
	Braking Forces																																				
	Cornering Forces																																				
Stiffness	Research																																				
Fatigue	Materials																																				
	Sn Curves																																				
Vibration	Research																																				
	Experiment																																				
New Design	CAD Work																																				
FEA	Simulations																																				
Report																																					
1st Term Evaluation	Viva Presentation																																				
Manufacturing	Design to CNC																																				
	CNC Machining																																				
	Welding																																				
Dissertation	Final Document																																				
2nd Term Evaluation	Report																																				
	Viva Presentation																																				
COURSE ASSIGNMENTS																																					

12.2. SUSPENSION REACTION FORCES USING REAL DATA

Resorting to the data acquisition software, 2D Data Recording, a table with the minimum and maximum values for the rear suspension position (mm), velocity (mm/s) and acceleration (mm/s²) was plotted.

Min/max table (1=0827-PSM_APO-01) Suspension Vel Acc.mms

Lap	Time	Meters	R_Sus			Susp_R_Vel			Sus_R_Acc	
			MIN	MAX	AVG	MIN	MAX		MIN	MAX
1	19:50.095 min	61916	1.29	53.16	22.07	-876.0	1248.0		-234400.03	217200.00
2	20:01.110 min	61865	0.64	53.11	21.03	-972.0	1506.0		-301200.00	301200.00
3	19:32.455 min	61916	1.02	54.81	22.24	-724.0	984.0		-202799.91	176400.00
4	19:49.880 min	61916	1.34	54.36	21.34	-706.0	1220.0		-203599.94	244000.09
MIN			0.64	53.11	21.03	-972.0	984.0		-301200.00	176400.00
MAX			1.34	54.81	22.24	-706.0	1506.0		-202799.91	301200.00
AVG			1.07	53.86	21.67	-819.5	1239.5		-235599.97	234700.02

Channels ☐ Alarm Use phases ☒ #Ch: Phases

0827-PSM_APO

Figure 78. Rear suspension data analysis extracted from 2D Data Recording

From the analysis of the table above, several conclusions can be outlined regarding the different suspension parameters. In relation to the rear suspension position, it can be observed that its full travel is being used, reassuring that it was well dimensioned and set up for the purpose.

The suspension velocity and acceleration values result from a calculated channel that uses the suspension position and time as inputs. Some errors in these velocity and acceleration values were expected, once the frequency used for the acquisition of the suspension position data was low for the purpose (200 Hz).

Not only for the reason stated above, but also because the suspension settings were deliberately set in a soft configuration for the Isle of Man TT, the values for suspension velocity and acceleration resulted to be much superior when comparing to values observed in a GP circuit: the velocity and acceleration values were 3 and 10 times bigger, respectively, to what was observed in a GP circuit.

To further investigate the error propagation and the veracity of the maximum values observed in the previously shown table, the velocity and acceleration values higher than 800 mm/s and 60000 mm/s² respectively, were highlighted over each one of the 4 laps recorded from data.

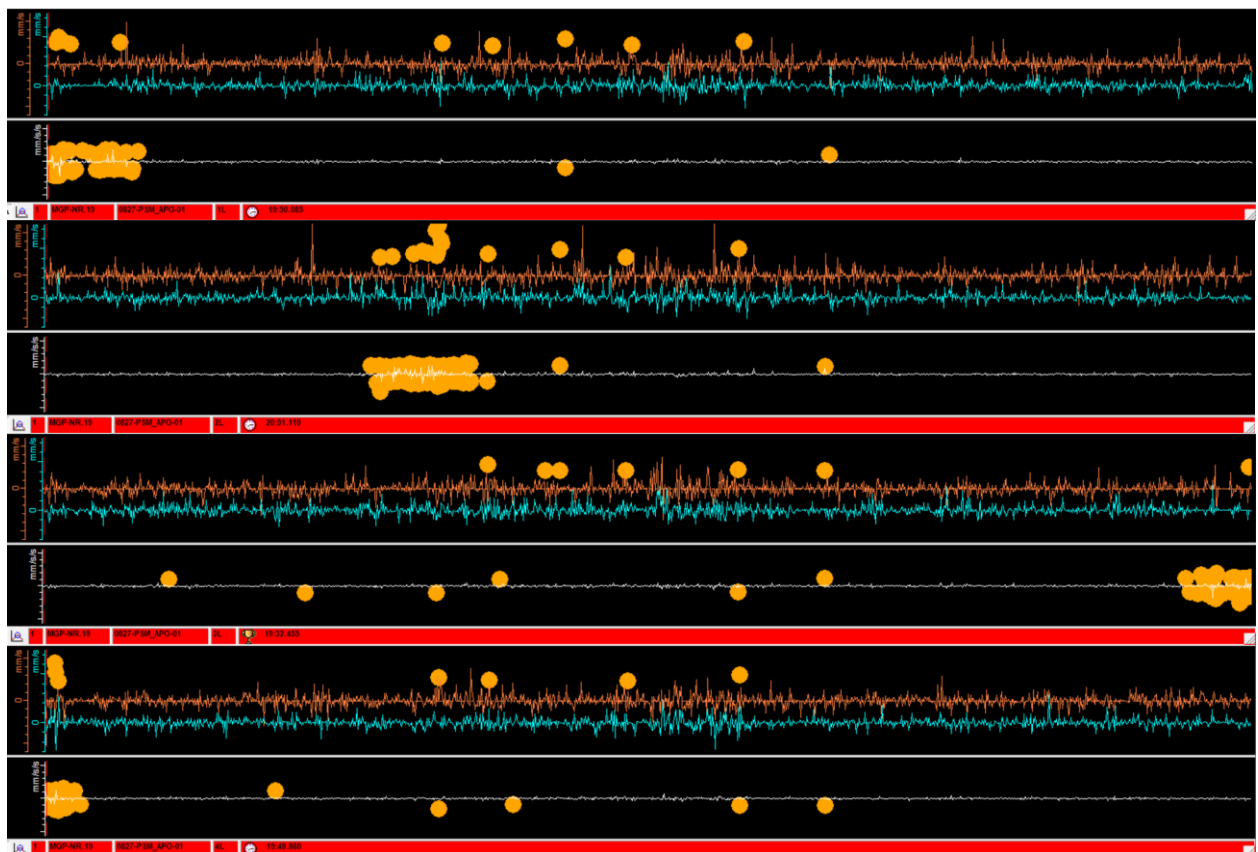


Figure 79. Rear suspension velocity and acceleration plot data extracted from 2D Data Recording

No pattern between laps could be observed through the analysis of the plots above (velocity values in the first half of each plot and acceleration in the second half). The higher values occurred in small percentages during each lap and at different sections of the lap. This proves the lack of accuracy of the values obtained, which resulted from the low frequency used to record the suspension position in the data acquisition system.

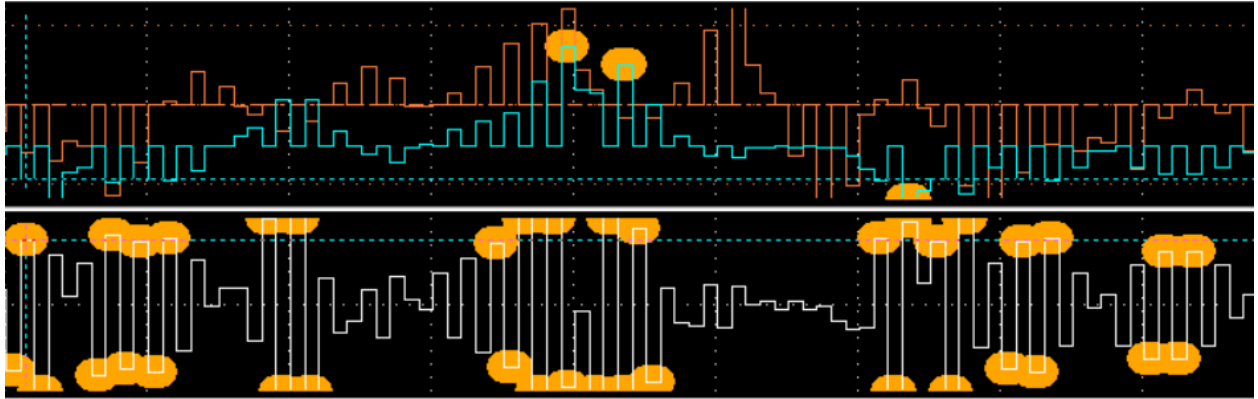


Figure 80. Rear suspension maximum velocity and acceleration highlighted data

The figure above is a zoom-in of a fraction of a lap where the situations where the velocity and acceleration values are maximum could be observed. Again, the insufficient frequency in the collection of suspension position data led to highly stepped velocity results, consequently compromising and having a negative impact on the calculated acceleration values.

12.3. DRIVING FORCE

In *Chapter 4.2. Driving Force*, the results from an acceleration simulation of the current motorcycle, from 60 kph to its maximum velocity, were presented.

To develop the simulation, the motorcycle acceleration was studied over three different limiting situations: wheelie, traction and power limited.

The free body diagram of a motorcycle is shown below, in order to analyse the wheelie and traction limited acceleration, followed by an explanation of the relevant functions of the Matlab code that drives the simulation.

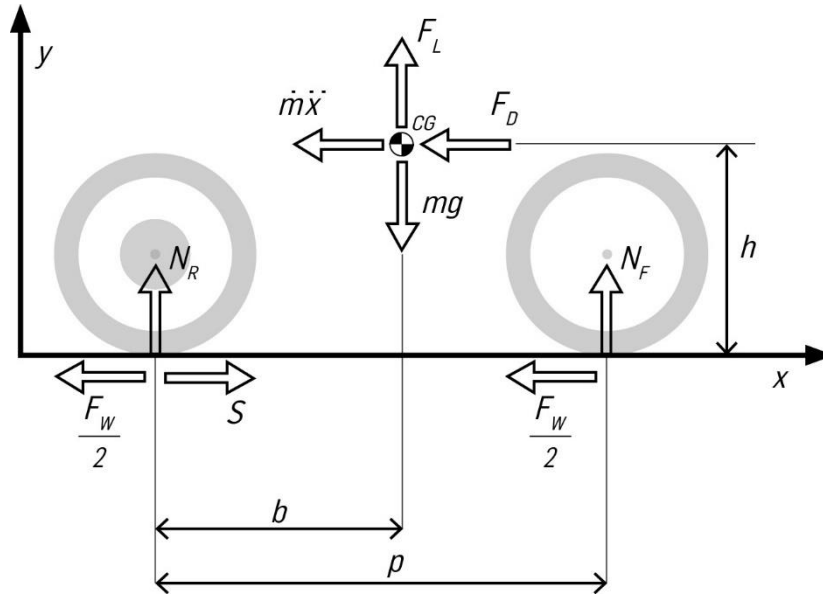


Figure 81. Free body diagram of motorcycle in rectilinear motion

- Horizontally

$$(1) \quad S = F_w + F_D + \dot{m} \cdot \ddot{x}$$

- Vertically

$$(2) \quad N_F = mg - F_L - N_R$$

- Moments around CG

$$(3) \quad N_R = S \cdot \frac{h}{P} + mg \cdot \frac{P-b}{P} + F_L \cdot \frac{-P-b}{P} + F_w \cdot \frac{h}{P}$$

Wheelie Limited Acceleration:

- When $N_F = 0$,

$$(4) \quad N_R = mg - F_L$$

$$(4) = (3)$$

- And by replacing S in (3) by (1):

$$\ddot{x} = \frac{m}{\dot{m}} g \frac{b}{h} - \frac{F_D}{\dot{m}} - \frac{F_L}{\dot{m}} \left(\frac{b}{h} \right)$$

Traction Limited Acceleration:

- When $S \leq \mu N_R$,

$$\mu N_R = (1)$$

$$\mu N_R = F_w + F_D + \dot{m} \cdot \ddot{x}$$

- By substituting N_R by (3):

$$\ddot{x} = \frac{\mu \left(\frac{m}{\dot{m}} g \left(\frac{P-b}{h} \right) + F_L \left(\frac{-P+b}{h} \right) - F_W \left(\frac{h}{P} \right) \right)}{\left(1 - \mu \left(\frac{h}{P} \right) \right)} - \frac{F_D}{\dot{m}} - \frac{F_W}{\dot{m}}$$

As for the Matlab code, the author chose to only represent the most relevant function, *AccelerationYamahaR6.m*. This function receives the motorcycle specifications, geometry values and velocity at the current motorcycle conditions as an input and calculates the motorcycle geometry and velocity in the following meter, according to the maximum possible acceleration - whether wheelie, traction or power limited.

```

AccelerationYamahaR6.m
1 function [gear, rpml, Fs_power, Fs_wheelie, Fs_traction, Fs, Fd, Fl, acc_power, acc_
2
3 %External Forces
4 Fw = 0.02*(Nf0+Nr0);
5 Fd = 0.5*ro*Cd*A*(v0)^2;
6 Fl = 0.5*ro*Cl*A*(v0)^2;
7 RWp = RWr*2*pi;
8 %Calculation of Driving Force and Acceleration with engine power curve
9 [rpm_gear, S] = deal(zeros(1, length(final_gear)));
10 for j = 1:length(final_gear)
11     rpm_gear(j) = v0*final_gear(j)*60/RWp;
12     S(j) = interp1(RPM, W, rpm_gear(j))/v0;
13 end
14 %Select gear where it has maximum driving force at v0
15 Fs0 = 0;
16 [Fs, gear] = max(S);
17 Fs_power = Fs;
18 if isnan(Fs) == 1
19     Fs = Fs0;
20     gear = gear0;
21     rpml = rpm_gear(gear);
22     acc_power = 0;
23     what_a = "acc_power";
24 else

```

```

25 -     rpml = rpm_gear(gear);
26 -     acc_power = (Fs - Fd - Fw)/mx(gear);
27 - end
28 -     %Max acceleration when Nf = 0;
29 -     acc_wheelie = (m*g*b/h - Fl*b/h - Fd)/mx(gear);
30 -     Fs_wheelie = Fw + Fd + mx(gear)*acc_wheelie;
31 -     %Max acceleration limited by traction;
32 -     acc_traction = ((u*(m*g*(p-b)/p + Fl*(-p+b)/p - Fw*h/p))/(1-u*h/p)-Fw-Fd)/mx(gear);
33 -     Fs_traction = Fw + Fd + mx(gear)*acc_traction;
34 -     %Acceleration used
35 -     [acc,n] = min([acc_power,acc_wheelie,acc_traction]);
36 -     if acc < 0
37 -         acc = 0;
38 -     end
39 -
40 -     switch n
41 -     case 1
42 -         what_a = "acc_power";
43 -     case 2
44 -         what_a = "acc_wheelie";
45 -         Fs = Fw + Fd + mx(gear)*acc_wheelie;
46 -
47 -     case 3
48 -         what_a = "acc_traction";
49 -         Fs = Fw + Fd + mx(gear)*acc_traction;
50 -     end
51 -     %Equation of motion to find next velocity value
52 -     v1 = sqrt((v0^2)+2*acc);
53 -     time = 1/v1;
54 -     %Normal forces
55 -     Nr1 = Fs*h/p + m*g*(p-b)/p + Fl*(-p+b)/p - Fw*h/p;
56 -     if Nr1 < 0
57 -         Nr1 = 0;
58 -     end
59 -     Nf1 = m*g - Nr1 - Fl;
60 -     if Nf1 < 0
61 -         Nf1 = 0;
62 -     end
63 - end

```

Figure 82. Matlab code for acceleration function

12.4. CORNERING FORCE

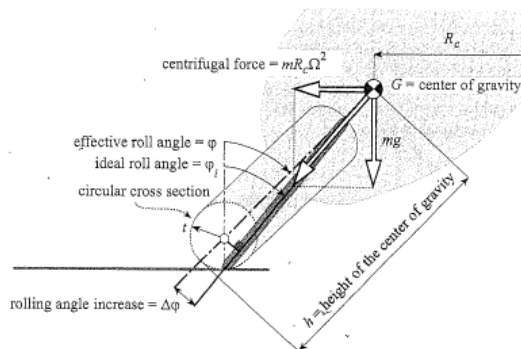


Figure 83. Free body diagram - cornering with tyre correction

Tyre thickness correction:

$$\frac{\sin(\Delta\varphi)}{t} = \frac{\sin \varphi_{ideal}}{(h - t)}$$

- Where,
 - t = average front and rear tyre width / 2;
 - h = center of mass height;

$$\varphi_{ideal} = \sin^{-1} \left(\frac{\sin \varphi_{effective}}{\frac{t}{h - t} + 1} \right)$$

Cornering Forces:

- Centrifugal Force:

$$F_{Lateral} = \frac{mv^2}{r}$$

- Normal Force:

$$Normal_{force} = mg$$

- Ideal Lean Angle:

$$\tan \sigma_{ideal} = \frac{m \cdot v^2}{r \cdot mg}$$

$$(1) \varphi_{ideal} = \tan^{-1} \left(\frac{v^2}{rg} \right)$$

- Maximum Corner Speed:

$$m \frac{v^2}{r} = \mu mg$$

$$(2) v_{Max} = \sqrt{r\mu g}$$

By replacing (2) in equation (1):

$$\mu = \tan(\varphi_{ideal})$$

12.5. FINAL DESIGN

12.5.1. OVERVIEW

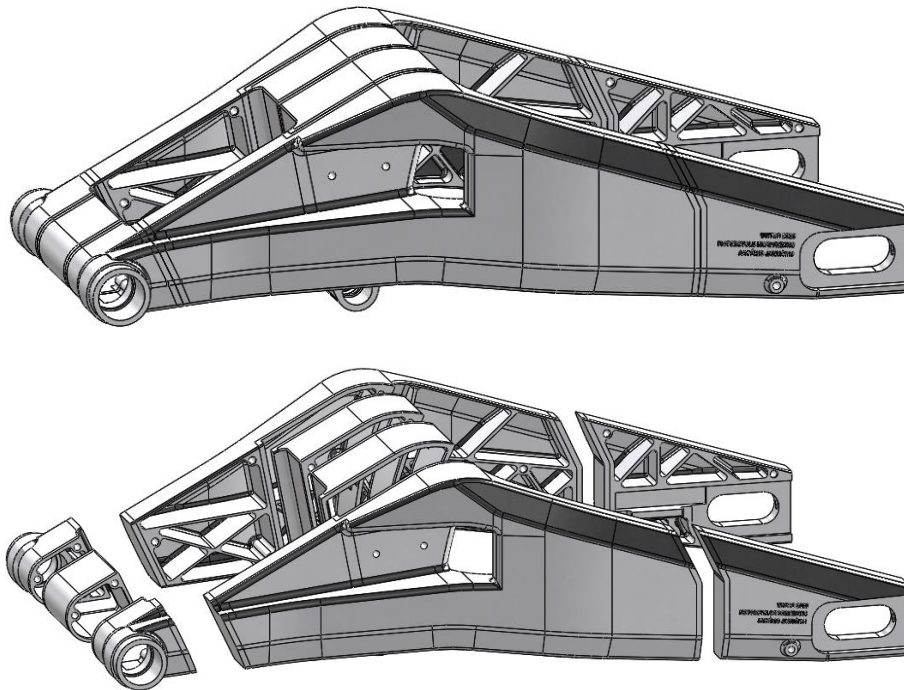


Figure 84. Final design - left view

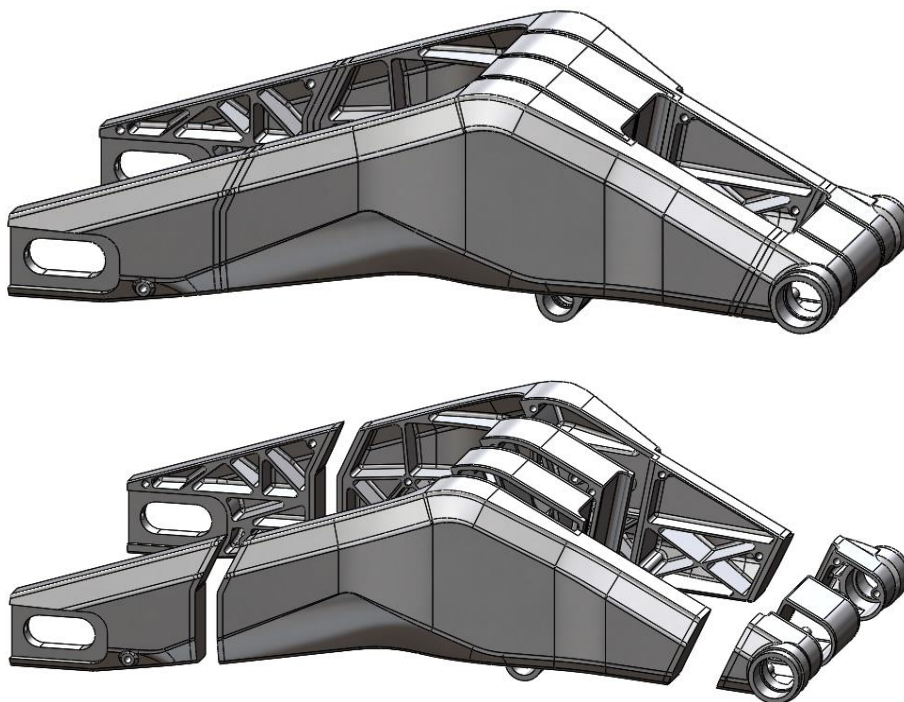


Figure 85. Final design - right view

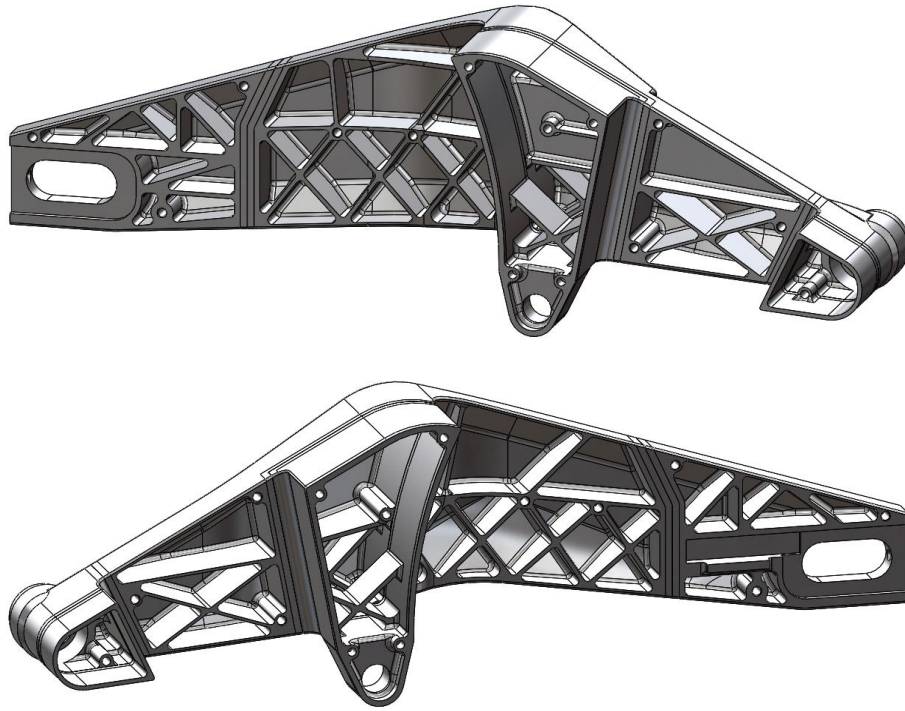


Figure 86. Final design - inner section views

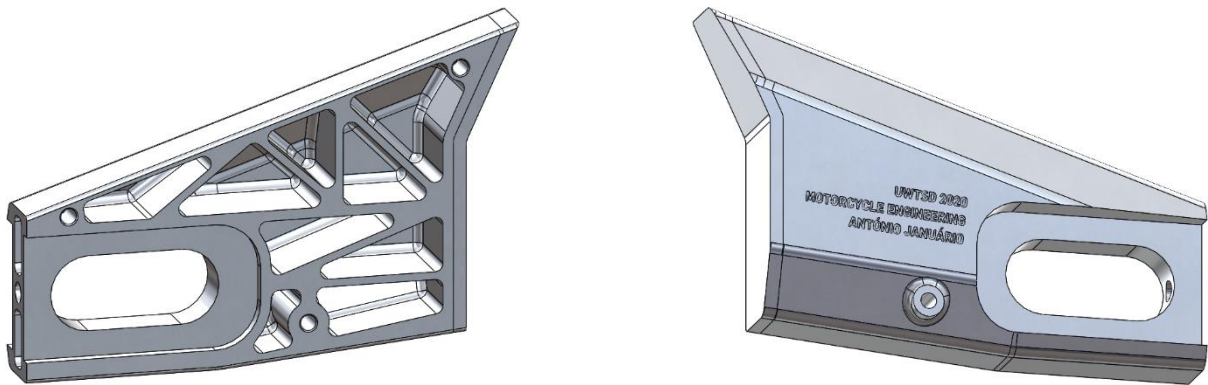


Figure 87. Final design - left rear part

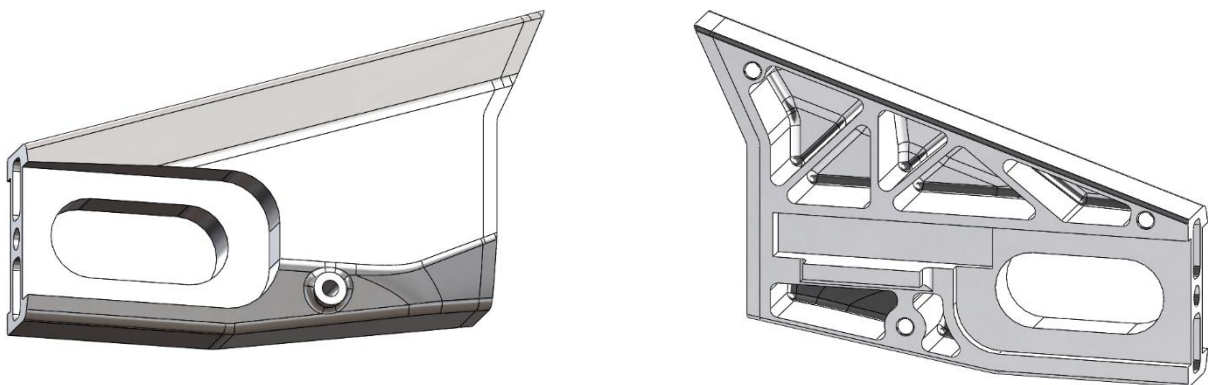


Figure 88. Final design - right rear part

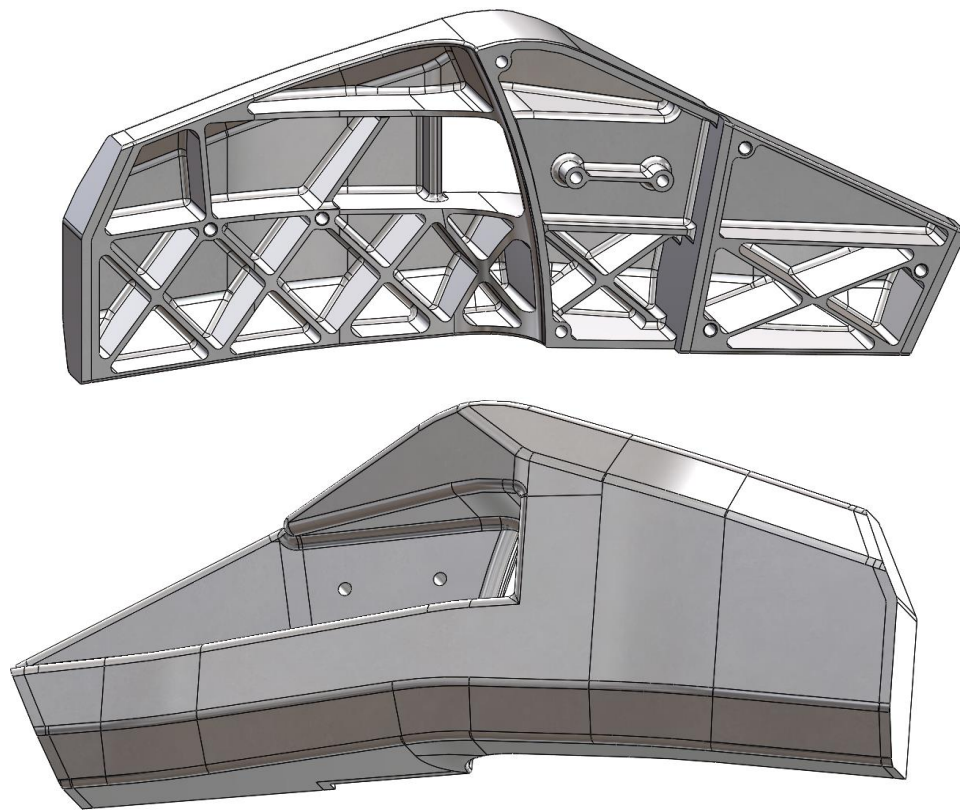


Figure 89. Final design - Left middle part

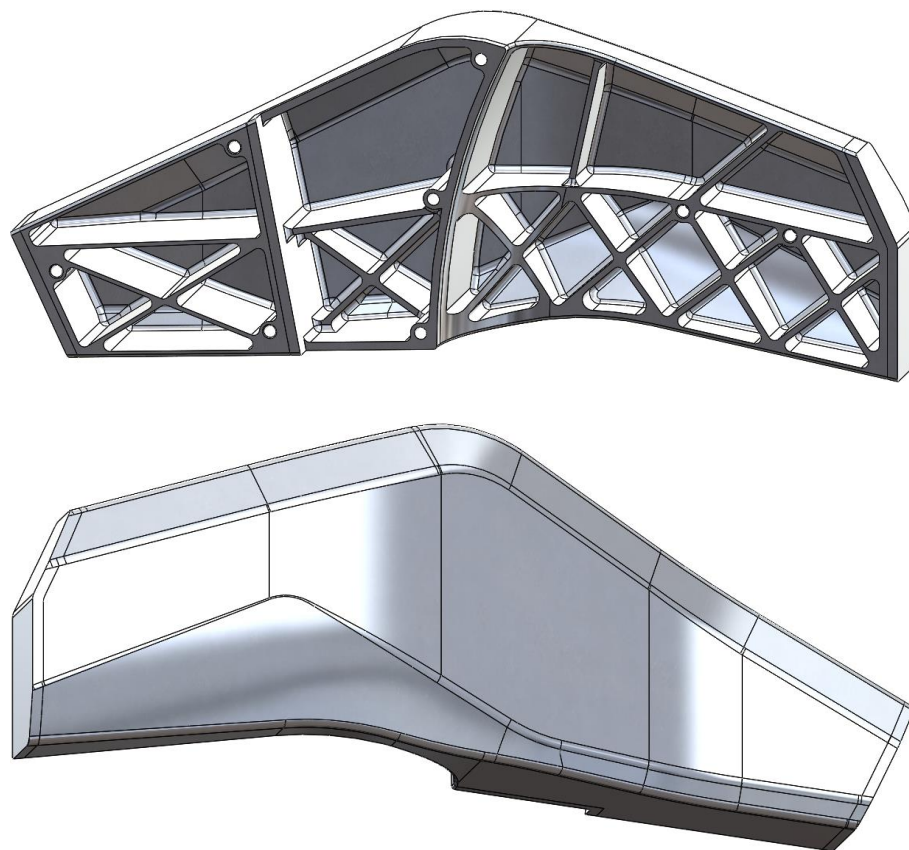


Figure 90. Final design - Right middle part

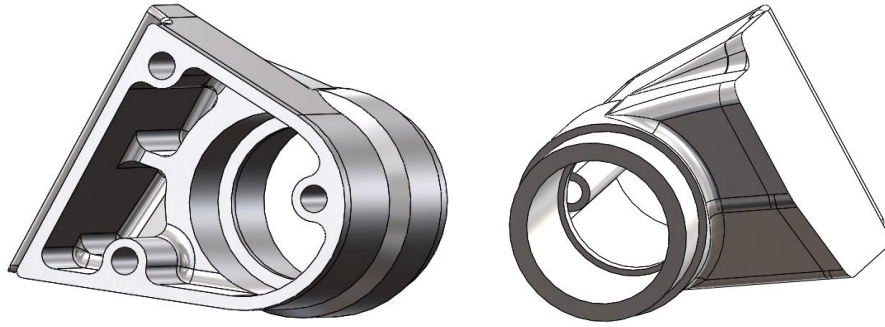


Figure 91. Final design - left front part

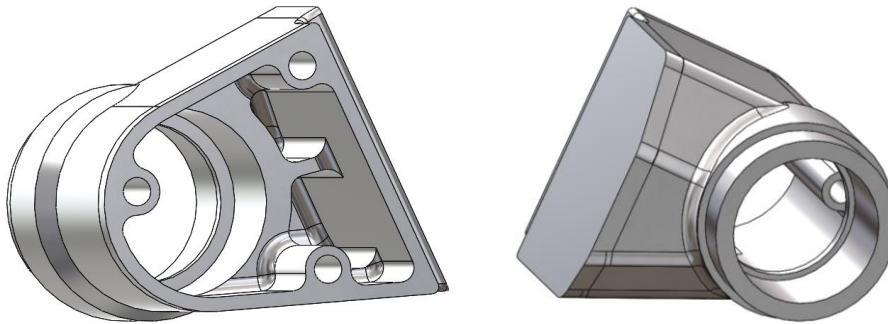


Figure 92. Final design - right front part

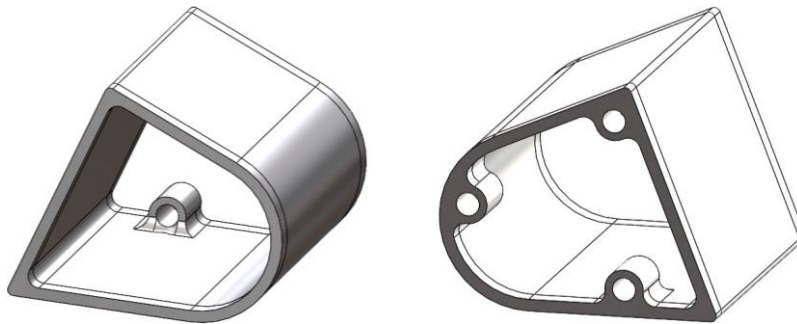


Figure 93. Final design - front part

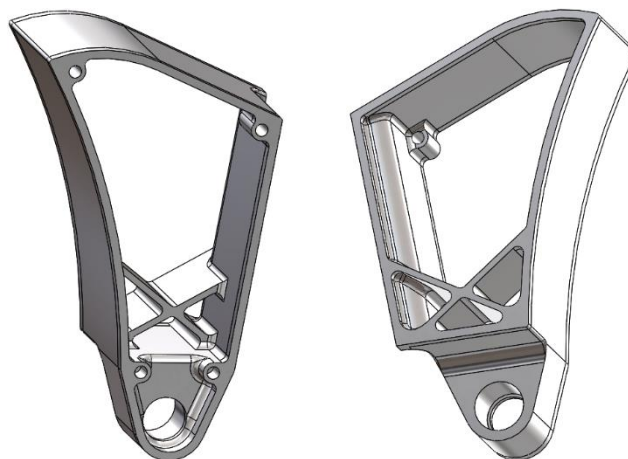


Figure 94. Final design - left suspension part

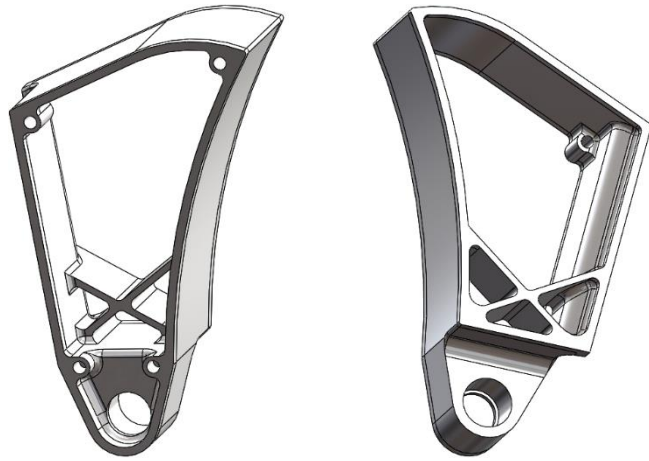
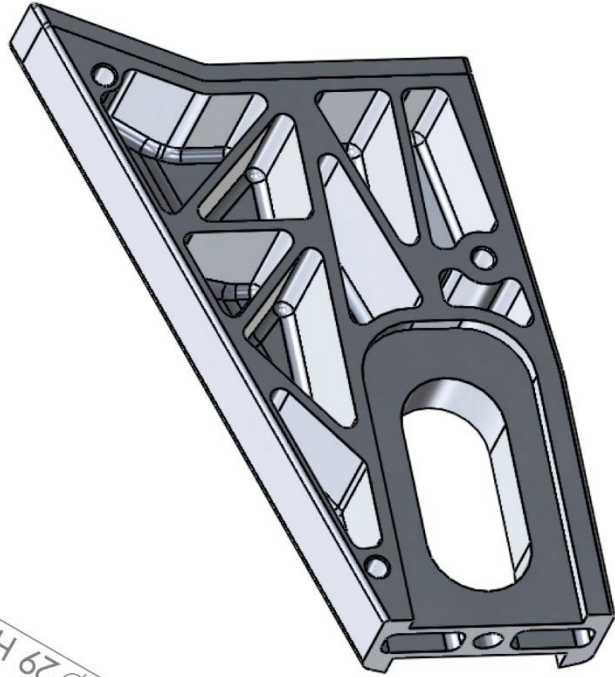
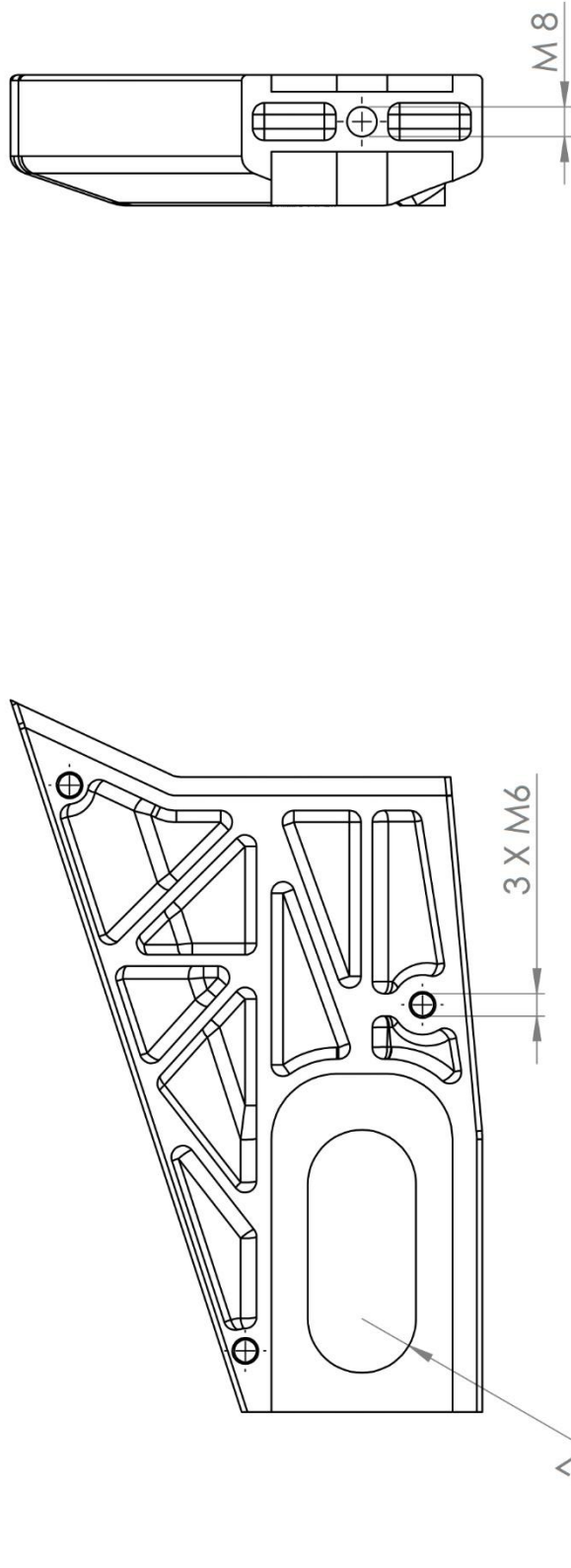
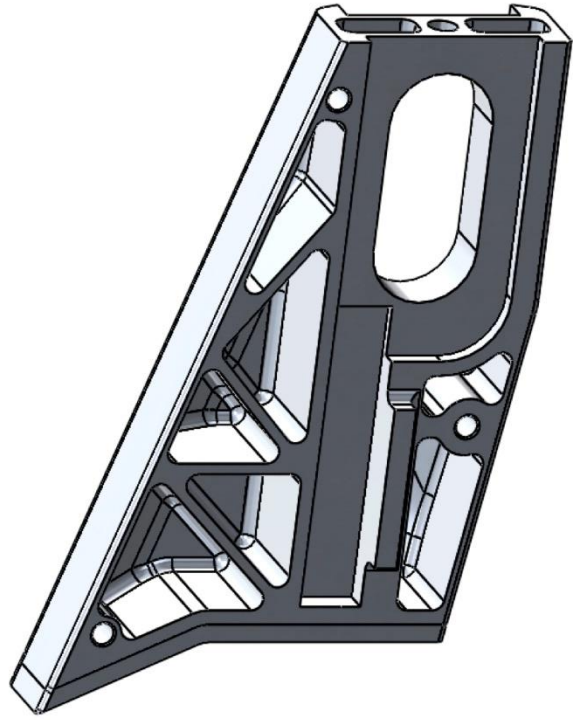
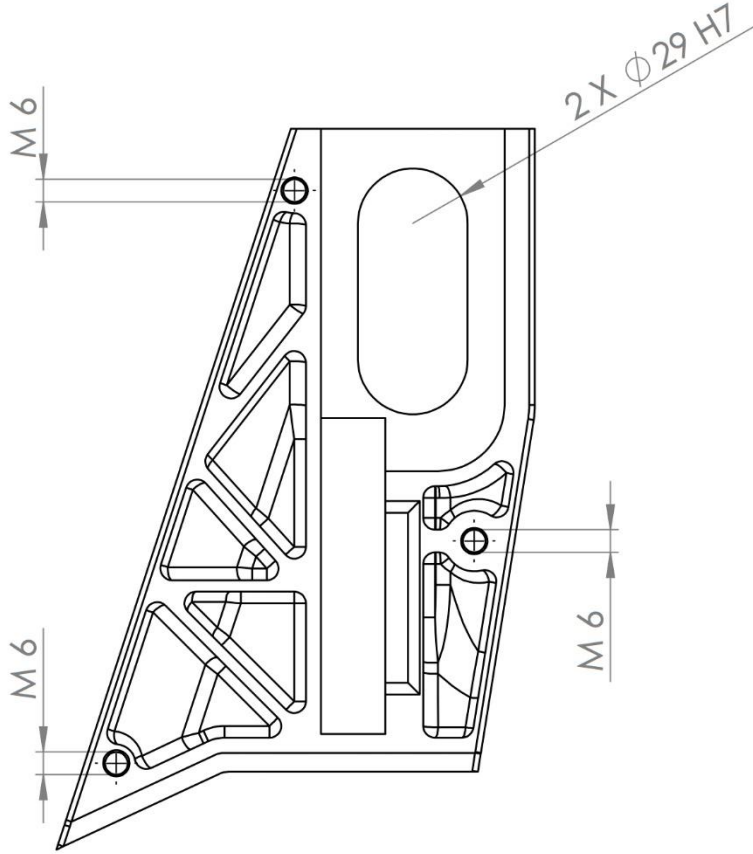
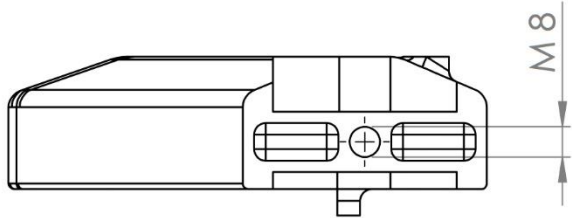


Figure 95. Final design - right suspension part

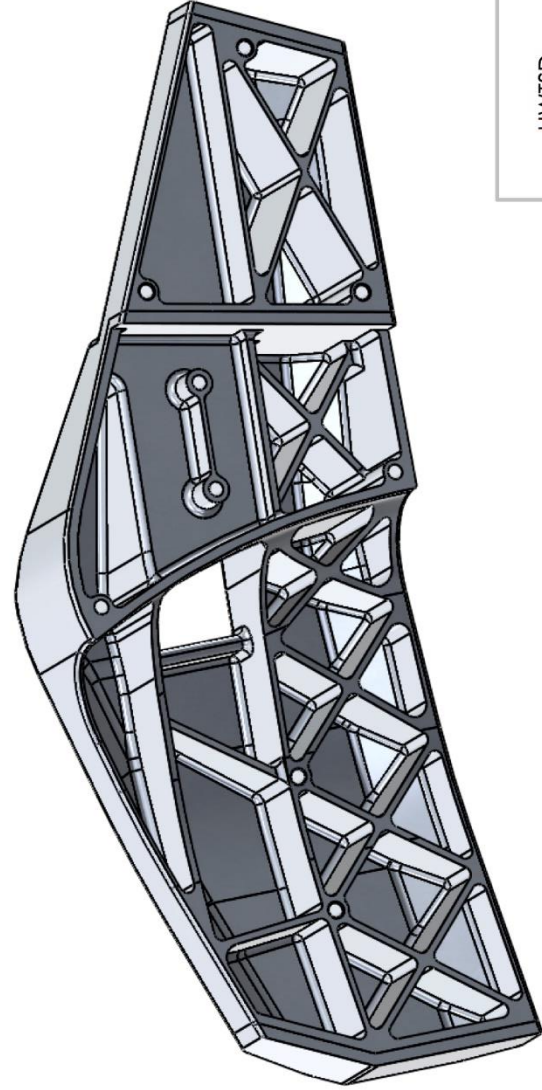
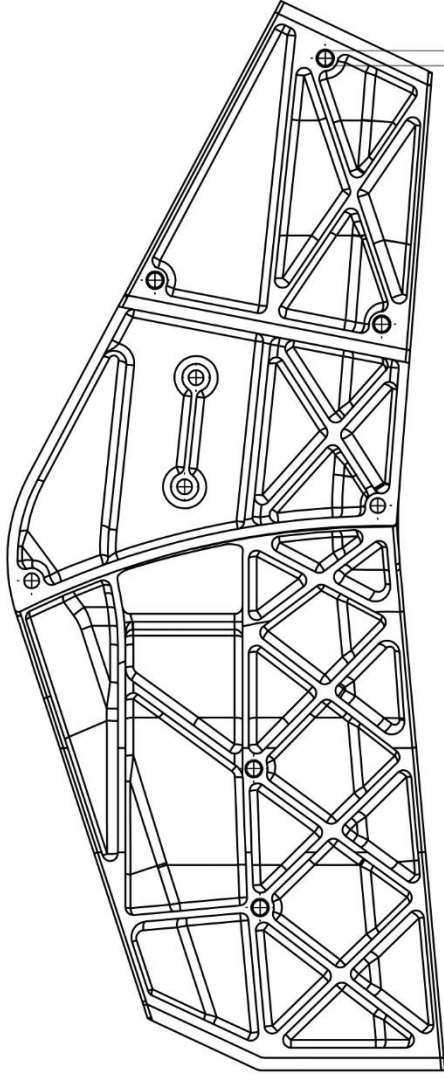
12.5.2. TECHNICAL DRAWINGS



UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 António Januário	Mass (g) WEIGHT: 594.21	Left Rear
Date: 17/02/2020	ISO	SCALE: 1:2
		SHEET 1 OF 1
		A4

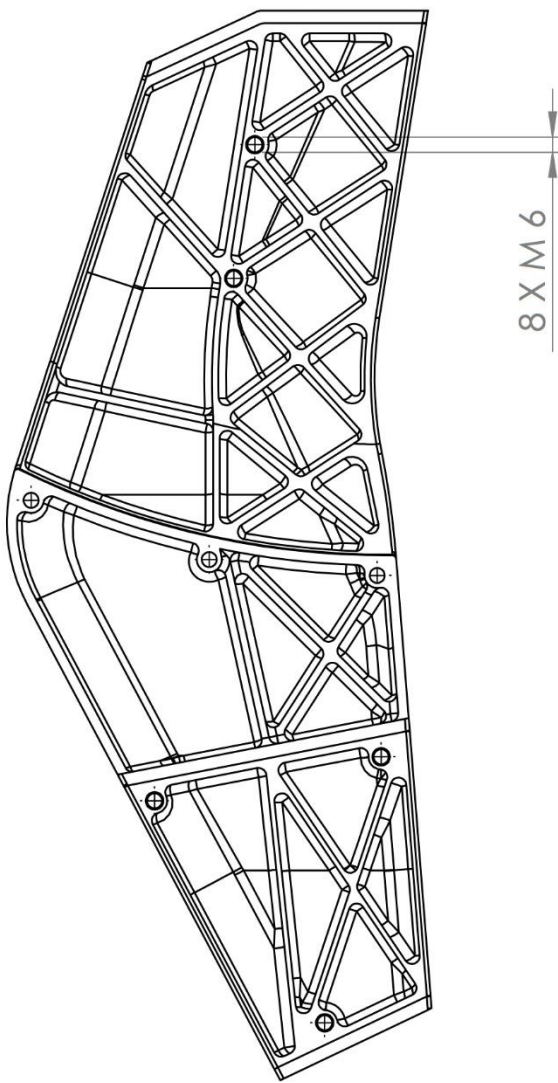
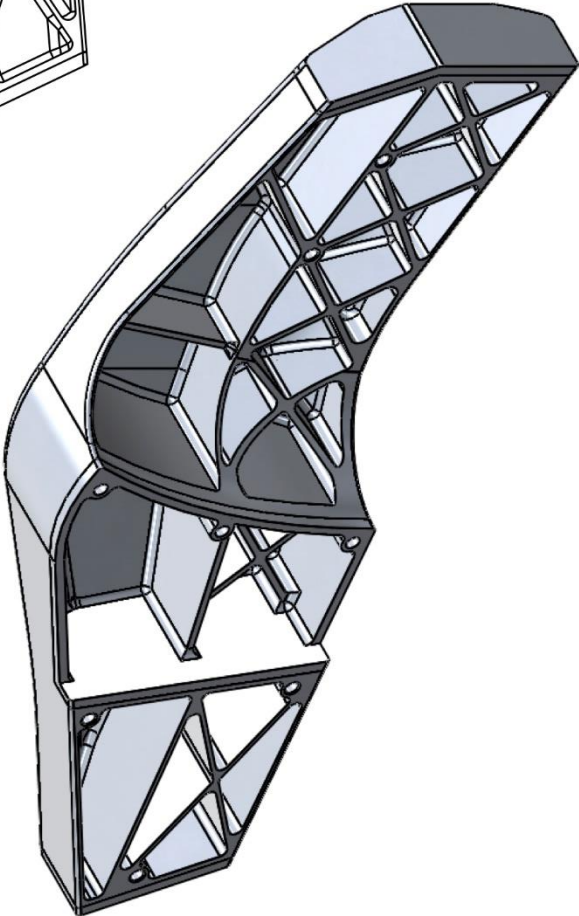


UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 665.50	Right Arm Back
Date: 17/02/2020	ISO	SCALE:1:2
		SHEET 1 OF 1
		A4

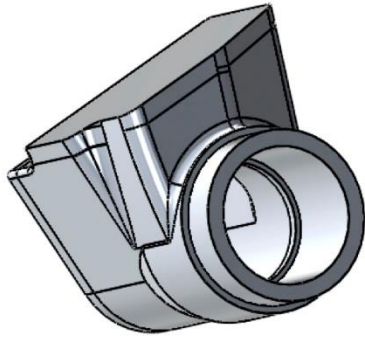
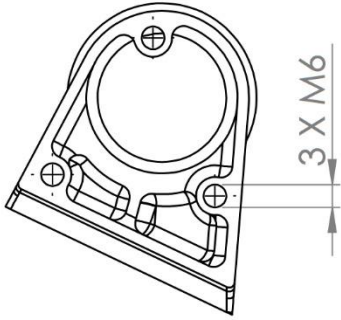
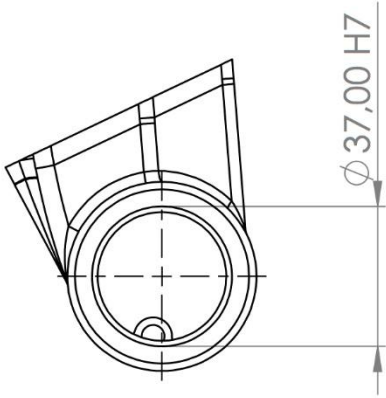


9 X M6

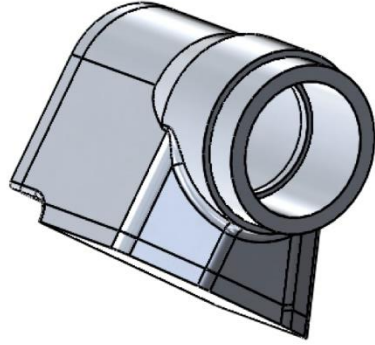
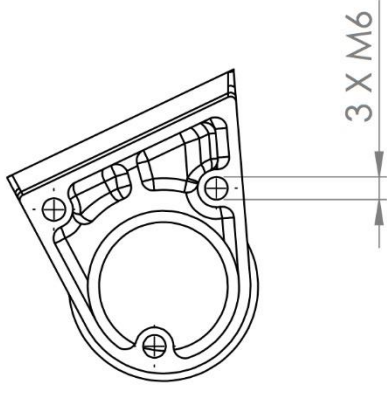
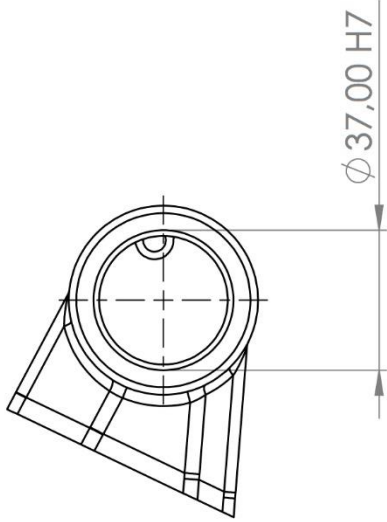
UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 1 629.53	Left Arm Middle
Date: 17/02/2020	ISO	SCALE:1:3
		SHEET 1 OF 1
		A4



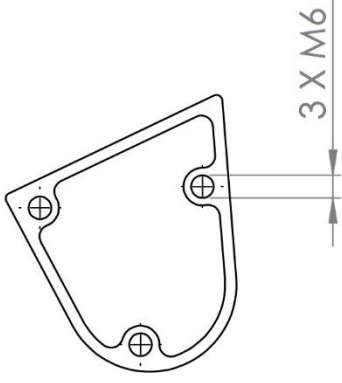
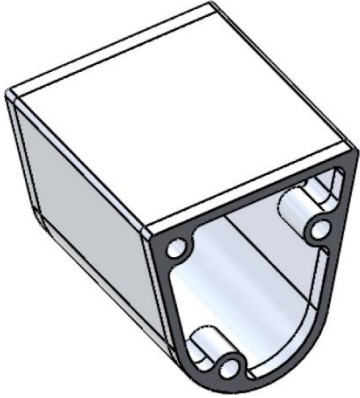
UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 1543.87	Right Arm Middle Section
Date: 17/02/2020	ISO	SCALE:1:3
		SHEET 1 OF 1
		A4



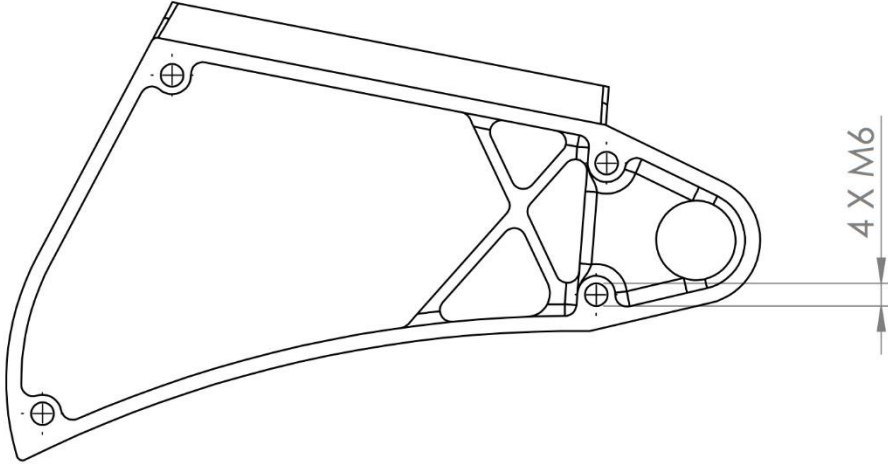
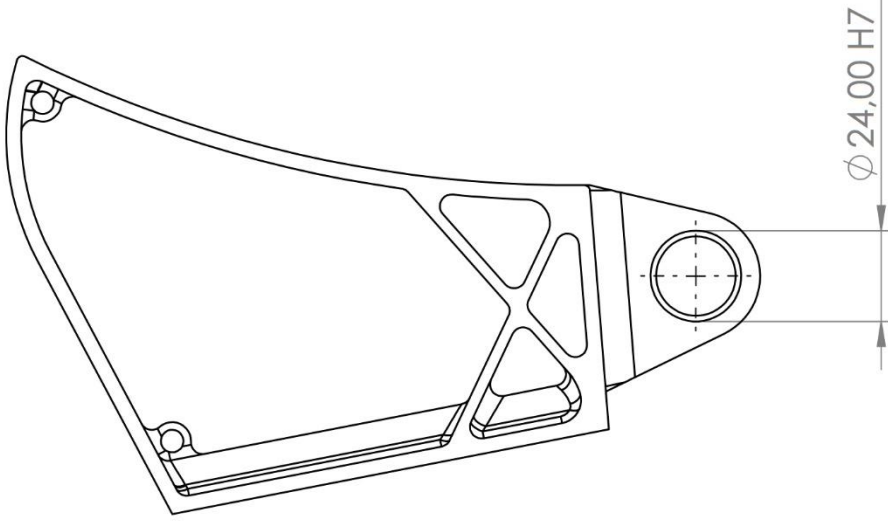
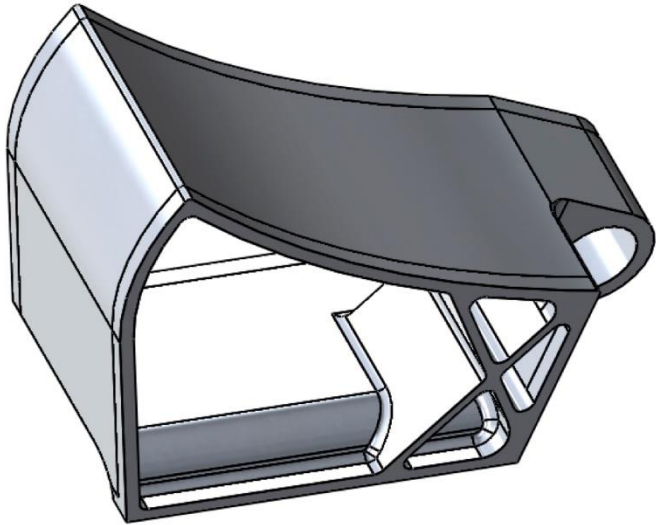
UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 284.13	Left Arm Front
Date: 17/02/2020	ISO SCALE: 1:2	SHEET 1 OF 1 A4



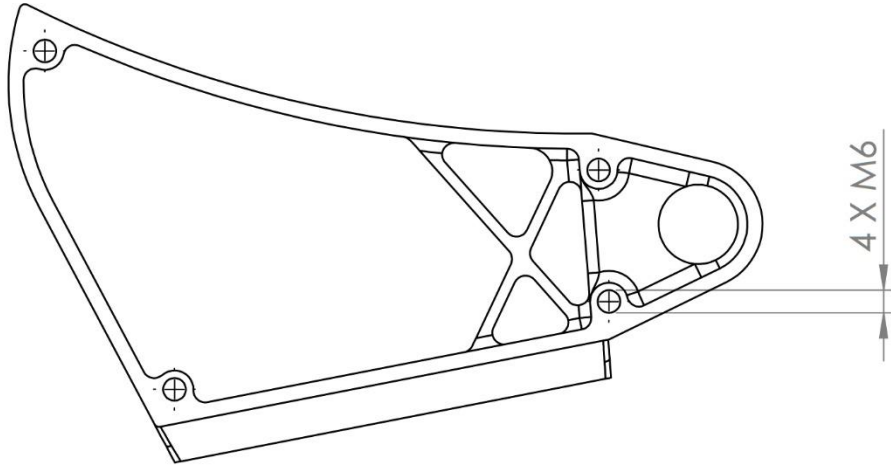
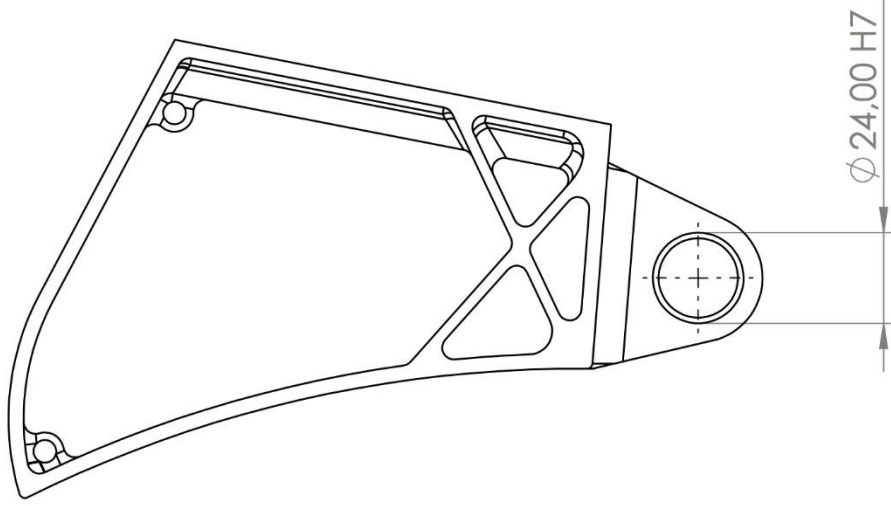
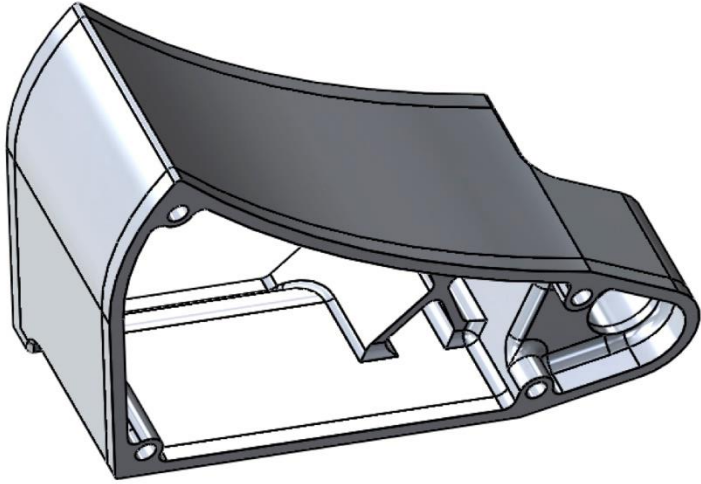
UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 277.45	Right Arm Front
Date: 17/02/2020	ISO SCALE: 1:2	SHEET 1 OF 1 A4



UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 145.35	Front Part
Date: 17/02/2020	ISO SCALE:1:2	SHEET 1 OF 1 A4



UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 521.66	Left Suspension
Date: 17/02/2020	ISO	SCALE:1:2
		SHEET 1 OF 1
		A4



UWTSD Motorcycle Engineering	Material 6061-T6 (SS)	Title Prototype Swingarm for IOM 2020	
Student N°: 1805566 Antônio Januário	Mass (g) WEIGHT: 482.87	Right Suspension	
Date: 17/02/2020	ISO	SCALE:1:2	SHEET 1 OF 1
			A4