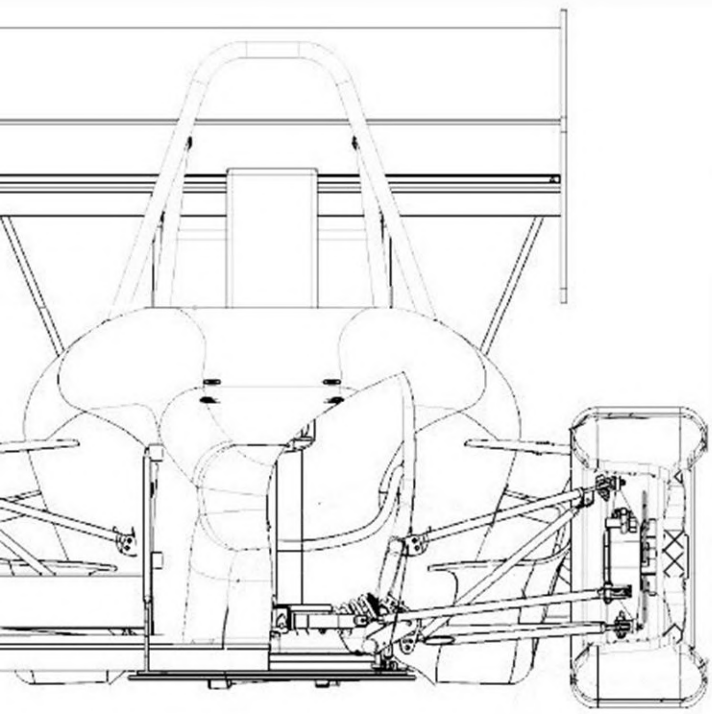
UNIVERSITY OF ALBERTA
FORMULA
RACING

UA-20 Final Design Report

MEC E 409 – Drivetrain System



Adam Tkalcic
1514092
atkalcic@ualberta.ca

December 6, 2019

Word Count (Excluding Figures & Tables): 1,864

TABLE OF CONTENTS

Introduction **1**
Design Specifications **1**
Conceptual Design **3**
UA-19 Design 3
UA-20 Concept 5
Final Design **10**
Critical Design Analysis 10
Cost Analysis 14
References **16**

LIST OF FIGURES

Figure 1: 3D CAD model of UA-19 car **4**
Figure 2: 3D CAD model of UA-19 drivetrain **4**
Figure 3: 3D CAD model of UA-19 car highlighting extreme driveshaft angles **5**
Figure 4: 3D CAD model of UA-19 drivetrain highlighting chain tensioner **5**
Figure 5: 3D CAD model of UA-20 concept car **6**
Figure 6: FEA results comparing UA-19 design and UA-20 concept **7**
Figure 7: 3D CAD model of UA-20 concept drivetrain **7**
Figure 8: Sketch of UA-20 concept chain tensioner **8**
Figure 9: 3D CAD model of UA-20 car highlighting the drivetrain **11**
Figure 10: 3D CAD model of UA-20 drivetrain final design **11**
Figure 11: 3D CAD model of UA20 driveshaft angles reduced to 2.9° **12**
Figure 12: 3D CAD model of UA20 idler sprocket assembly **12**
Figure 13: Chain tension vs. Engine Speed carried through KTM 690 6-speed transmission **13**
Figure 14: 3D CAD model of UA20 left and right differential hanger final design **13**
Figure 15: Mass analysis of UA20 drivetrain mounting system **14**

LIST OF TABLES

Table 1: Drivetrain Design Specifications **2**
Table 2: Concept Evaluation Matrix **9**
Table 3: Future Work Project Schedule **14**

APPENDICES

Appendix A: Design Compliance Matrix **17**
Updated Design Compliance Matrix 18
Appendix B: Preliminary Concept Finite Element Analysis **20**
Appendix C: Concept Differential Hanger Weight Analysis **24**
Appendix D: Differential Hanger Loading Calculations **28**
Section 1 – Chain Tension 29
Section 2 – Differential Hanger Reaction Forces 34
Appendix E: Differential Hanger FEA Studies **36**
Section 1 – Finite Element Analysis Pre-Processing 37
Section 2 – Iterative Results 49
Section 3 – Final Design Finite Element Analysis Post-Processing 53
Section 4 – Finite Element Analysis Validation 63
Appendix F: Differential Bearing Lifespan Calculations **67**
Appendix G: Cost and Manufacturing Analysis **70**
Appendix H: Detailed Engineering Drawings **76**
Appendix I: Design Authorities **92**

Introduction

Formula SAE is a worldwide collegiate competition where groups of engineering students design, manufacture and race small scale open wheel race cars. The overarching goal at competition is to score the highest amount of points over a series of events, ultimately determining which team's design is the best. The purpose of this report is to outline the design problems, specifications and overall objective for the University of Alberta Formula Racing teams' 2020 drivetrain development. The drivetrain is defined as all components that transmit power and torque to the driving wheels apart from the engine itself. It is a vital component of the vehicle and reliability is one of the most important factors because a failure with the drivetrain would render the car undrivable.

The deliverables for this project is a drivetrain system that helps the team improve upon 2019 competition results by means of being more reliable and lightweight than the previous iteration. In order to achieve this, the approach is to maintain a similar design philosophy to the UA-19's successful design while improving specific areas to meet our goal. The major focus is incorporating the rear engine mounts into the differential mount, reducing the total number of parts needed to be manufactured and thus the overall weight. Doing so will also create one rigid connection between the power unit and drivetrain improving the overall efficiency and transmission of power through the sprockets.

By improving UA-19's already well-established drivetrain, the team is confident that the updates will pay dividends throughout all static and dynamic events at competition.

Design Specifications

Rule T.5.1 (SAE International) states "Any transmission and drivetrain may be used" thus creating no limitations on the choice of drivetrain components. Due to the reliability of the UA-19 Drexler differential, a decision has been made to re-use the differential for UA-20 and save on overall cost. The differential hangers must accommodate the Drexler Formula SAE differential and allow it to freely spin when power is transferred from the engine through the chain. In addition, the differential hangers should incorporate mounts for the two engine mounting points on the rear of the engine. Reducing the weight of the differential mounting assembly by 10% (to approx. <1900 g) for UA-20 is a major focus, however it must also maintain strength to resist engine loads under acceleration, braking and cornering, and a maximum tension of 9600N from the chain.

Packaging constraints are governed by rear chassis design and the location of the differential is constrained by the suspension geometry and location of the rear hubs. Drivetrain design will have to be executed closely with chassis and suspension leads to ensure no conflicts arise and that the differential is positioned where the driveshaft angles are no more than 12 degrees.

An adequate chain tensioning method must also be designed to manually set a preload tension on the chain and accommodate for chain stretch throughout vehicle use. SAE International rules (2020) T.5.2.7 states all chain drives must have a scatter shield manufactured from 0.105" minimum thickness steel and a minimum width equal to three times the width of the chain. The chain scatter shield should be designed as close to the minimum requirements as possible to minimize additional unnecessary weight.

Materials for the differential hangers will have a big impact on manufacturing, budget, weight and strength and therefore will be carefully considered. However, due to the success on the UA-19 car, the usage of aluminum alloys is highly likely.

Table 1: Drivetrain Design Specifications

	Design Authority	Specification	Priority
Chain Drive	Formula SAE: T.5.2.7	All chain driven vehicles must have a shield in case of failure. Chain scatter shield must be made of 0.105” minimum thickness steel and a minimum width equal to three times width of the chain.	5
	Formula SAE: T.5.2.9	Chain scatter shield must be mounted using 6 mm or 0.25” minimum diameter critical fasteners.	5
	Formula Racing Team	Due to chain stretching during use and the nature of assembly, design must provide a manual chain tensioning method so that the chain is at the necessary preferred tension.	5
	Formula Racing Team	Design should allow the chain and chain tensioner to be assembled and adjusted in <2 hours for easier serviceability at competition and in time limited scenarios.	1
Fasteners	Formula SAE: T.8.2.1	Although drivetrain is not explicitly mentioned in this rule, the loads being applied are significant enough that fasteners should meet or exceed SAE Grade 5 or Metric Grade 8.8.	5
	Formula SAE: T.8.3.(1,2)	Although drivetrain is not explicitly mentioned in this rule, positive locking mechanisms should be used to prevent catastrophic failure due to fasteners coming loose.	5
Strength	Formula Racing Team	Must be able to support minimum 2 g lateral and longitudinal engine loads in acceleration/braking and cornering. Differential mounts must also withstand a maximum 9600 N of chain force, based on UA-19 horsepower, torque and final gearing ratio.	5
Differential	Formula Racing Team	Differential hangers must be designed to house a Drexler FSAE differential and allow it to freely spin.	5
Weight	Formula Racing Team	Differential/Engine mounts to be at minimum 10% less weight than total UA-19 mounting assembly (2100 g).	4
Driveshafts	RCV Performance	Design differential mount so that RCV Performance driveshafts have less than 12˚ of deviation in all planes as dictated by driveshaft supplier. As driveshaft angles approach 0˚, reliability and performance both increase.	4

		UA-19 vehicle had 9˚ of driveshaft angle increasing component wear and the possibility of a CV joint failure.	
Chain and Sprocket Clearance	Formula Racing Team	Chain must not drag on chassis or other components under normal operation and a range of sprocket sizes (Ø177.1 mm – Ø202.3 mm) must clear the chassis.	5
Vibration Dampening	Formula Racing Team	Engine mounts must be soft-mounted to provide vibration dampening. In previous years, engine was hard mounted to chassis causing vibrations throughout entire vehicle and discomfort to driver.	3

Conceptual Design

The drivetrain system consists of many indispensable components required to transfer power to the tires including the differential, final drive sprockets, driveshafts, chain and chain tensioner. Packaging of all these components within chassis, while providing the required strength to resist forces applied to the assembly prevails as the largest design challenge when approaching new ideas. Due to this, an updated design specification matrix for the UA-20 drivetrain can be found in appendix A.

UA-19 Design

The UA-19 drivetrain pictured in figure 1 and figure 2, proved successful and performed reliably, however it also came with many disadvantages which ultimately guides the conceptual design for the UA-20 car. Packaging of the UA-19 drivetrain was little concern as it was secured to the rear of the chassis in a cantilevered fashion with minimal constraints from the chassis and other components of the vehicle. This provided easy access for drivetrain maintenance if needed, however, mounting the drivetrain in this fashion came with disadvantages. Shown in figure 3, the UA-19 driveshafts were forced into extreme angles of approximately 9 degrees. These angles increase wear on the CV joints and heighten the probability of a drivetrain failure. Furthermore, due to the location of the differential outside of the chassis, and separated from the engine, the forces applied by the chain under load induce a moment on the entire assembly causing displacement of the differential hangers.

The UA-19 chain tensioning method consisted of an eccentric disk mounted to each differential hanger and when spun, would move the differential and rear sprocket backwards increasing the chain tension. Both extreme positions of the eccentric disks can be seen in figure 4. The eccentric disks provided a rigid method to tension the chain since the disks were hard mounted to the differential hangers. This method however, brought about a complicated process to achieve the desired chain tension and only 1 total link of adjustability.

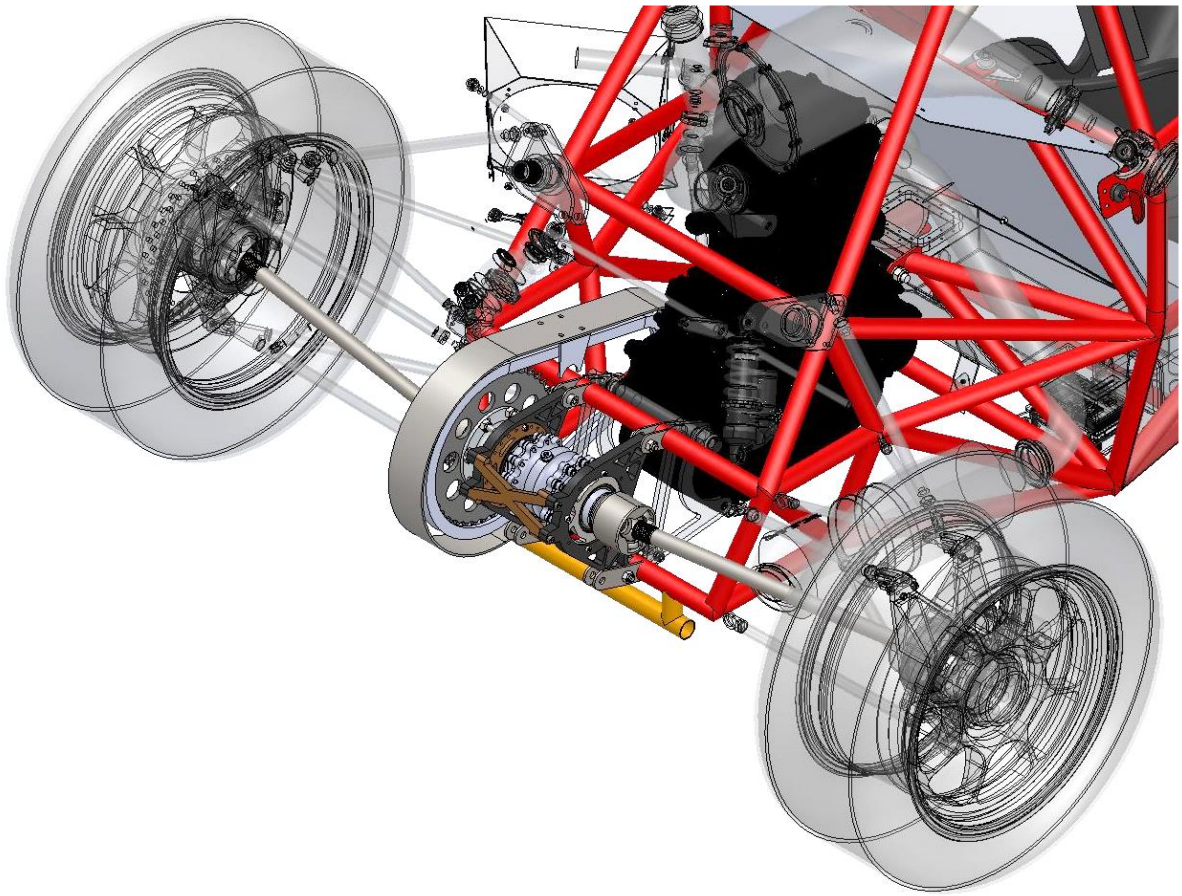


Figure 1: 3D CAD model isometric view of the UA-19 car highlighting externally mounted drivetrain system to the rear of the chassis.

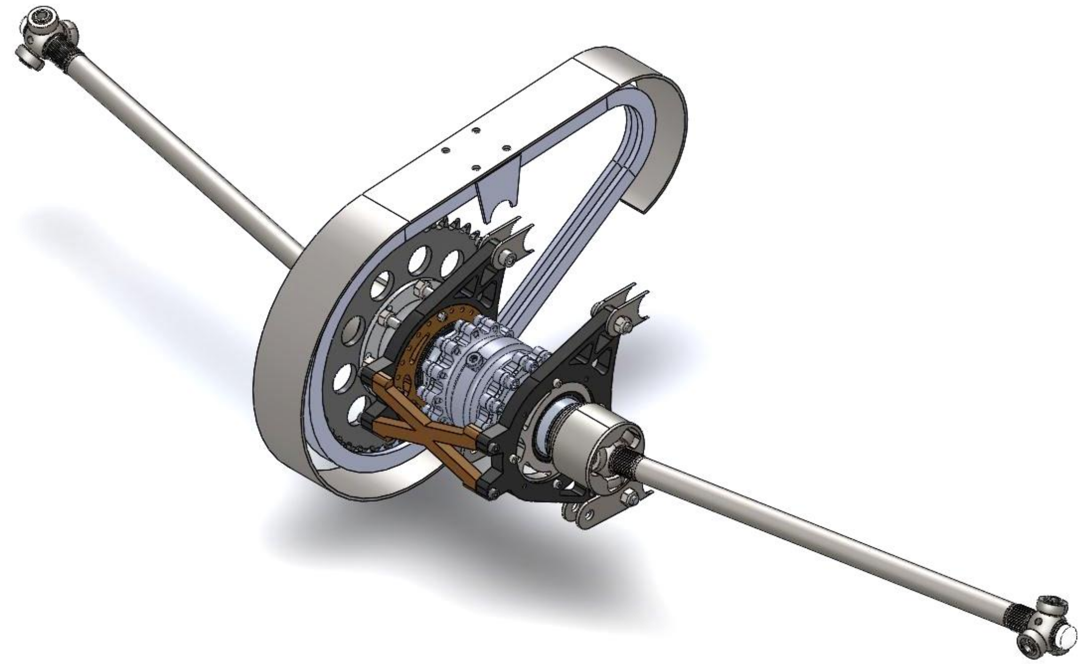


Figure 2: 3D CAD model isometric view of the UA-19 drivetrain sub-assembly.

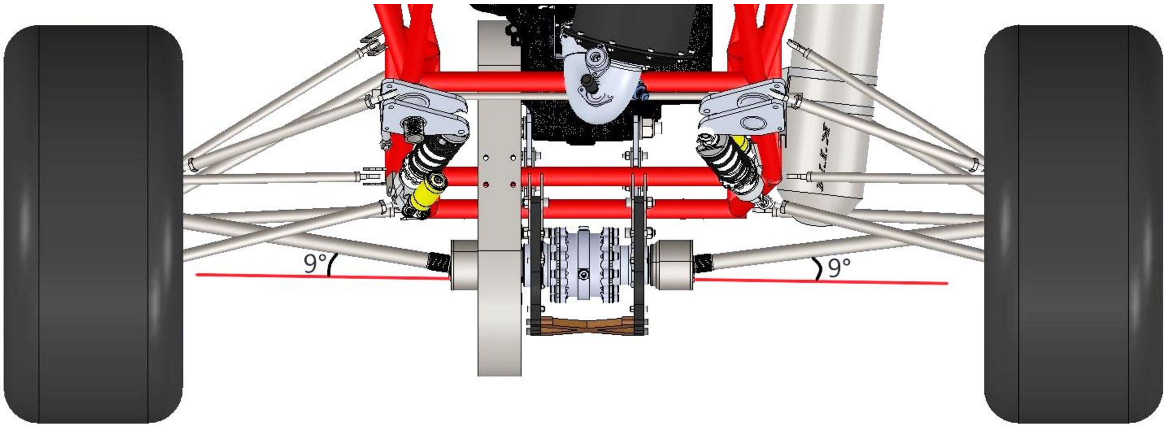


Figure 3: 3D CAD model top view of the UA-19 drivetrain showing the extreme driveshaft angles.

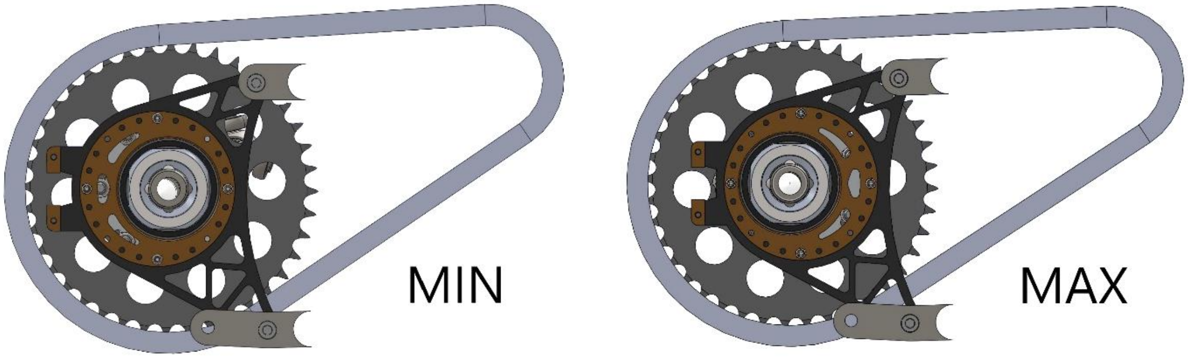


Figure 4: 3D CAD model cross sectional view of the drivetrain assembly highlighting the adjustability of the eccentric disk chain tensioner.

UA-20 Concept

The UA-20 design aims to eliminate the disadvantages uncovered about the UA-19 design beginning with the concept of combining the differential hangers with the rear engine mounts as pictured in figure 5. Not only does this create a more rigid connection between the drivetrain and powertrain, it also reduces the number of parts to be manufactured, potential weight and solves many of the drawbacks previously stated.

Mounting the drivetrain in conjunction with the engine creates a more rigid connection between both the driving sprocket and driven rear sprocket and adds an additional differential hanger mounting point. Finite element analysis was conducted to compare the UA-19 design with the UA-20 concept and validates an increased magnitude of rigidity. Constraints, loads and a more in-depth analysis of the studies can be viewed in appendix B. The studies reveal that under the same arbitrary chain load the UA-19 differential hanger deforms approximately 8 times more than the UA-20 concept as shown in figure 6.

In addition, mounting the drivetrain in this fashion moves the entire drivetrain assembly closer to the engine reducing the driveshaft angles to 2 degrees and thus decreasing overall driveshaft component wear. The UA-20 concept also has the potential to weight less than the 2100 g UA-19 full assembly which includes the engine mounts. The UA-20 concept, weighing in at approximately 2300 g in its current state, has a high probability of being lighter after FEA and topology studies to be performed in the final design phase. See appendix C for a more in-depth concept differential hanger weight analysis.

Due to the spatial constraints caused by moving the drivetrain assembly inside of the chassis, the eccentric disks can no longer be used to tension the chain. The UA-20 concept utilizes an idler sprocket pictured in figure 7 to manually set the chain tension. A sketch describing how the desired tension would be obtained is shown in figure 8. Tensioning the chain in this fashion would allow for greater overall adjustability than the eccentric disks in a much quicker and easier way.

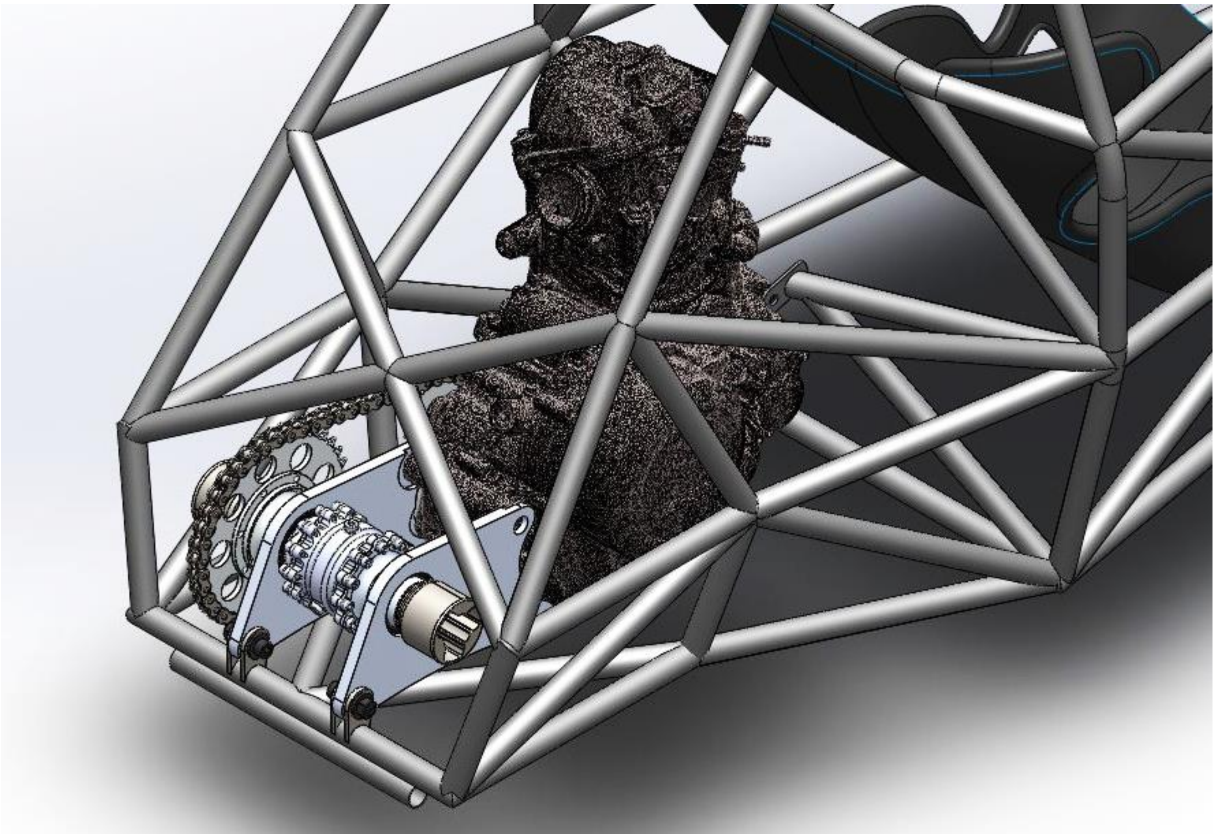
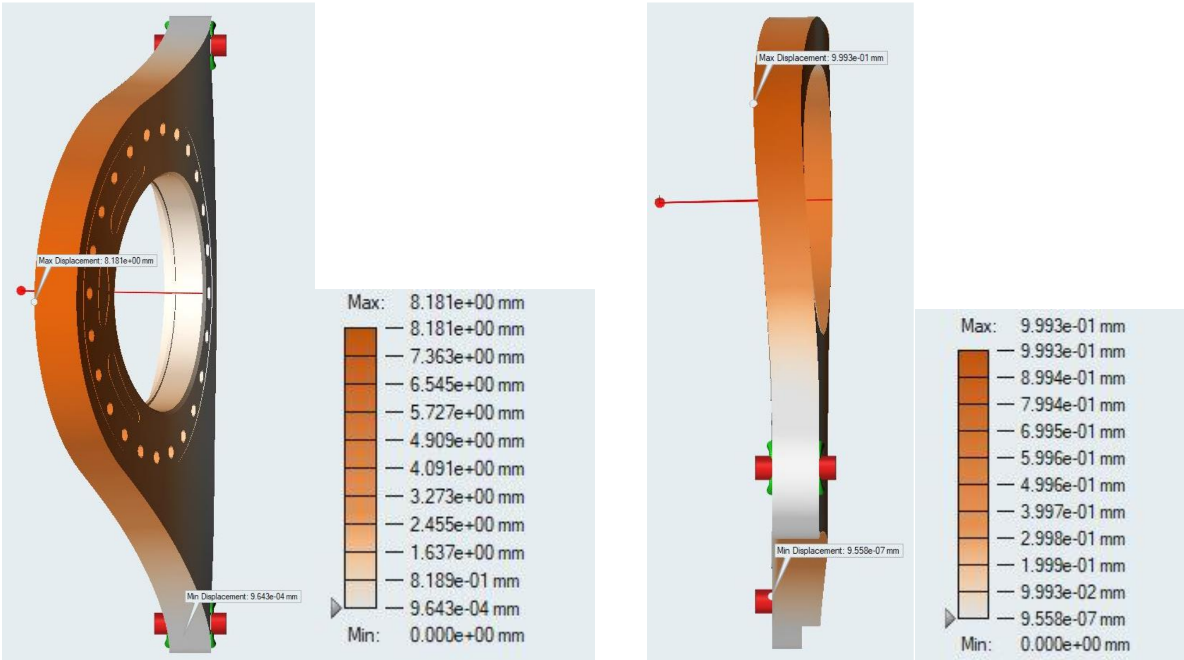


Figure 5: 3D CAD model isometric view of the UA-20 concept highlighting internally mounted drivetrain assembly with differential hangers mounting to rear of engine.



UA-19 Design **UA-20 Concept**

Figure 6: Displacement study performed in Altair Inspire 2019.3 comparing maximum deflection of UA-19 design and UA-20 concept. Both deformed states pictured above are exaggerated equally.

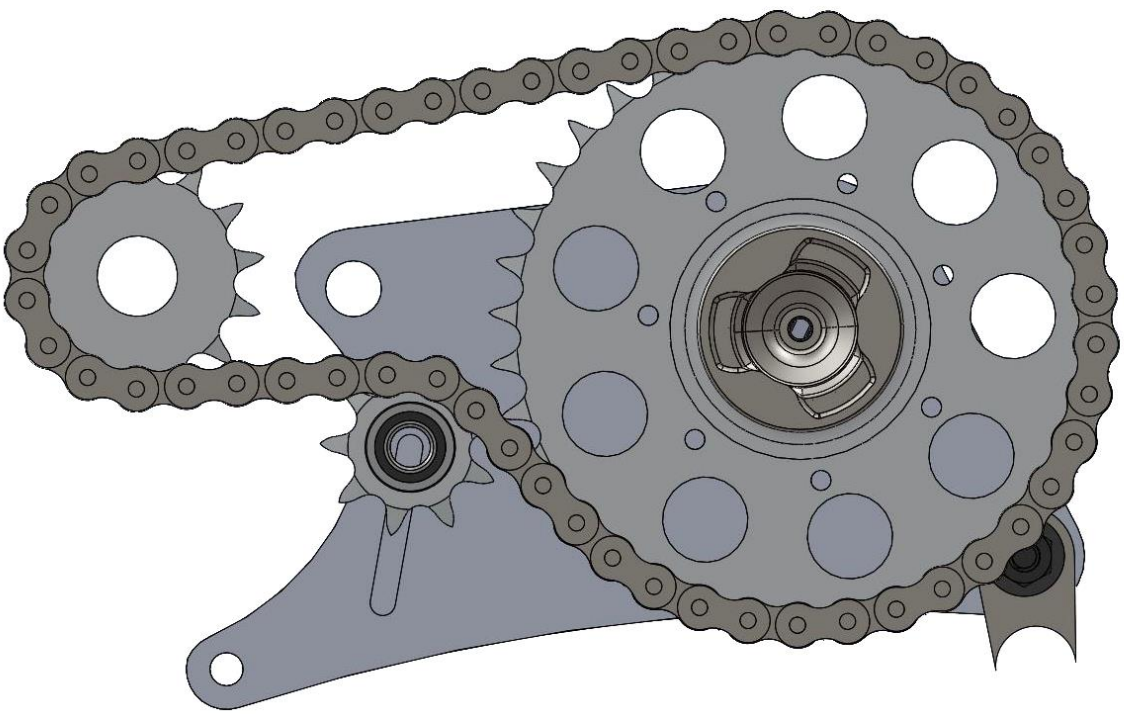


Figure 7: 3D CAD model side view of the UA-20 drivetrain assembly highlighting idler sprocket in adjustable slot.

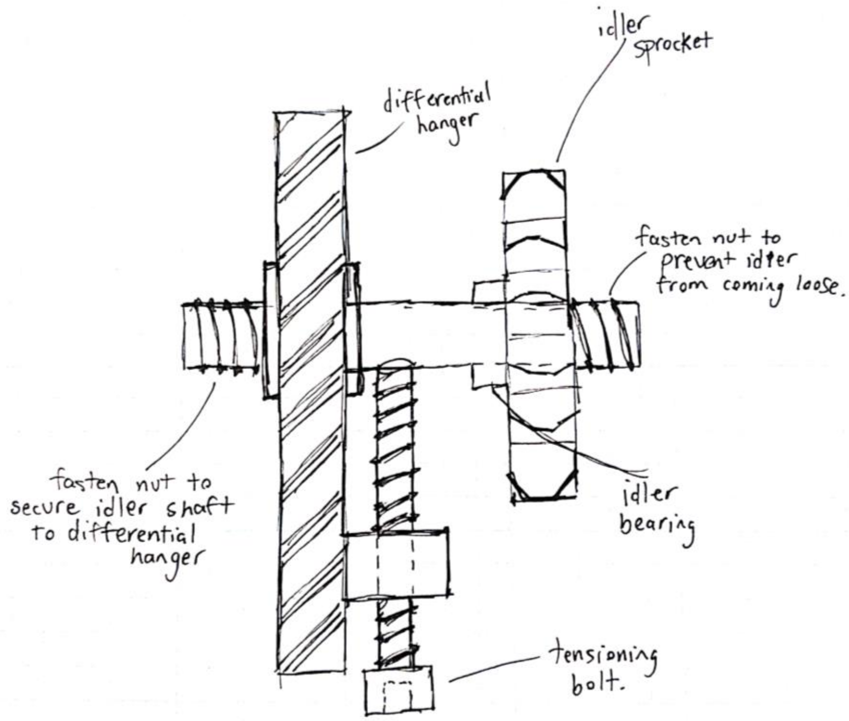


Figure 8: Hand drawn sketch of the UA-20 idler sprocket tensioning system.

Table 2: Concept Evaluation Matrix

Specifications	Notes on Specifications	Notes on Score	Weight Factor	UA-19 Design		UA-20 Concept	
				Score (/10)	Weighted Score	Score (/10)	Weighted Score
Rigidity and Strength	The differential hangers must withstand the forces applied to it and deform as little as possible under full loading to avoid the differential bearings from being dislodged.	UA-19 design was strong enough to withstand failure, however displaced considerably due to it's cantilevered mounting method. UA-20 concept proves approximately 8 times more rigid than previous design. (see appendix B: FEA Displacement Analysis)	5	1	5	8	40
Weight	Total weight of the assembly must be minimized as the drivetrain system influences a major portion of the vehicles' overall power to weight ratio.	UA-20 concept has the potential to weight less than the UA-19 design after final design optimization (see appendix C: Differential Hanger Weight Analysis). UA-19 full assembly weighed 2100 g with a 25% weight reduction due to topology. Unoptimized UA-20 concept weighs 2300 g.	5	6	30	7	35
Driveshaft Angles	Differential should be mounted in a location relative to the vehicles wheel hubs as to minimize the angle they are forced to take.	UA-20 concept is a stronger design. New concept achieves 2° of angle in comparison to the UA-19 design which exhibited 9° of angle, almost exceeding the manufacturer's acceptable limit.	4	2	8	9	36
Sprocket Tuning	Assembly should allow for enough clearance to swap rear sprocket sizes and achieve different final drive gearing.	Due to the external mounting of UA-19 design and limited chassis constraints a large range of sprockets could be utilized. UA-20 concept leaves less room to work with, however still an acceptable range (< Ø192 mm).	3	8	24	6	18
Accessibility	The differential hangers should be easy to work on, assemble and disassemble for maintenance or in the event of a component failure.	Previous design was much more accessible as it was mounted externally to the chassis. UA-20 concept utilizes the engine as a mounting point adding significantly more work to assemble and disassemble.	2	9	18	1	2
Chain Tensioning	Drivetrain must have a method to manually adjust the chain tension in an easy and efficient manner.	Use of the eccentric disks in the UA-19 design proved hard to use and offered limited range of motion. The new idler design offers a much larger range of motion and infinitely adjustable in a more timely and convenient process.	2	3	6	7	14
Total		**Max: 210**			91		145

Final Design

The final design of the UA-20 drivetrain maintains the general geometry and packaging outlined in the conceptual stage, combining the differential hangers with the rear engine mounts shown in figures 9 and 10. This design was chosen due to its tight and efficient packaging within the chassis, more rigid connection between the drivetrain and powertrain, and weight savings when compared to the UA-19 design. The final design commits to a custom final gear ratio of 2.6 utilizing a 14-tooth engine driven sprocket and 37-tooth drive sprocket with the ability to go up or down 1-tooth in the front and rear for tuning purposes. The design maintains driveshaft angles less than 3 degrees as shown in figure 11 and makes use of an idler sprocket to easily adjust tension on the chain, shown in figure 12. The updated design compliance matrix for the final design can be seen in appendix A.

Critical Design Analysis

Strength and weight were the two key design analyses used to determine the success of the final differential hanger design. An iterative design process was used to determine the final shape of the differential hangers by conducting an initial FEA study with the geometry outlined in the conceptual report to determine where major stress concentrations existed. In regions where stress concentrations were low, material was removed with the goal of reducing the weight as much as possible while maintaining the necessary strength to resist all loading. 7075-T6 aluminum was the material of choice for the differential hangers due to is lightness, ease of manufacturing and cheap cost relative to other options considered such as Ti-6Al-4V titanium. The primary loading applied to the differential hangers is the chain tension which was calculated to be a maximum of 12,131 N. See appendix D for more in depth calculations on chain tension and differential hanger reaction forces. Figure 13 displays a graph of the chain tension vs. engine RPM. Additional loading considered was the mass of the engine affixed to the differential hangers and the ability to support 2 g of lateral and longitudinal acceleration. Engine loads were found to be negligible on the differential hangers and is explained further in detail in appendix E. The final differential hanger design was determined strong enough to not reach material failure and the design is shown in figure 14.

Mass analysis of the UA-20 drivetrain was conducted in SolidWorks and the final mass of the drivetrain mounting solution including the idler assembly was calculated to be 1400 g as shown in figure 15. This is a 33% reduction in mass from the UA-19 drivetrain mounting assembly which included the eccentric chain tensioning disks.

Differential bearings were selected based on the differential outer bore size and were required to be as small as possible to reduce overall rotating mass. The left and right differential bearings were calculated with 90% reliability to fatigue after 1900 km and 57,000 km of driving respectively. In depth bearing lifespan calculations are shown in appendix F.

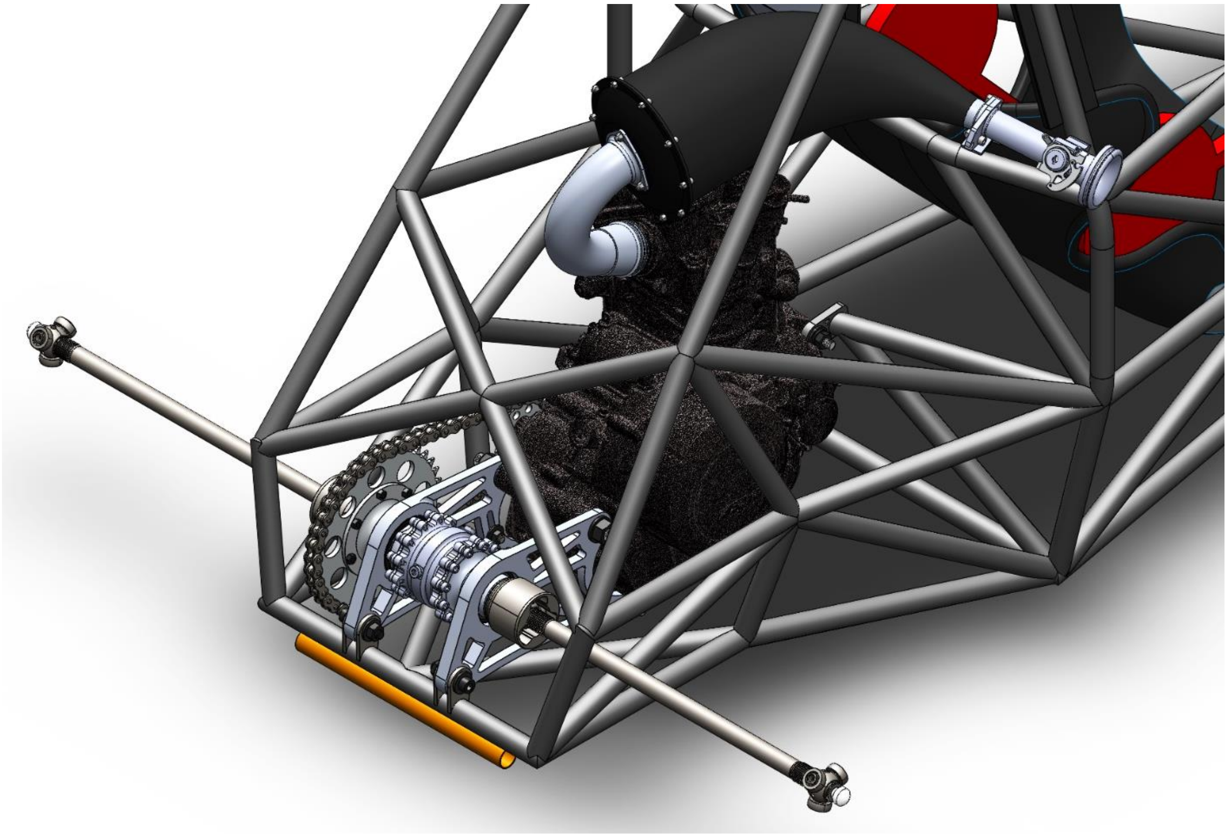


Figure 9: UA-20 final drivetrain assembly design packaged with the KTM 690 engine and chassis.

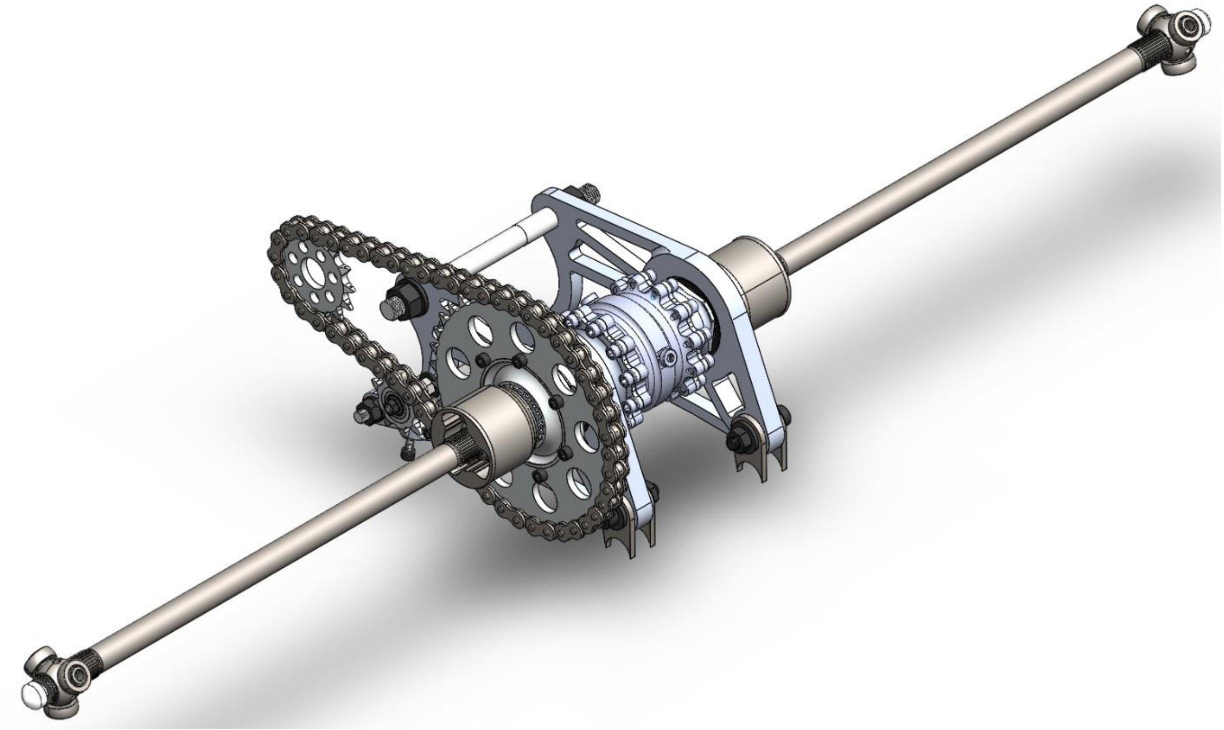


Figure 10: UA-20 final drivetrain assembly design.

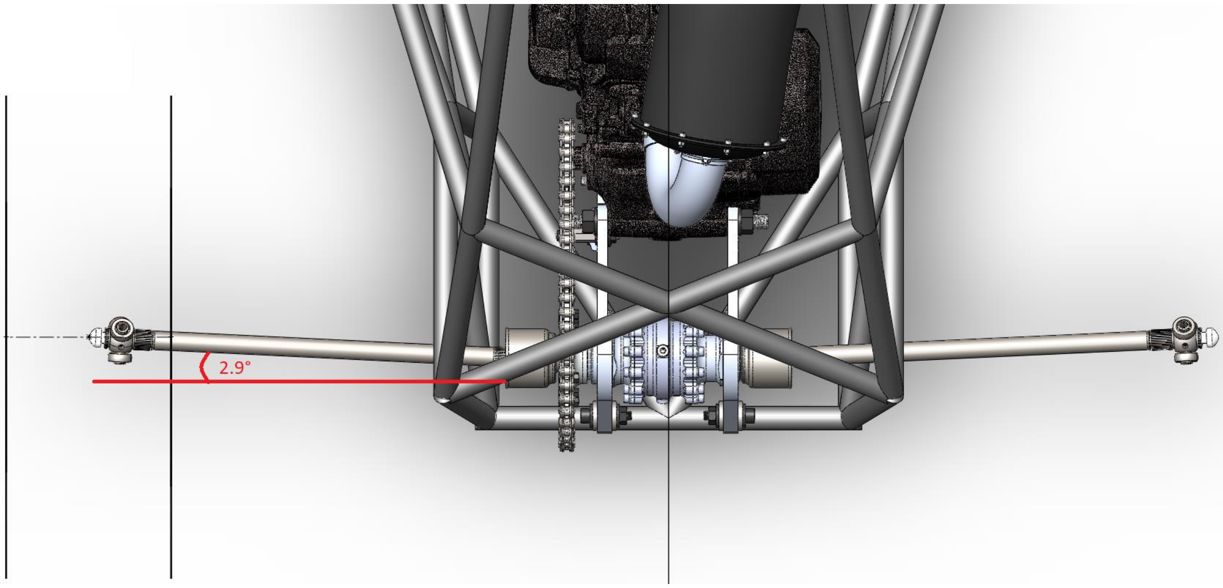


Figure 11: UA-20 driveshaft angles reduced to 2.9°.

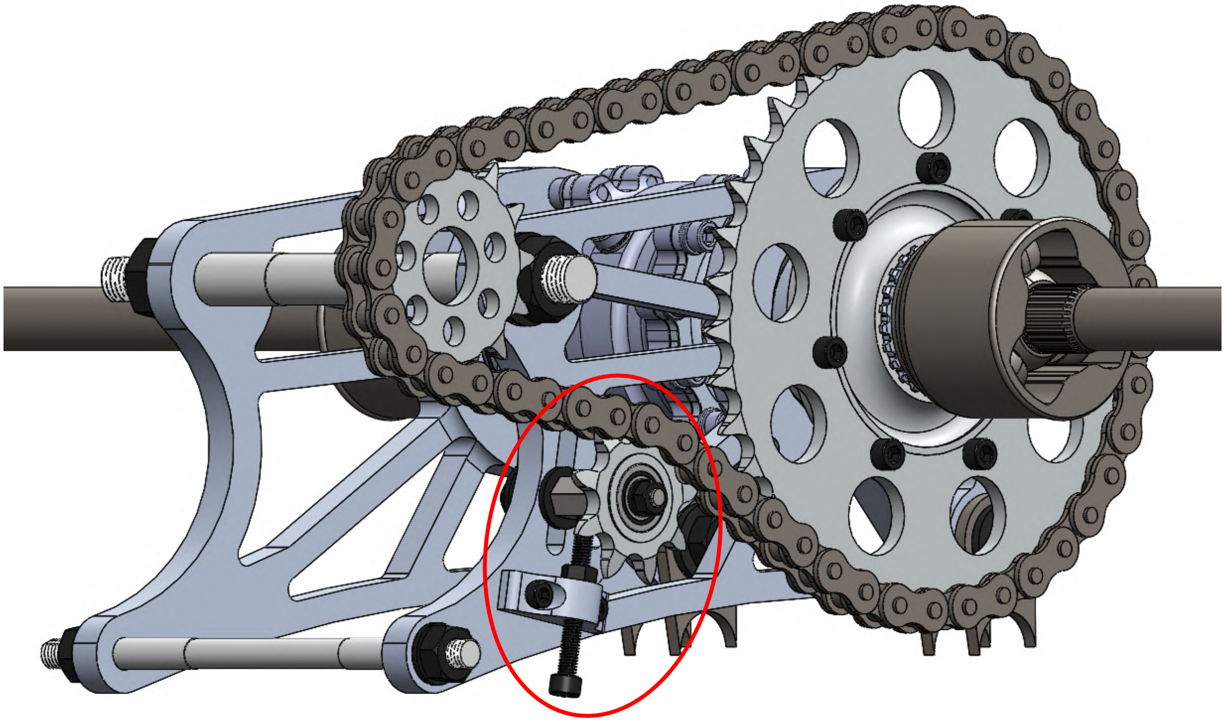


Figure 12: UA-20 idler sprocket assembly used to manually pre-tension the chain.

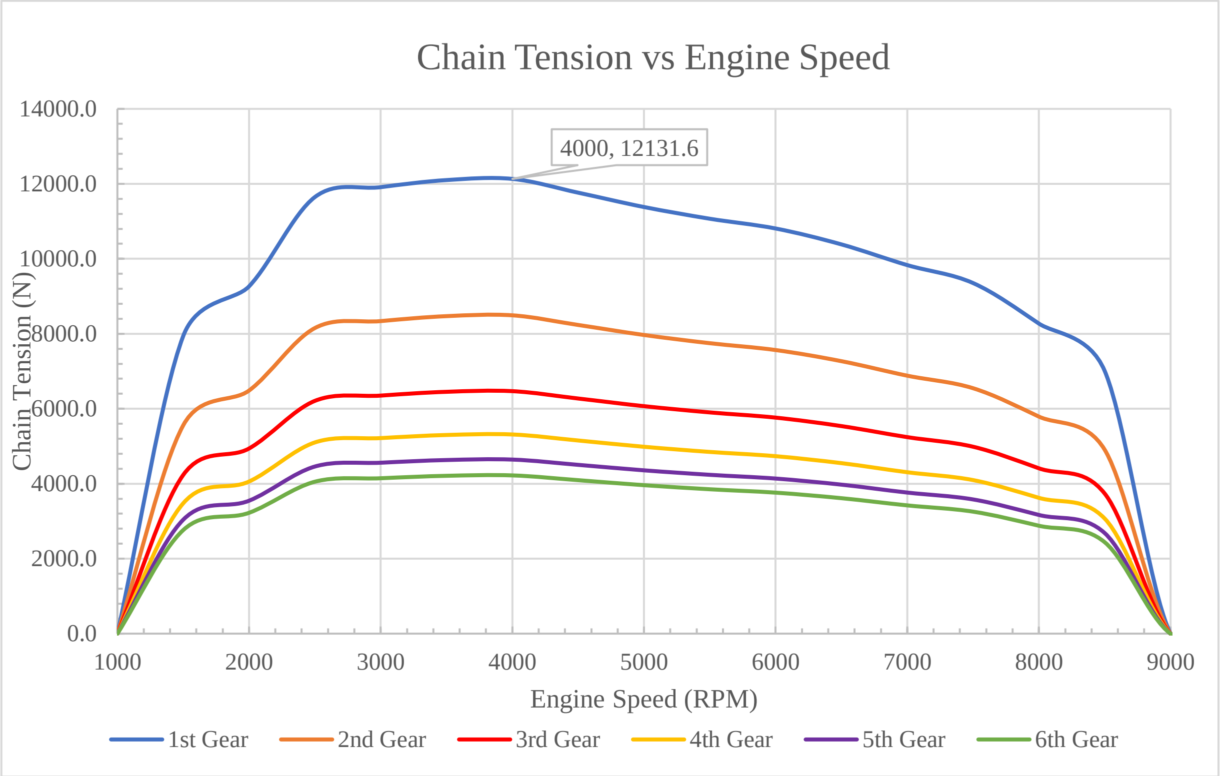
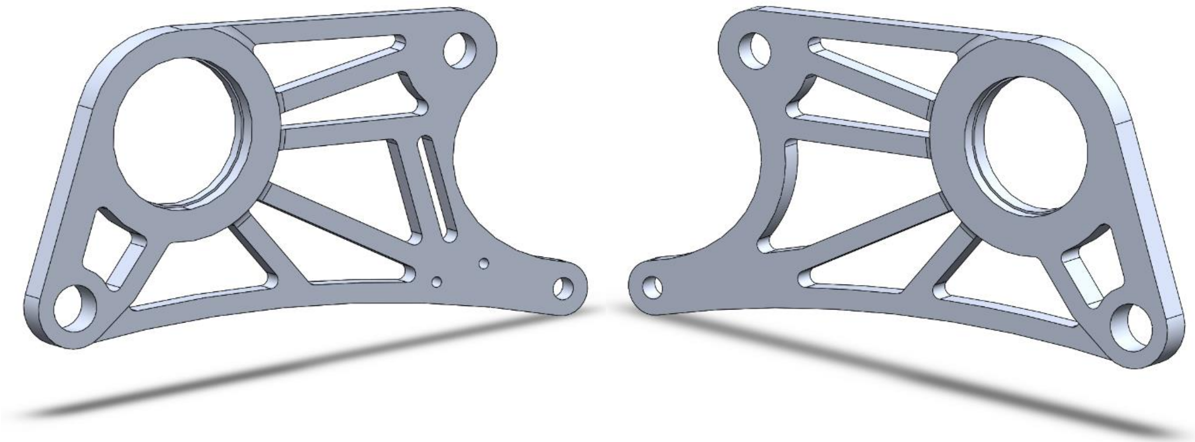


Figure 13: Chain tension (N) vs. Engine Speed (RPM) carried through KTM 690 6-speed transmission.



Left Differential Hanger ***Right Differential Hanger***

Figure 14: UA-20 left and right differential hanger final design.

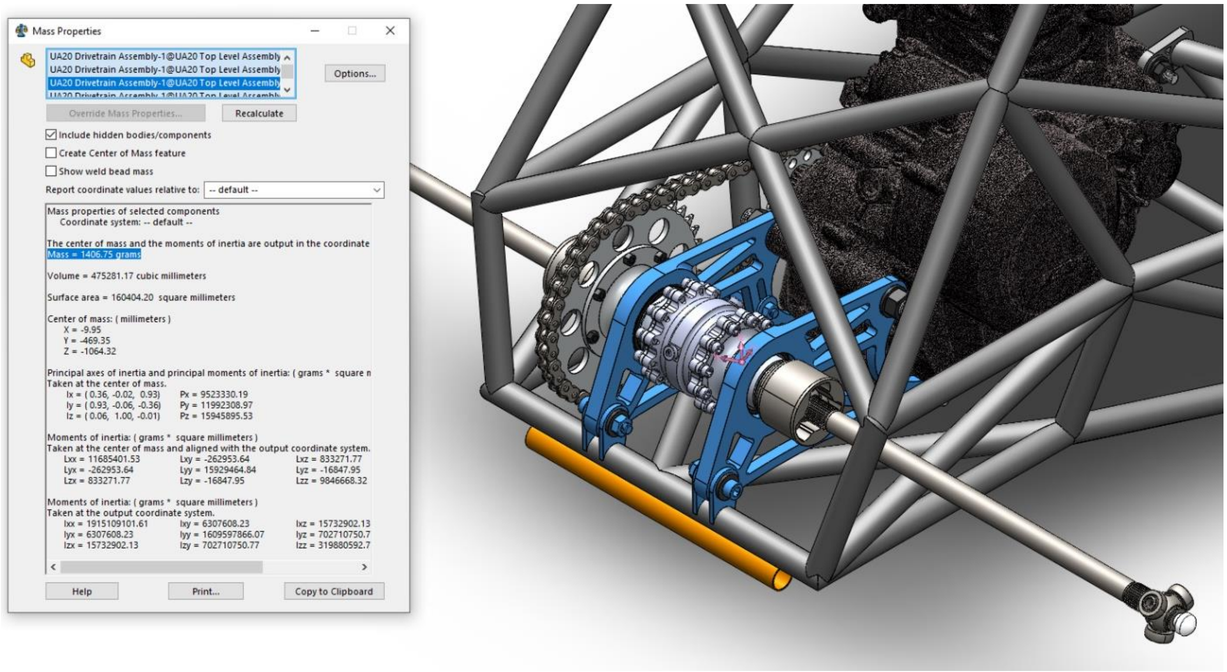


Figure 15: Mass analysis done in SolidWorks displaying drivetrain mounting system to be 1406.75 g.

Cost Analysis

Formula SAE competition utilizes a cost analysis program used to estimate the cost of mass production for parts on each vehicle based on material and machining operations. Although this cost analysis does not accurately portray how much parts will debt the team to fabricate, it provides a detailed and systematic way to compare parts across different vehicles. In the UA-19 FSAE cost report, the differential hangers, engine mounts and chain tensioning system costed the team $75 to mass produce. Using the same method to input the UA-20 designs cost, the differential hangers which are also the engine mounts and the idler assembly cost $82. This is slightly more expensive than the previous design due to the increased machining operations on the differential hangers. Seeing that the UA-20 drivetrain budget is roughly the same as UA-19 and that the FSAE cost report value is not considerably more expensive this year, the UA-20 design has been deemed economically feasible. A more extensive analysis of cost and manufacturing operations required for the UA-20 design is shown in appendix G.

Table 3: Future Work Project Schedule

Task	Expected Completion Date	Criteria for Evaluating Success
Investigate the need for cross bracing on the differential hangers to prevent the two plates from parallelograming due to engine loading.	December 20, 2019	If stress concentration is deemed too high on rear mounting tabs due to added engine loading, cross bracing will be designed.

3D print prototype differential hangers for engine mock-up and jigging when chassis fabrication begins.	December 20, 2019	3D printed parts are an accurate representation of final machined product and aids in the welding and assembly of chassis tubes.
Waterjet cut chassis mounting tabs.	December 20, 2019	Mounting tabs fix differential hangers and engine in desired location.
Manufacturer differential hangers, idler sprocket shaft and all sprockets.	March 31, 2020	Manufactured parts satisfy drawing dimensions and any critical tolerances specified.
Assemble entire drivetrain assembly.	April 10, 2020	Differential spins freely within differential hangers, chain fits around sprockets, idler sprocket provides adequate pre-tension on the chain.
Design, manufacture and assemble chain scatter shield to minimum rules requirement.	April 10, 2020	Chain scatter shield meets rules requirements and does not interfere with normal operation of the chain and sprockets.
Test drivetrain assembly on track.	May 1, 2020	Power is reliably transferred to the rear sprocket and differential with minimum visual deflection in the differential hangers. Idler sprocket maintains chain tension over time and use.

References

"Formula SAE Rules 2020" Formula SAE, SAE International, 25 July 2019, pp. 61-62 and pp. 65-66, https://www.fsaeonline.com/cdsweb/gen/DocumentResources.aspx

"FSAE Brochure" RCV Performance Products, 29 April 2019, pp. 3, https://www.rcvperformance.com/media/fsae/FSAE_Brochure.pdf

Appendix A
Design Compliance Matrix

Table A1: Updated Design Compliance Matrix

	Design Authority	Specification	Priority	Within Scope of Concept	Reasons Why Concept Does Not Comply	Compliant with Final Design	Final Design Compliance Notes
Chain Drive	Formula SAE: T.5.2.7	All chain driven vehicles must have a shield in case of failure. Chain scatter shield must be made of 0.105" minimum thickness steel and a minimum width equal to three times width of the chain.	5	Yes		Yes	Chain scatter shield is a future work project.
	Formula SAE: T.5.2.9	Chain scatter shield must be mounted using 6 mm or 0.25" minimum diameter critical fasteners.	5	Yes		Yes	
	Formula Racing Team	Due to chain stretching during use and the nature of assembly, design must provide a manual chain tensioning method so that the chain is at the necessary preferred tension.	5	Yes		Yes	UA-20 final design uses an idler sprocket to tension the chain.
	Formula Racing Team	Design should allow the chain and chain tensioner to be assembled and adjusted in <2 hours for easier serviceability at competition and in time limited scenarios.	1	Yes		Yes	Idler sprocket requires 2 bolts to be adjust chain tension, UA-19 eccentric disks required 16 bolts.
Fasteners	Formula SAE: T.8.2.1	Although drivetrain is not explicitly mentioned in this rule, the loads being applied are significant enough that fasteners should meet or exceed SAE Grade 5 or Metric Grade 8.8.	5	Yes		Yes	
	Formula SAE: T.8.3.(1,2)	Although drivetrain is not explicitly mentioned in this rule, positive locking mechanisms should be used to prevent catastrophic failure due to fasteners coming loose.	5	Yes		Yes	
Strength	Formula Racing Team	Must be able to support minimum 2 g lateral and longitudinal engine loads in acceleration/braking and cornering. Differential mounts must also	5	Yes	With potential engine horsepower gains and reduced sprocket sizes, maximum chain	Yes	

		withstand a maximum ~~9600 N~~ 12,300 N of chain force, based on ~~UA-19~~ UA-20 horsepower, torque and final gearing ratio.			force has increased to 12,300 N.		
Differential	Formula Racing Team	Differential hangers must be designed to house a Drexler FSAE differential and allow it to freely spin.	5	Yes		Yes	Differential bearings have been selected and will last at least the UA-20 season.
Weight	Formula Racing Team	Differential/Engine mounts to be at minimum 10% less weight than total UA-19 drivetrain and engine mounting assembly (2100 g).	4	Yes		Yes	UA-20 drivetrain mounting solution is 33% lighter than the UA-19 design.
Driveshafts	RCV Performance	Design differential mount so that RCV Performance driveshafts have less than 12˚ of deviation in all planes as dictated by driveshaft supplier. As driveshaft angles approach 0˚, reliability and performance both increase. UA-19 vehicle had 9˚ of driveshaft angle increasing component wear and the possibility of a CV joint failure.	4	Yes		Yes	Final design places differential centerline closer to the wheel centers decreasing driveshaft angles to 2.9°.
Chain and Sprocket Clearance	Formula Racing Team	Chain must not drag on chassis or other components under normal operation and a range of sprocket sizes (Ø177.1 mm – Ø202.3 mm) must clear the chassis.	~~5~~ 2	No	Large range of sprockets for tuning is no longer a critical specification. The same powertrain as last year is being used and enough data has been gathered to narrow down a final gear ratio.	No	Final design only allows for one tooth of sprocket adjustment front and rear.
Vibration Dampening	Formula Racing Team	Engine mounts must be soft-mounted to provide vibration dampening. In previous years, engine was hard mounted to chassis causing vibrations throughout entire vehicle and discomfort to driver.	3	Yes		Yes	Drivetrain is mounted to the chassis with polyurethane bushings to reduce vibrations.

Appendix B
Preliminary Concept Finite Element Analysis

The purpose of this preliminary FEA study is to validate that mounting the UA-20 concept to the chassis and engine is a more rigid method than the UA-19's cantilevered method which has only two mounting points secured solely to the chassis.

In pursuance of making the FEA results comparable between both the UA-19 design and UA-20 concept, both models were assigned the same ½" thick 7075-T6 aluminum alloy material (same material the UA-19 differential hangers were manufactured from). Weight saving topology results from the UA-19 design was also eliminated – pictured in figure B1 – to give a fair comparison between both models. The eccentric disk in the UA-19 design is rigidly bonded to the differential hanger, shown in figure B1, as it contributes to the total rigidity of the assembly.

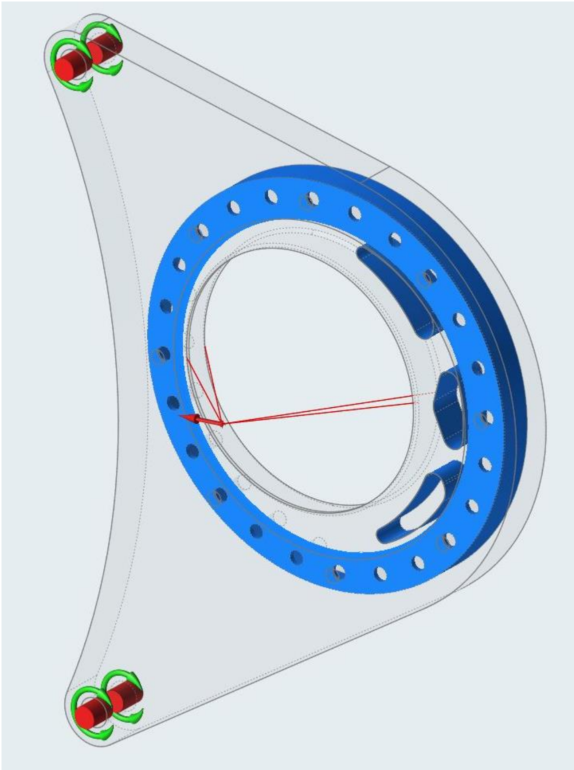


Figure B1: Blue highlighted region signifies a rigidly bonded connection between two parts.

Figure B2 shows the loads and constraints applied to the UA-19 differential hanger. The part is pin connected in two places representing the two bolts that fasten the differential hanger to the chassis tabs welded to the rear chassis tubes. An arbitrary load of 24000 N is applied to the center of the differential bearing bore and is offset outside of the part 38.25 mm to represent the chain tension. Offsetting the force simulates the moment created by the chain tension on the differential hanger. 24000 N was arbitrary chosen as a chain force as it represents approximately 50% tensile strength of a AISI 520 series chain. Deformation results can be seen in figure B3.

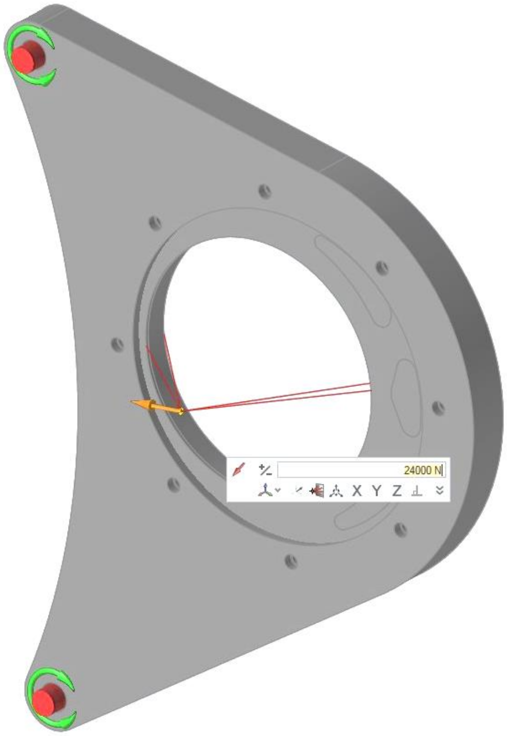


Figure B2: Constraints and load applied to UA-19 differential hanger design.

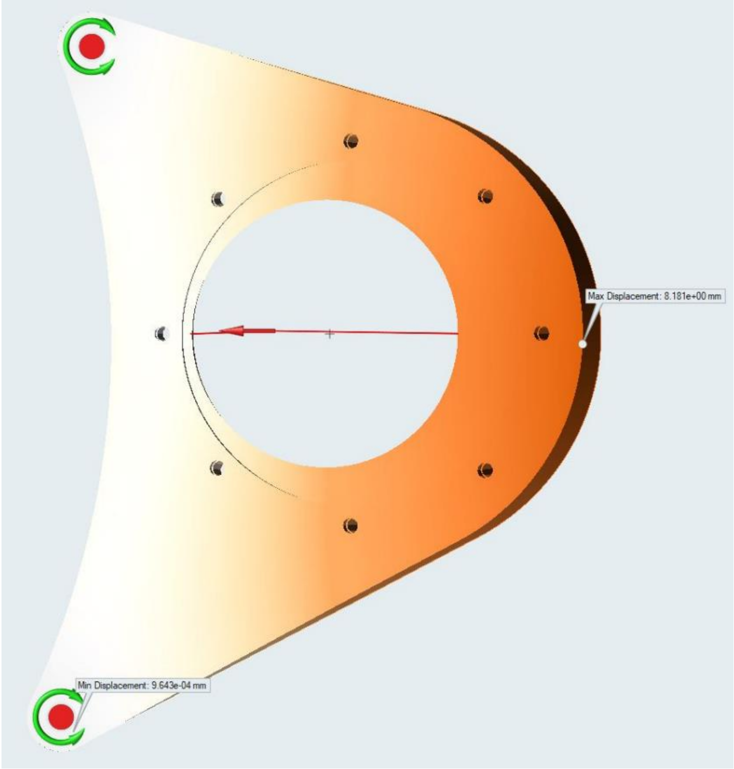


Figure B3: UA-19 design deformation results. Maximum displacement of $8.181 \cdot 10^0$ mm.

Figure B4 shows the loads and constraints applied to the UA-20 differential hanger concept. The part is rigidly pin connected in two places representing the two bolts that fasten the differential hanger to the rear engine mounting points. The part is also pin connected in one place representing the bolt that fastens the differential hanger to the tabs welded to the rear chassis tube. A load of 24000 N is applied to the center of the differential bearing bore and is offset outside of the part 38.25 mm exactly as conducted on the UA-19 differential hangers. Deformation results can be seen in figure B5.

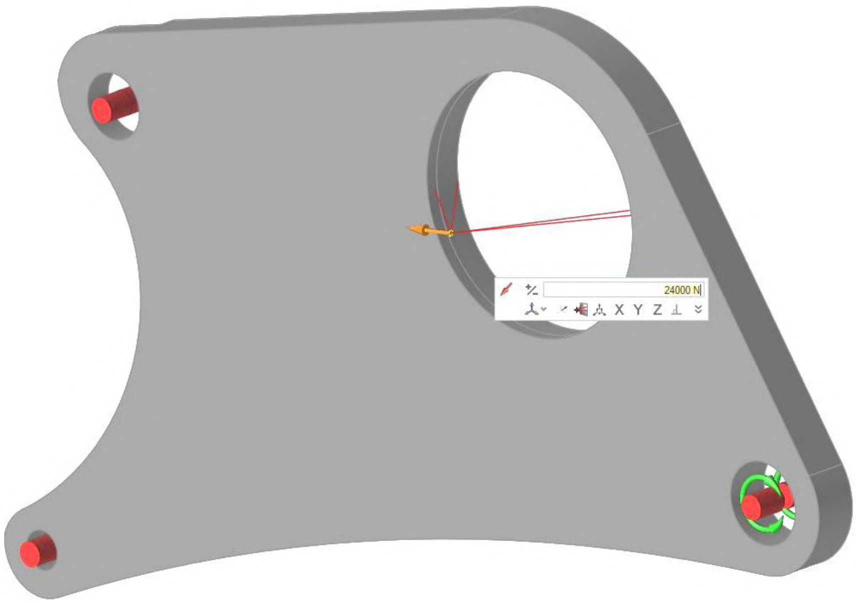


Figure B4: Constraints and load applied to UA-20 differential hanger concept.

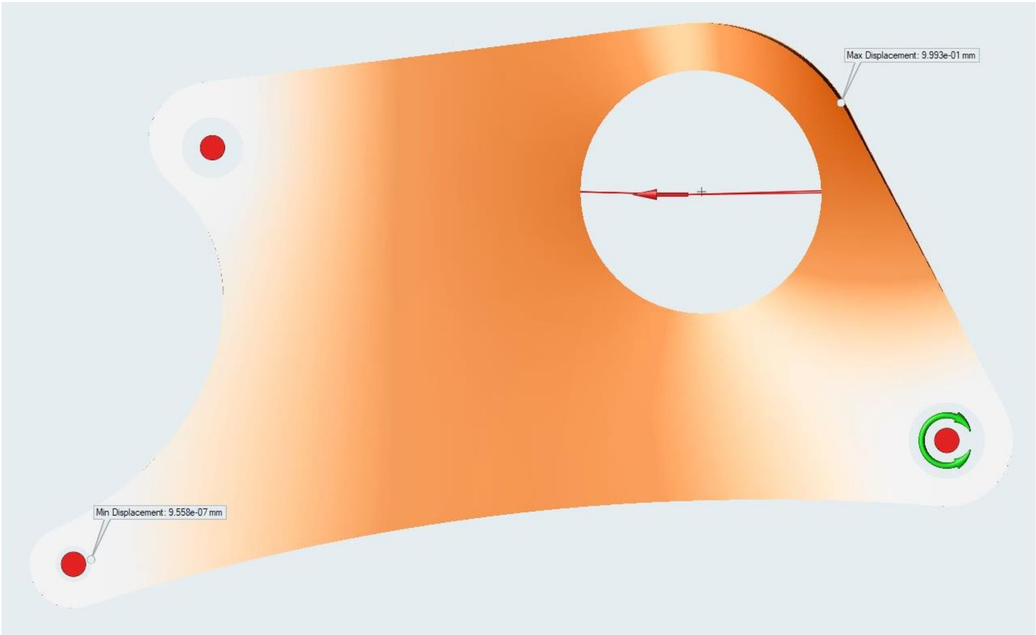


Figure B5: UA-20 concept deformation results. Maximum displacement of $9.993 \cdot 10^{-1}$ mm.

Appendix C
Concept Differential Hanger Weight Analysis

Weight analysis of the UA-19 design and UA-20 design was conducted using SolidWorks mass evaluations to compare both designs and attempt to validate a 10% weight reduction as per the design specifications.

Figure C1 shows the total mass of 2119.64 g for all UA-19 drivetrain mounting components (everything highlighted in blue is being evaluated). The rear engine mounts and hardware were also included in the mass evaluation as the UA-20 concept combines these with the differential hangers. All mounting hardware, and the chassis tabs were also included in the mass calculation.

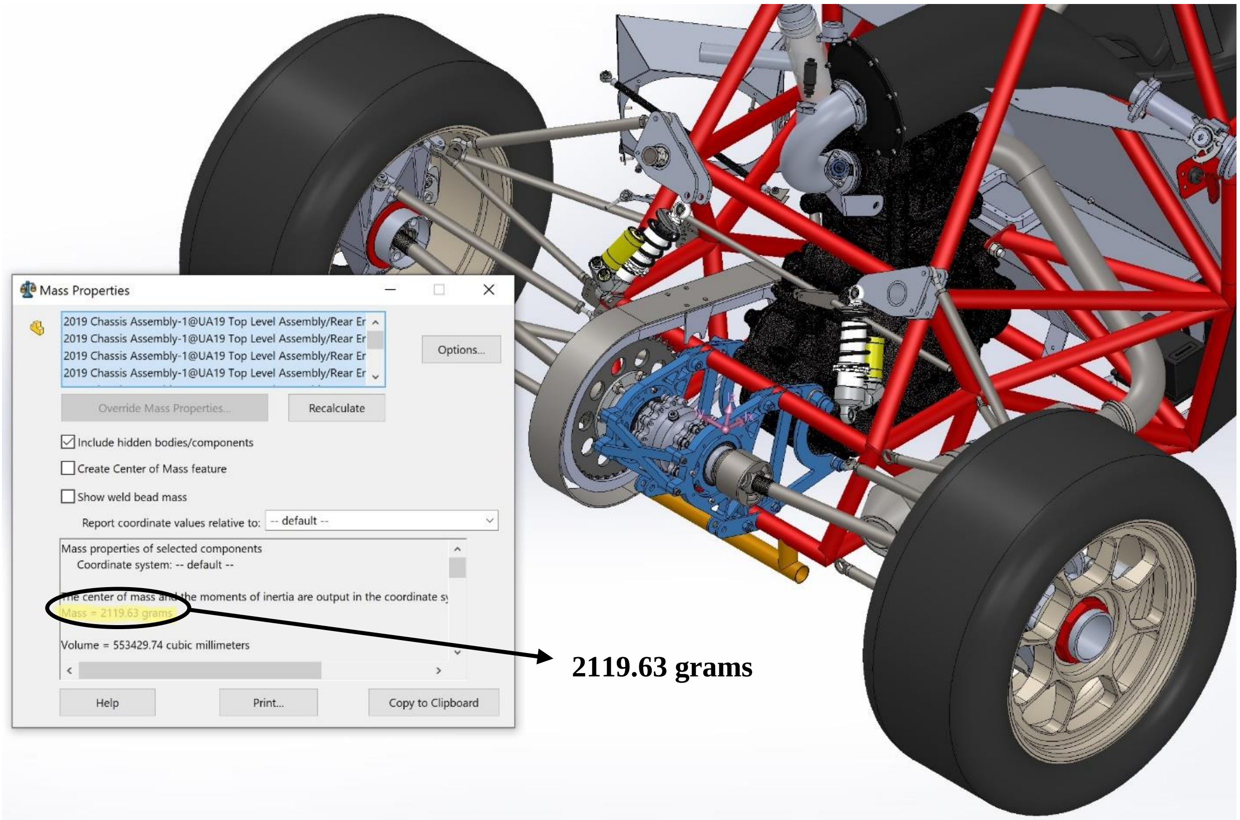
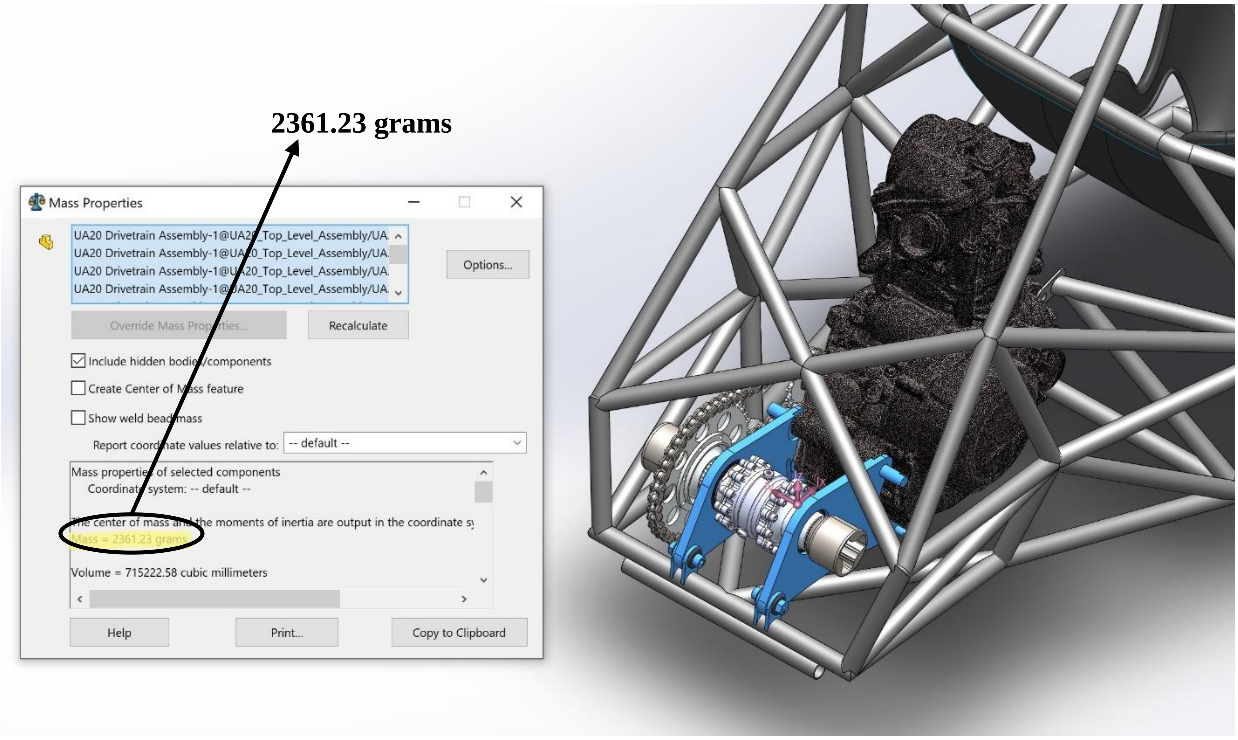


Figure C1: UA-19 drivetrain design SolidWorks mass evaluation.

Figure C2 shows the total mass of 2361.23 g for all UA-20 drivetrain mounting components (everything highlighted in blue is being evaluated). Mass evaluations were done with the same AISI 4130 steel chassis tabs, the same AISI 4140 rear engine bolts, and metric 12.9 grade fasteners to give a fair comparison between the UA-19 design and UA-20 concept.



<u>Figure C2:</u> UA-20 drivetrain concept SolidWorks mass evaluation.

It is evident the UA-20 concept weighs approximately 250 g more than the UA-19 design, however it is important to note that the UA-19 differential hangers, eccentric disks and engine mounts were topologically optimized to minimize weight. Figure C3 shows an example on how the UA-19 differential hangers were optimized, and weight was reduced by approximately 25%.

By going through the same process with the UA-20 concept in the final design phase we expect to reduce the weight of the differential hangers by a conservative 20% while maintaining the structural integrity it requires. By doing so, the UA-20 differential hangers would lose approximately 350 g and result in the full assembly weighing 2000 g. If in the final design phase, topology and FEA studies allow for a 25% reduction of mass in the differential hangers, the UA-20 drivetrain assembly will exceed the 10% weight reduction target proposed in the design specifications.

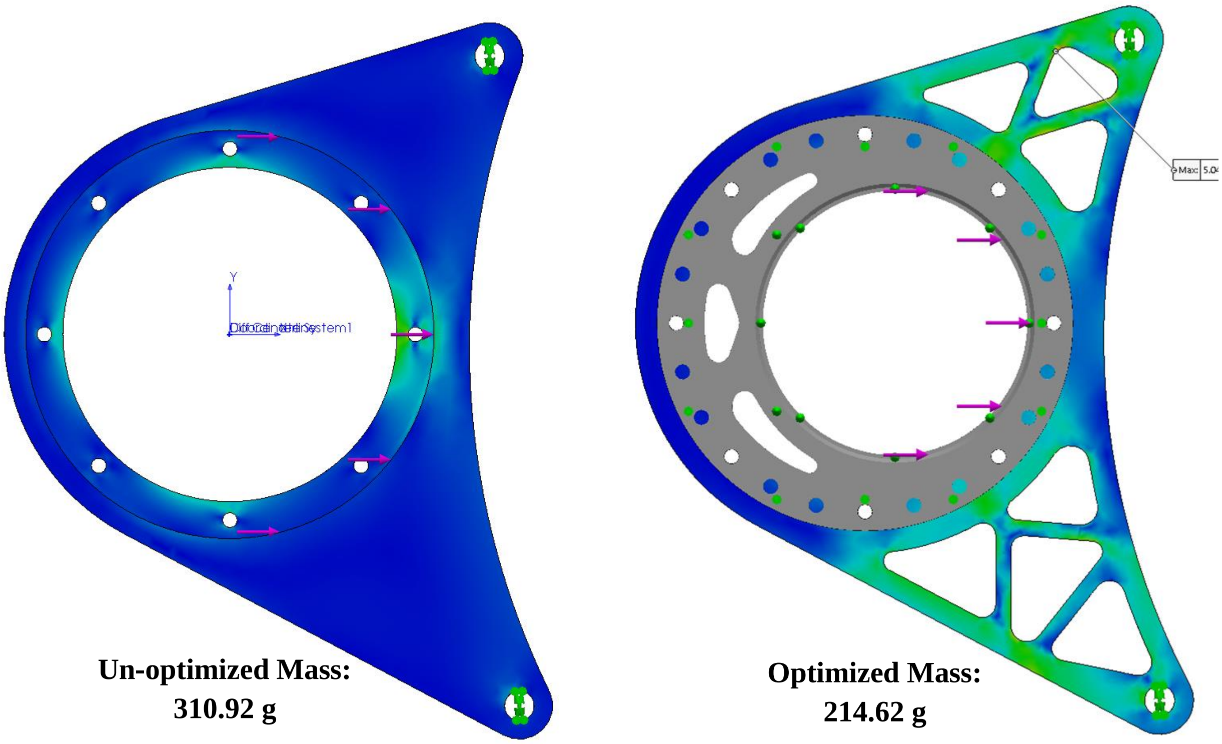


Figure C3: UA-19 differential hangers highlighting the weight saving results from topological optimization.

Appendix D
Differential Hanger Loading Calculations

Section 1 – Chain Tension

Main loading applied to the differential hangers is the chain tension which acts to pull the rear 37-tooth sprocket fixed to the differential towards the 14-tooth sprocket fixed to the engine. The calculations outlined below use engine horsepower values extrapolated from the UA-19 KTM 690 dynamometer data. To account for the UA-20 potential increase in power, horsepower data from the UA-19 KTM 690 dynamometer has been increased by a generous 20% shown in table D1. All constants used in the chain tension calculations are listed in figure D1.

	(DYNO)
Engine Speed (RPM)	Power (hp)
11500	
11000	
10500	
10000	
9500	
9000	
8500	54
8000	60
7500	64
7000	62
6500	61
6000	59
5500	55
5000	52
4500	48
4000	44
3500	38
3000	32
2500	26
2000	17
1500	11
1000	

Table D1: UA-20 KTM 690 projected horsepower figures based on UA-19 dynamometer results.

Constants

Gearing:

$prime := 2.194$
$first := 2.500$
$second := 1.74$
$third := 1.333$
$fourth := 1.095$
$fifth := 0.957$
$sixth := 0.870$

Note: From stock KTM 690 transmission.

$N_{rear} := 37$

$d_{rear} := 187.1929\ mm$

$N_{front} := 14$

$d_{front} := 71.3409\ mm$

$d_{tire} := 520.7\ mm$

$$final := \frac{N_{rear}}{N_{front}} = 2.643$$

Figure D1: Constant values used in chain tension calculation.

Figure D2 shows a free body diagram of the chain, sprockets and some of the constants highlighted above.

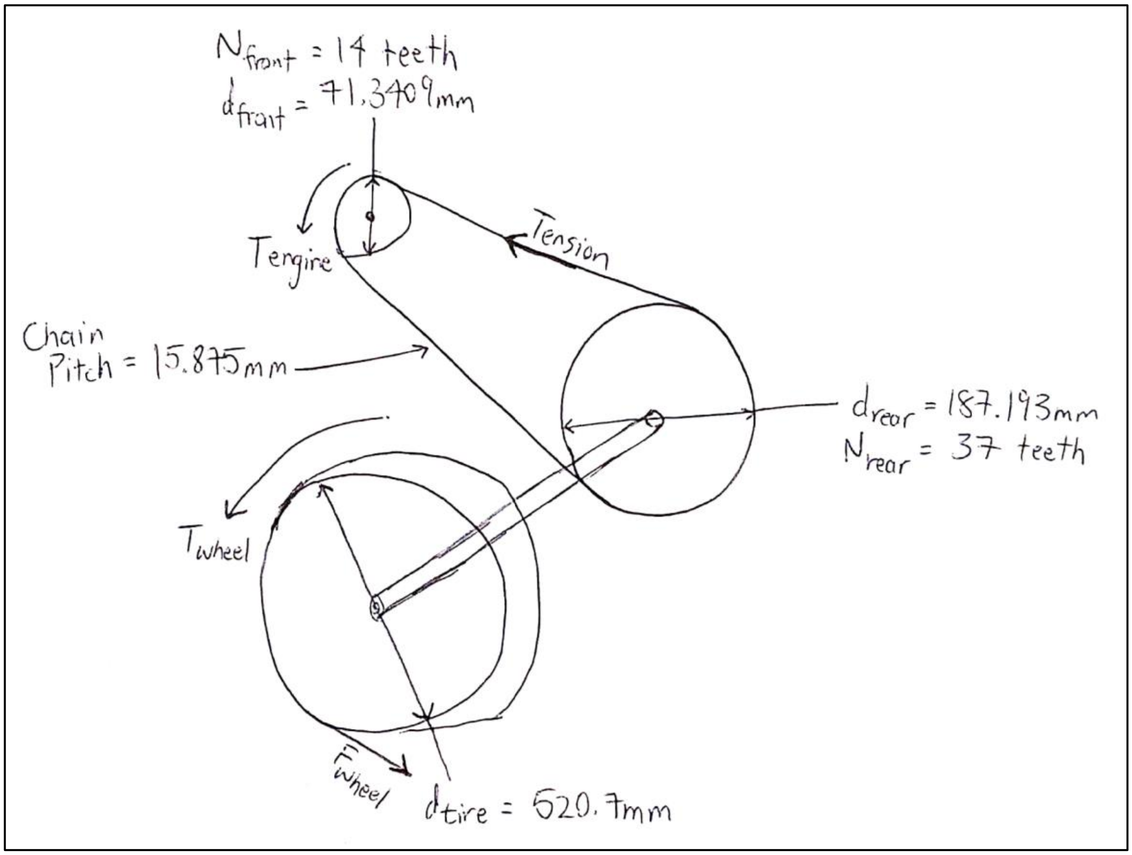


Figure D2: Free body diagram of chain, sprockets and wheel.

Chain tension sample calculations shown below are for 1st gear at 4000 RPM and 44 hp as expressed in figure D3.

Sample Calculations in first gear at 4000 RPM and 44 hp:

$$power := 44\ \mathbf{hp} \qquad speed := 4000\ \frac{\mathbf{rev}}{\mathbf{min}}$$

Figure D3: Engine power and speed variables for sample calculation.

Engine Torque:

$$T_{engine} := 5252.113\ \frac{\mathbf{rpm}}{\mathbf{hp}} \cdot \mathbf{lbf} \cdot \mathbf{ft} \cdot \frac{power}{speed} = 57.773\ \mathbf{ft} \cdot \mathbf{lbf}$$

Figure D4: Sample calculation for engine torque.

Wheel Torque:

$$T_{wheel} := T_{engine} \cdot prime \cdot first \cdot final = 837.485\ \mathbf{ft} \cdot \mathbf{lbf}$$

Figure D5: Sample calculation for wheel torque.

Wheel Force:

$$F_{wheel} := \frac{T_{wheel}}{0.5 \cdot d_{tire}} = \left(4.361 \cdot 10^{3}\right)\ \mathbf{N}$$

Figure D6: Sample calculation for wheel force.

Chain Tension:

$$Tension := F_{wheel} \cdot \frac{d_{tire}}{d_{rear}} = \left(1.213 \cdot 10^{4}\right)\ \mathbf{N}$$

Figure D7: Sample calculation for chain tension.

* 5252.113 is a well-known constant value used to calculate engine torque:

$$1hp = (3.3 \cdot 10^4)\frac{ft \cdot lbf}{min} = power\ \textbf{or}\ work\ per\ minute$$

$$2 \cdot \pi \cdot RPM = distance\ traveled\ per\ minute$$

$$Power = \frac{Torque}{\cancel{Radius}} \cdot \cancel{Radius} \cdot 2 \cdot \pi \cdot RPM \qquad and \qquad hp = \frac{Power}{(3.3 \cdot 10^4)}$$

$$\therefore hp = \frac{Torque \cdot 2 \cdot \pi \cdot RPM}{(3.3 \cdot 10^4)} = \frac{Torque \cdot RPM}{5252.113}$$

These calculations were carried out for each RPM range and corresponding engine horsepower through all 6 gears of the KTM 690 stock transmission. All calculated values are shown in table D2.

<u>Table D2:</u> Calculated wheel torque, wheel force and chain tension at each RPM range and all six KTM 690 transmission gears.

	(DYNO)		**Wheel Torque (*ft·lbf*)**						**Wheel Force (N)**						**Chain Tension (N)**					
Engine Speed (*RPM*)	Power (*hp*)	Engine Torque (*ft·lbf*)	1st Gear	2nd Gear	3rd Gear	4th Gear	5th Gear	6th Gear	1st Gear	2nd Gear	3rd Gear	4th Gear	5th Gear	6th Gear	1st Gear	2nd Gear	3rd Gear	4th Gear	5th Gear	6th Gear
11500		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
11000		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
10500		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
10000		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
9500		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
9000		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
8500	54	33.37	484	339	258	212	185	168	2519	1763	1343	1103	964	877	7006.5	4904.6	3735.9	3068.8	2682.1	2438.2
8000	60	39.39	571	400	304	250	219	199	2974	2082	1586	1302	1138	1035	8271.6	5790.1	4410.4	3622.9	3166.4	2878.5
7500	64	44.54	646	452	344	283	247	225	3362	2354	1793	1473	1287	1170	9352.4	6546.7	4986.7	4096.3	3580.1	3254.6
7000	62	46.82	679	475	362	297	260	236	3534	2474	1885	1548	1353	1230	9831.3	6881.9	5242.1	4306.1	3763.4	3421.3
6500	61	49.45	717	502	382	314	274	249	3733	2613	1990	1635	1429	1299	10384.0	7268.8	5536.7	4548.2	3975.0	3613.6
6000	59	51.47	746	522	398	327	286	260	3886	2720	2072	1702	1487	1352	10808.2	7565.7	5762.9	4734.0	4137.4	3761.2
5500	55	52.71	764	535	407	335	293	266	3979	2785	2122	1743	1523	1385	11068.9	7748.2	5901.9	4848.2	4237.2	3851.9
5000	52	54.20	786	550	419	344	301	273	4092	2864	2182	1792	1566	1424	11381.7	7967.2	6068.7	4985.2	4356.9	3960.8
4500	48	56.02	812	568	433	356	311	283	4229	2960	2255	1852	1619	1472	11764.0	8234.8	6272.6	5152.6	4503.3	4093.8
4000	44	57.77	837	586	447	367	321	291	4361	3053	2325	1910	1670	1518	12131.6	8492.1	6468.6	5313.6	4644.0	4221.7
3500	38	57.62	835	585	445	366	320	291	4350	3045	2319	1905	1665	1514	12100.1	8470.1	6451.8	5299.9	4631.9	4210.8
3000	32	56.72	822	576	438	360	315	286	4282	2997	2283	1876	1639	1490	11911.1	8337.7	6351.0	5217.0	4559.6	4145.0
2500	26	55.46	804	563	429	352	308	280	4187	2931	2232	1834	1603	1457	11646.4	8152.5	6209.8	5101.1	4458.2	4052.9
2000	17	44.12	640	448	341	280	245	223	3330	2331	1776	1459	1275	1159	9264.2	6484.9	4939.6	4057.7	3546.3	3223.9
1500	11	37.82	548	384	292	240	210	191	2855	1998	1522	1250	1093	993	7940.7	5558.5	4234.0	3478.0	3039.7	2763.3
1000		0.00	0	0	0	0	0	0	0	0	0	0	0	0	0.0	0.0	0.0	0.0	0.0	0

Section 2 – Differential Hanger Reaction Forces

Analysis and calculations of the reaction forces present on the differential hangers as a result of the chain tension is shown below. A free body diagram of the system is shown in figure D8. The Drexler differential is assumed to be a rigid beam supported at both differential bearings.

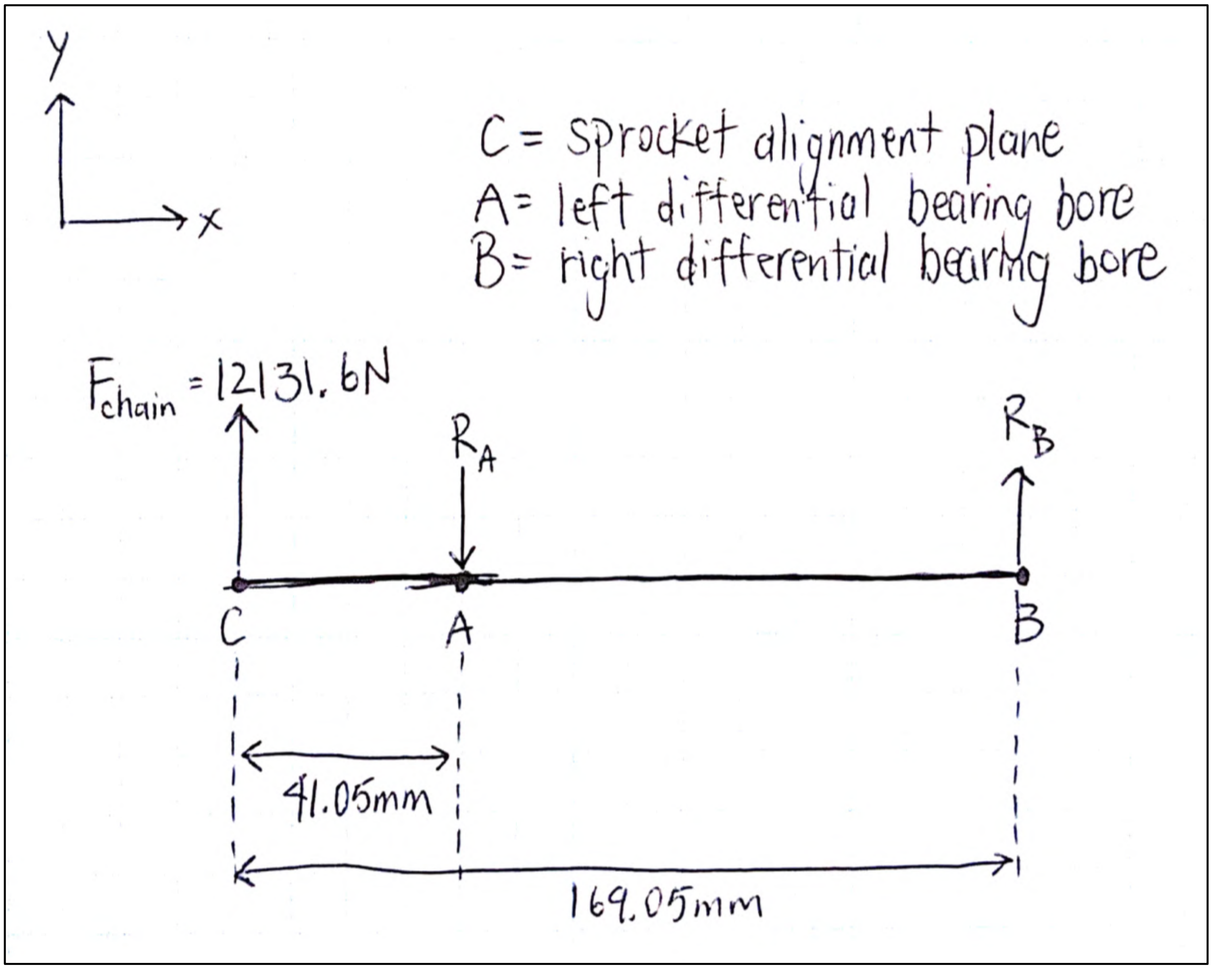


Figure D8: Free body diagram showing the maximum chain force and reaction forces at both differential hangers.

$$\sum F_y = 0 = (12131.6\ N) - R_A + R_B \qquad (1)$$

$$\sum M_B = 0 = (12131.6\ N) \cdot (169.05\ mm) - R_A \cdot (128\ mm) \qquad (2)$$

Solving (1) and (2):

$$\boxed{\begin{aligned} R_A &= 16022.2\ N \\ R_B &= 3890.6\ N \end{aligned}}$$

From the chain, left differential hanger sees 412% more force than the right differential hanger. A moment is also created by the chain force on the left differential hanger of 498 N·m which should be considered a major load.

Appendix E
Differential Hanger FEA Studies

Section 1 – Finite Element Analysis Pre-Processing

A baseline static finite element analysis study was conducted on the differential hanger geometry derived in the conceptual design phase – shown in figure E1. Interpretation of the baseline results aided in removing material where stress concentrations were low to save as much weight as possible. To conserve computing power and time, static studies are only conducted on the left differential hanger as it sees 4 times the load of the right differential hanger. The loads and constraints defined in SolidWorks Simulation were kept constant throughout all iterative simulations. The loads and constraints are further explained below.

Figure E1: Baseline study assembly which includes rear mounting tabs.

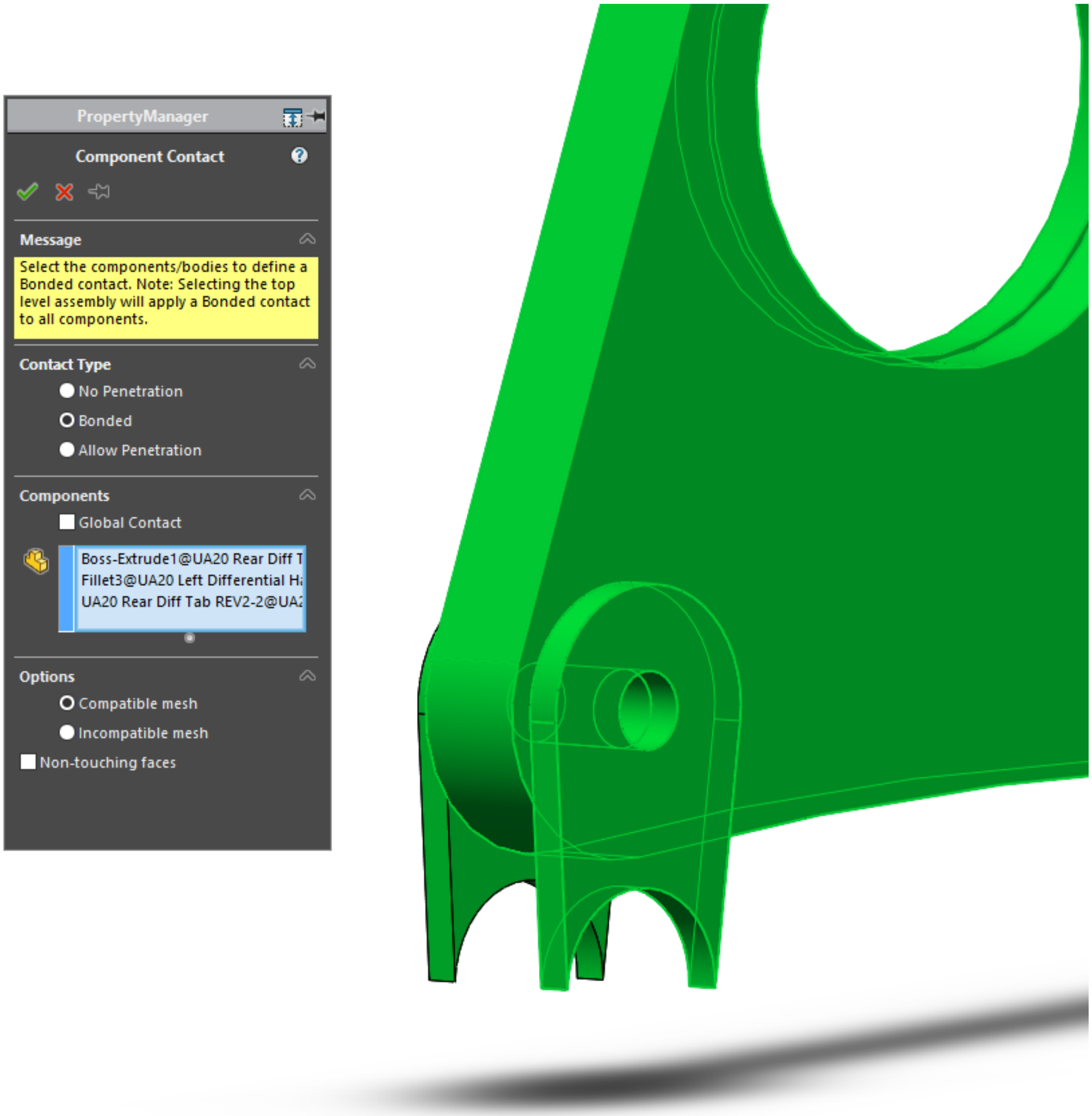
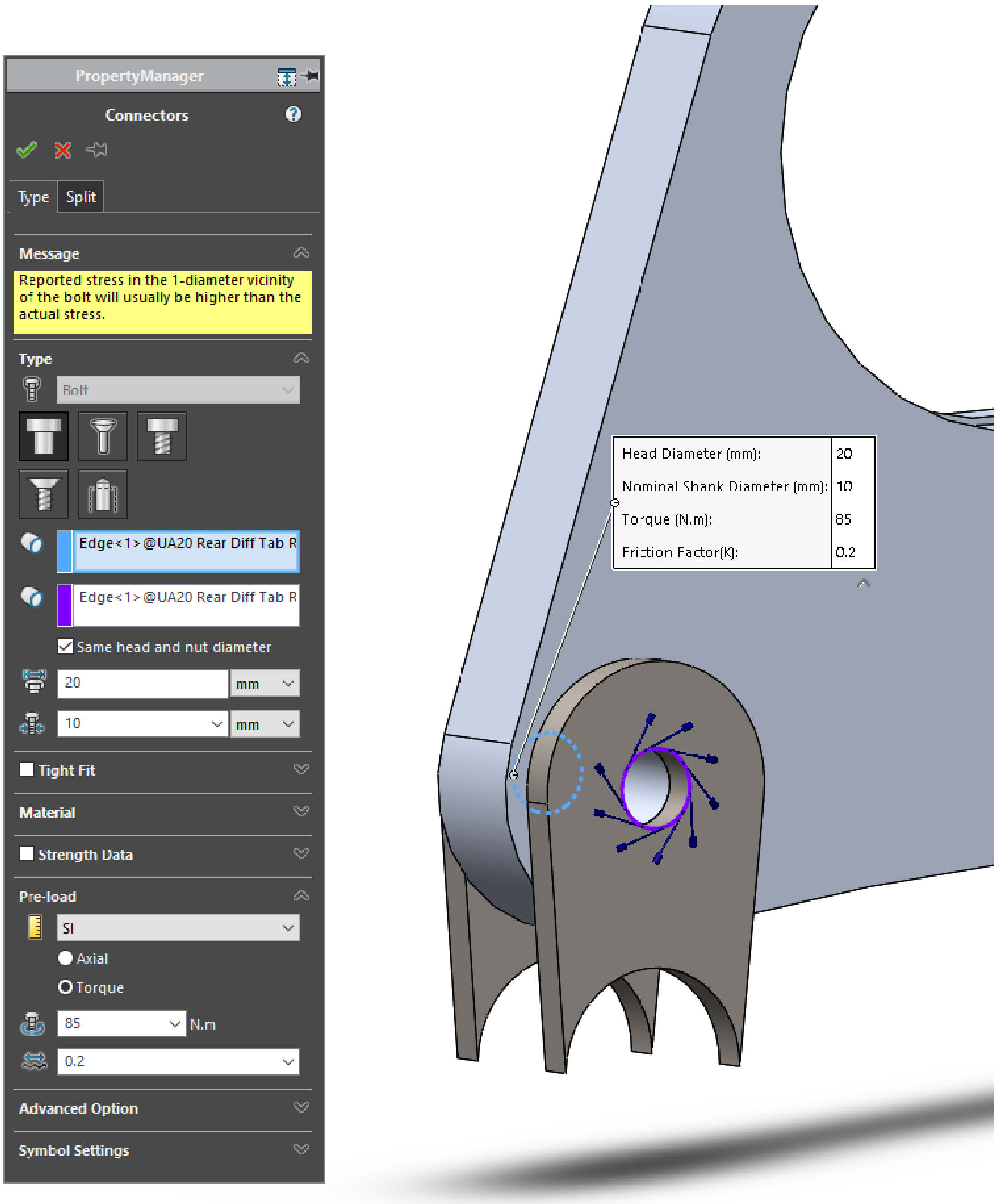


Figure E2: Component contact between differential hanger and tabs.

Contact constraint set between the two rear mounting tabs and the differential hanger is set to bonded. For study purposes, the bushing is eliminated from the rear mounting tabs. This should not create an issue as the bushing flange goes on the outside of the tabs in the drivetrain assembly, and the bolt head and nut compress the two tabs between the plate.



<u>Figure E3:</u> Nut and bolt connect through both mounting tabs.

M10 nut and bolt connector is applied to the outside edges of the differential mounting tabs. Bolt head and nut diameter is set to 20 mm to represent the presence of M10 washers.

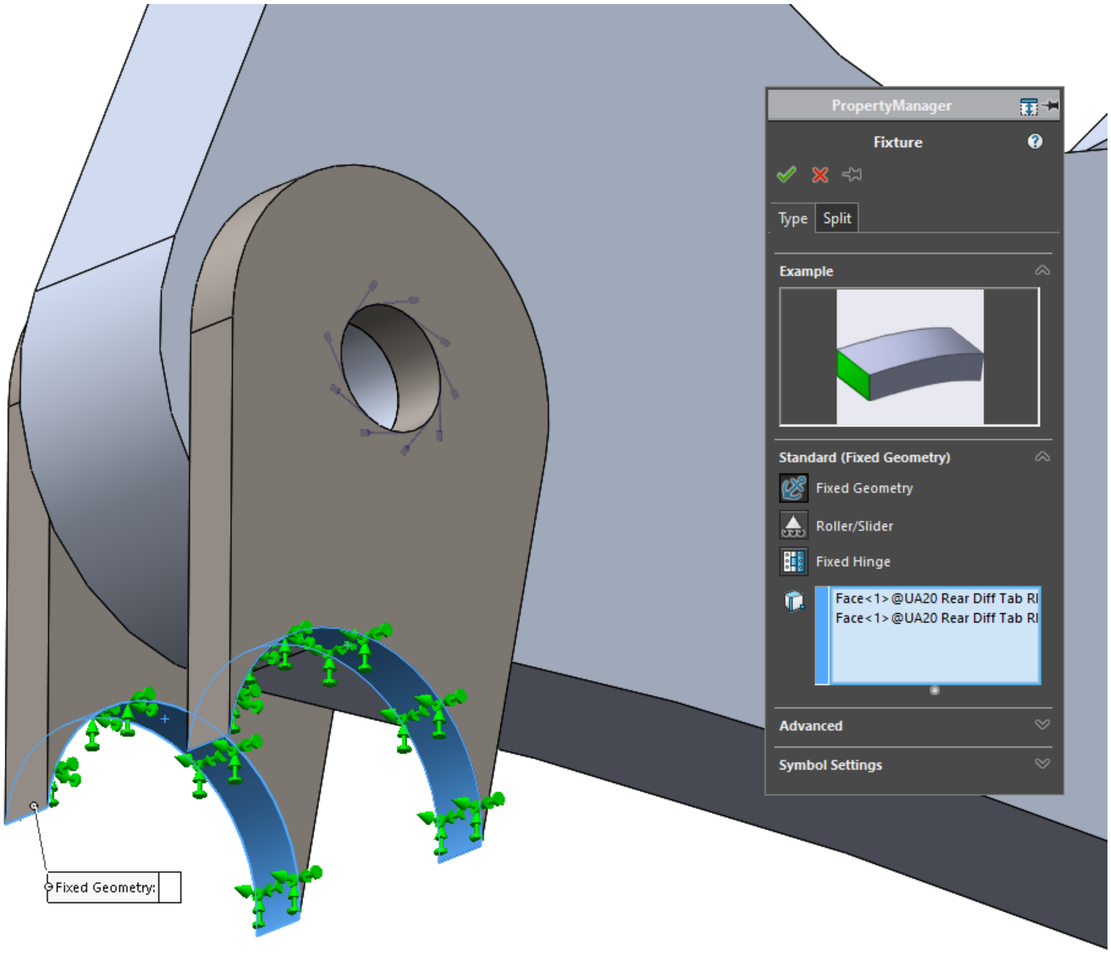


Figure E4: Fixed geometry mounting tabs.

The bottom faces of the mounting tabs are fixed in place to represent being welded to the rear chassis tube.

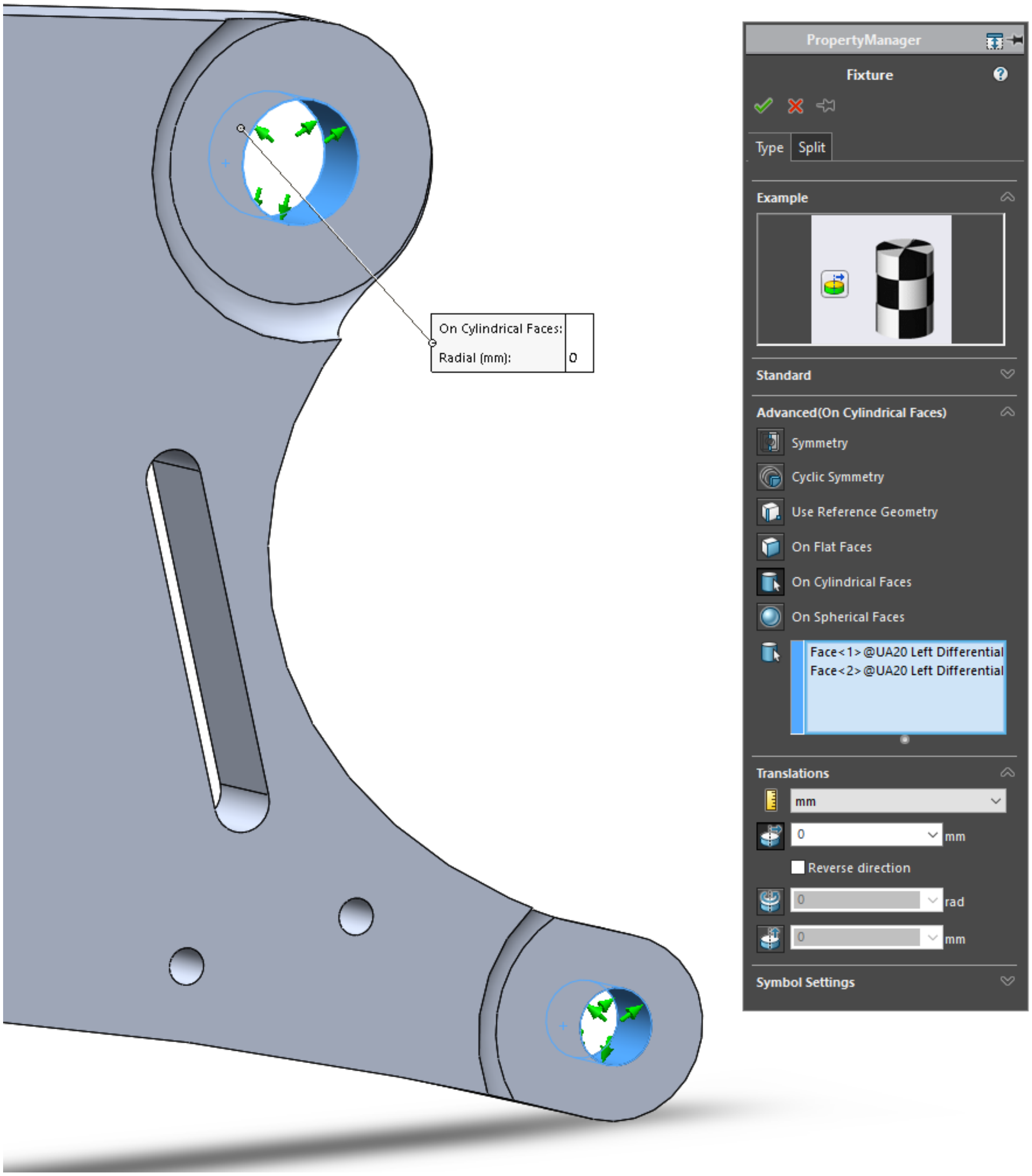


Figure E5: Advanced on cylindrical faces fixture to engine mount holes.

Advanced on cylindrical faces fixture is applied to the engine mount holes and a fixed radial translation constraint is applied. This simulates the bolt shank through the mounting holes and does not allow the cylinders to move radially from their respective axes.

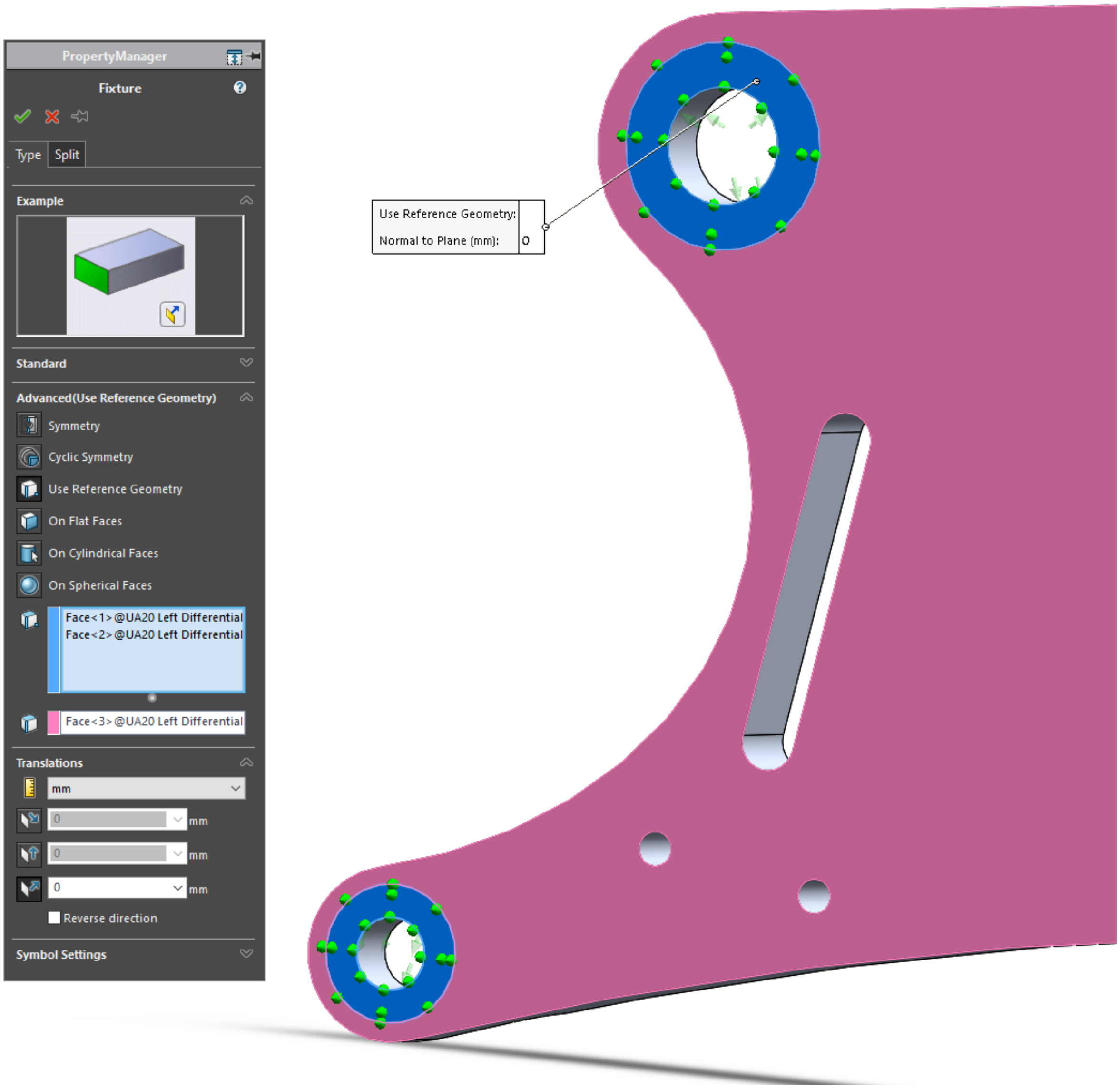
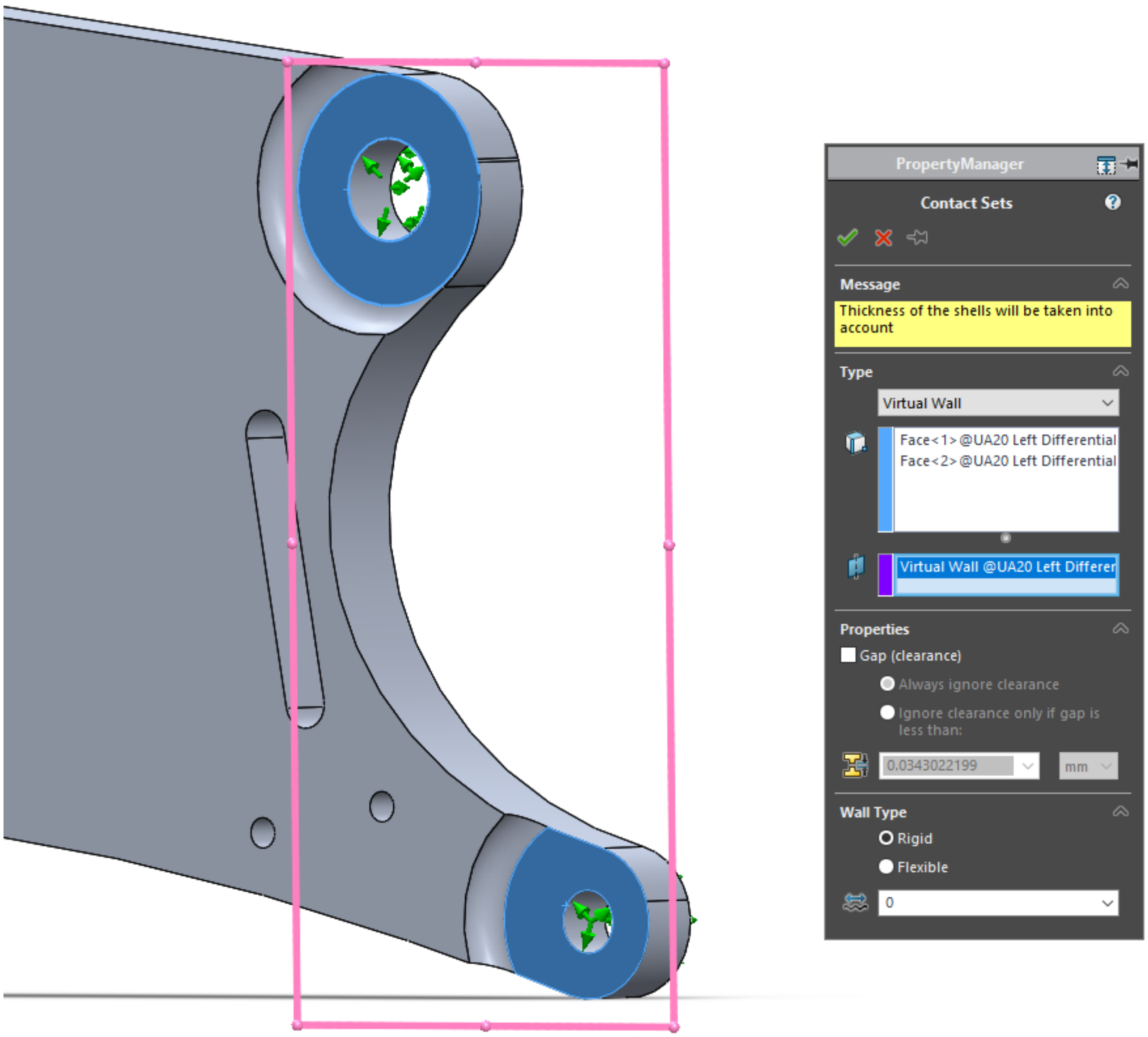


Figure E6: Advanced reference geometry fixture on washer split lines.

Advanced reference geometry fixture is applied to split line faces to represent the washer and nut that secures the differential hanger to the engine bolts. The selected split line faces are only fixed in place normal to the highlighted pink reference face.



<u>Figure E7:</u> Virtual wall contact set applied to engine contact regions.

A virtual wall at the differential hanger-engine contact interface is applied. The virtual wall is set to rigid to represent the engine and the differential hanger is constrained so that it cannot deform through the virtual wall.

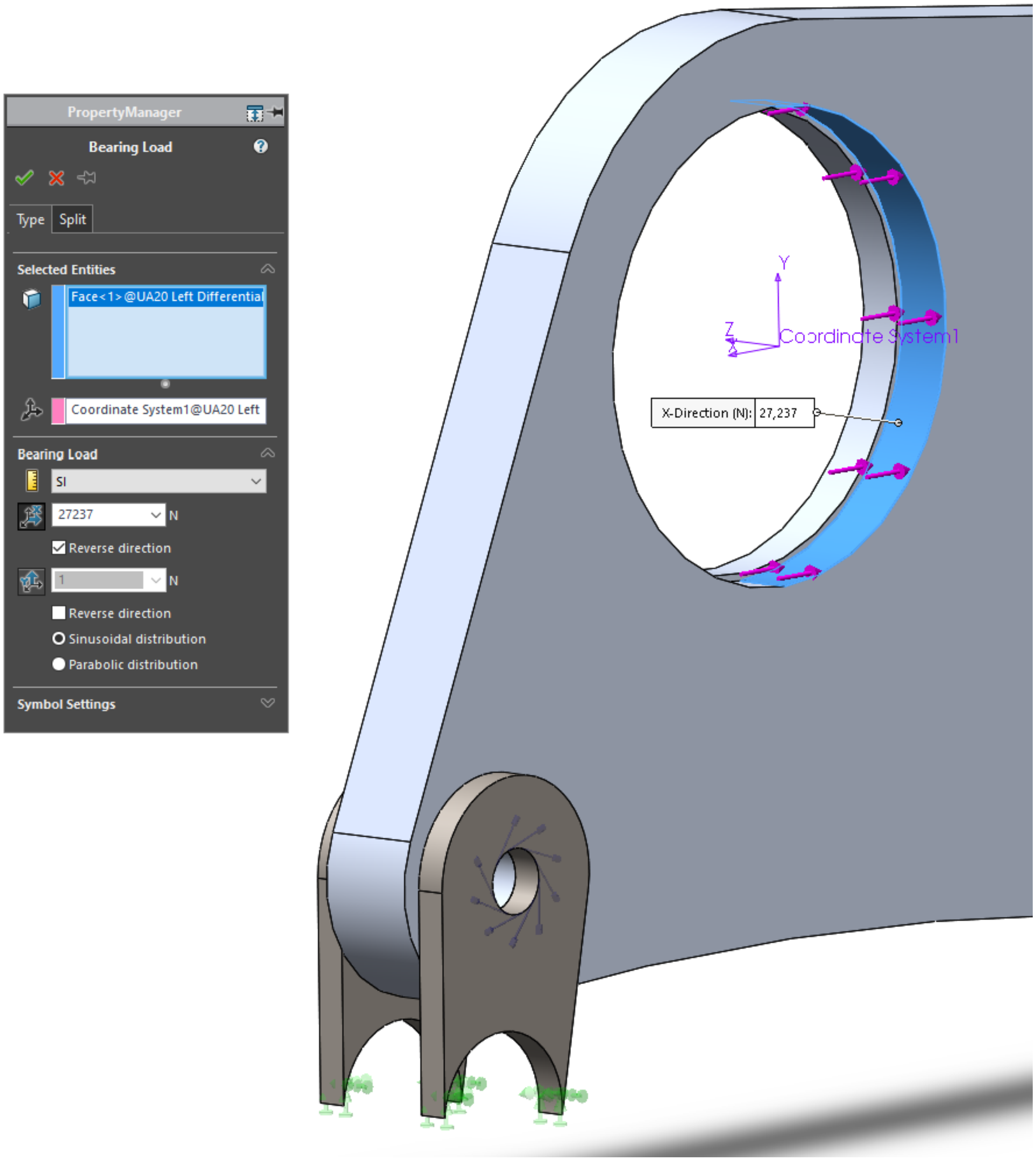


Figure E8: Bearing load.

Bearing load of 27,237 N is applied parallel to the chain force and to the inner bearing bore of the differential hanger. This bearing load is equal and opposite direction to the maximum reaction force calculated in appendix D, section 2 and is scaled to a factor of safety of 1.7 as per the teams UA-20 design guidelines for critical components.

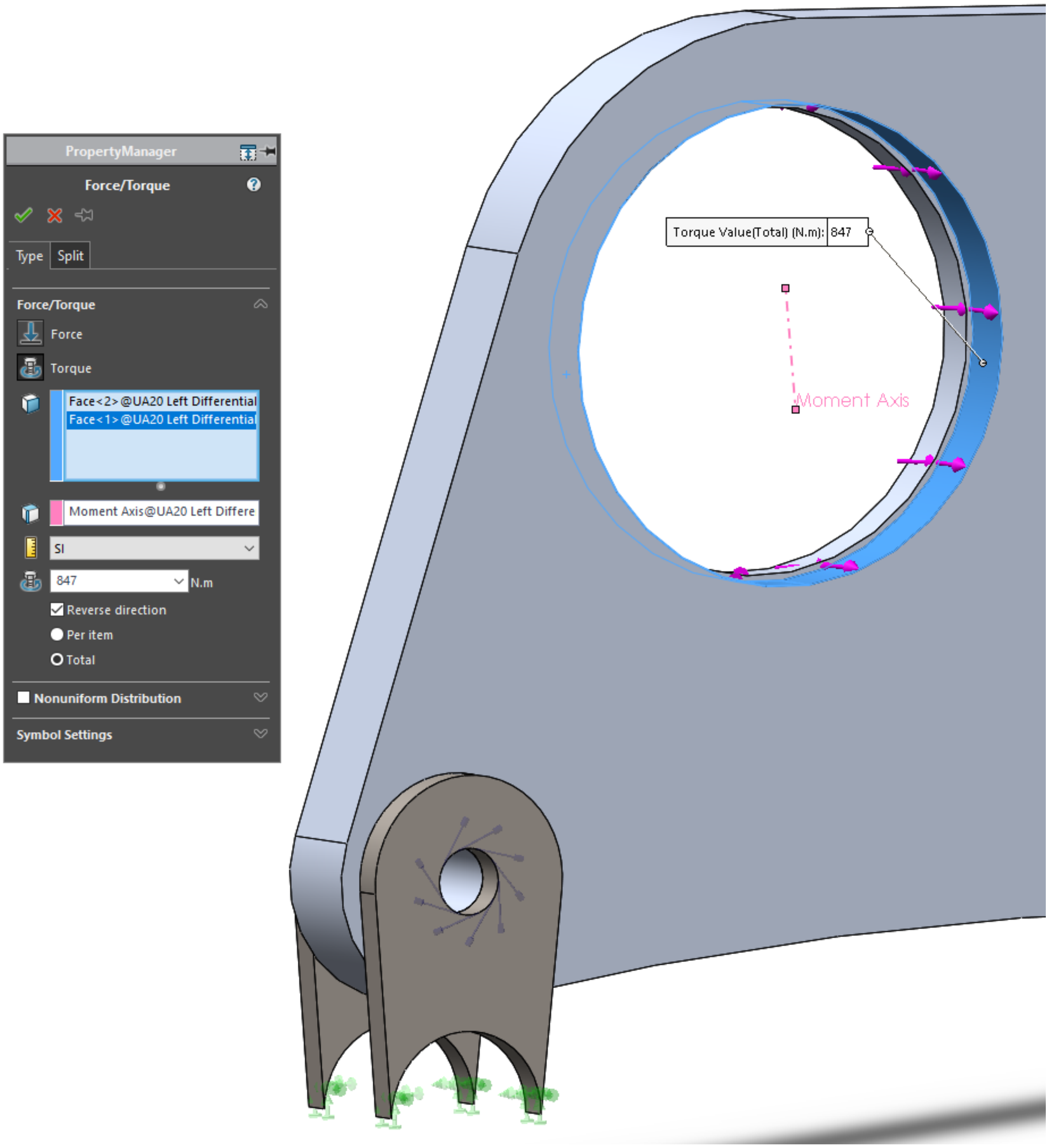
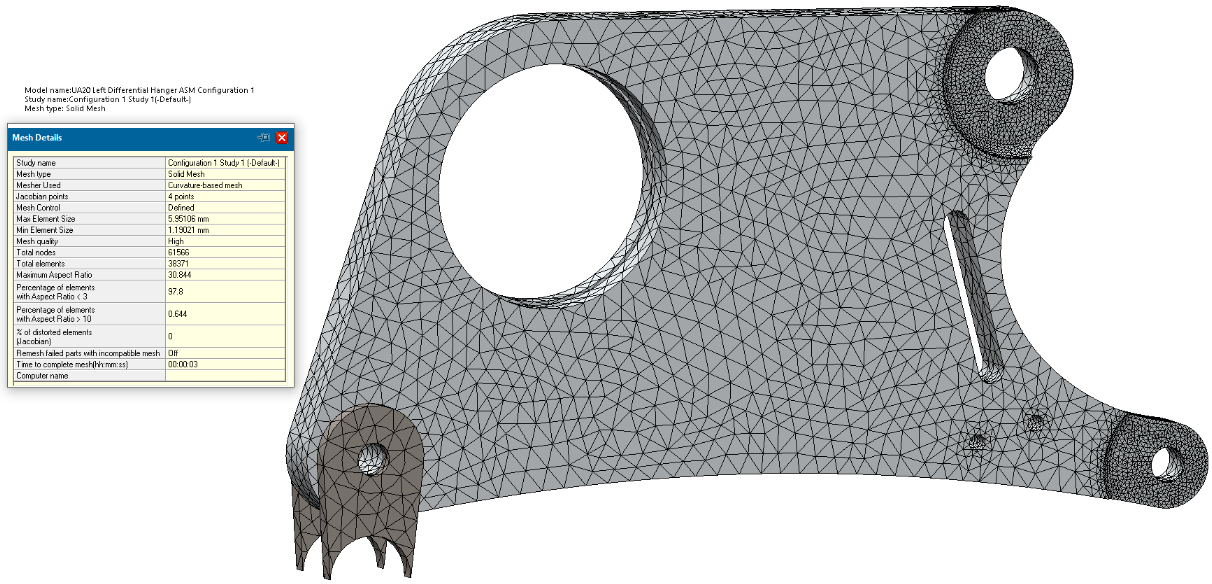


Figure E9: Torque applied to moment axis.

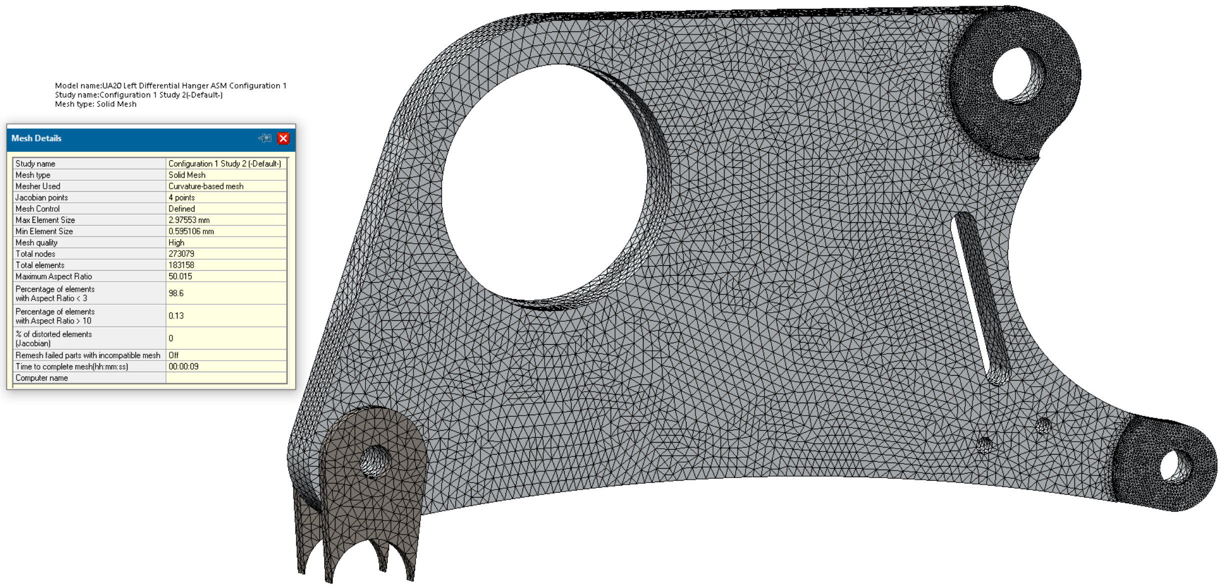
Torque of 847 N·m is applied to the differential hanger to simulate the moment caused by the chain force. The torque value has been calculated in appendix D, section 2 and is scaled to a factor of safety of 1.7 as per the teams UA-20 design guidelines for critical components.

In order to find an appropriate mesh to solve the following iterative studies for weight optimization without results being distorted, multiple studies were conducted in the baseline configuration with differing meshes to verify results convergence in the differential hanger. Two different meshes are shown in figure E10 and figure E11.



Study name	Configuration 1 Study 1 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Curvature-based mesh
Jacobian points	4 points
Mesh Control	Defined
Max Element Size	5.95106 mm
Min Element Size	1.19021 mm
Mesh quality	High
Total nodes	61566
Total elements	38371
Maximum Aspect Ratio	30.844
Percentage of elements with Aspect Ratio < 3	97.8
Percentage of elements with Aspect Ratio > 10	0.644
% of distorted elements (Jacobian)	0
Remesh failed parts with incompatible mesh	Off
Time to complete mesh(hh:mm:ss)	00:00:03
Computer name	

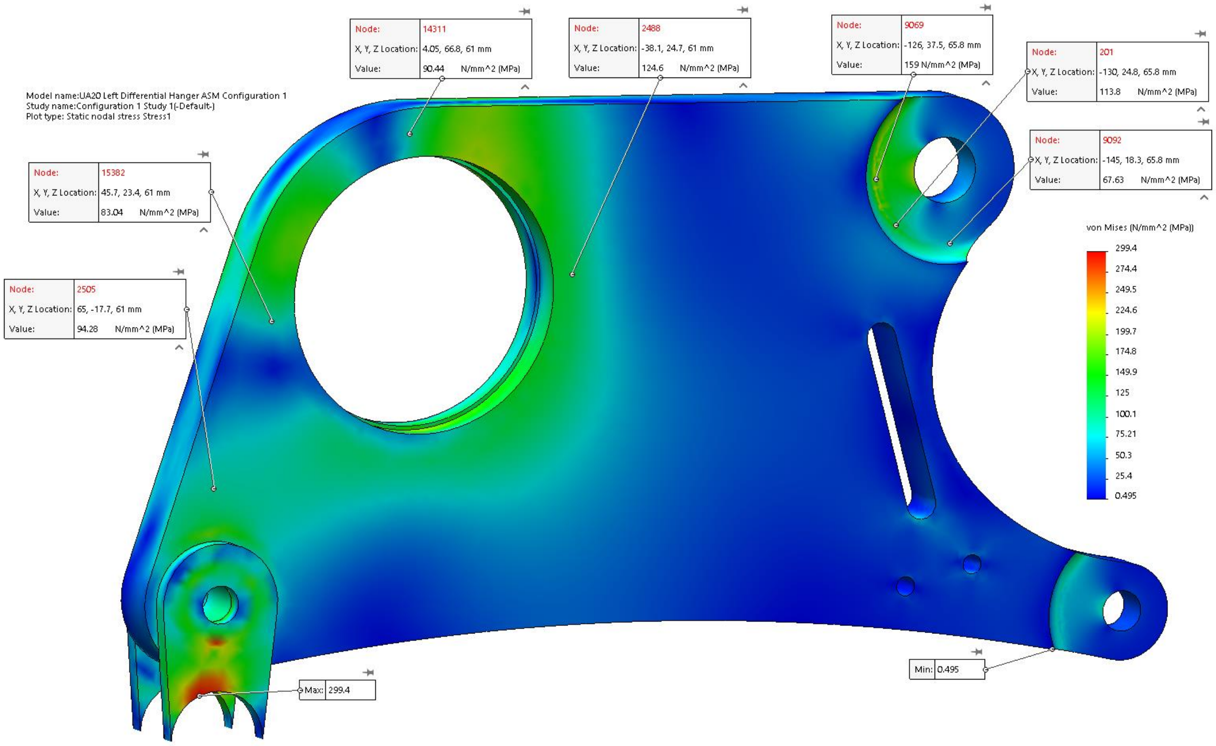
<u>Figure E10:</u> Mesh 1 with maximum element size of 5.95 mm throughout the differential hanger and a mesh control size of 1.2 mm throughout the engine mounting points.



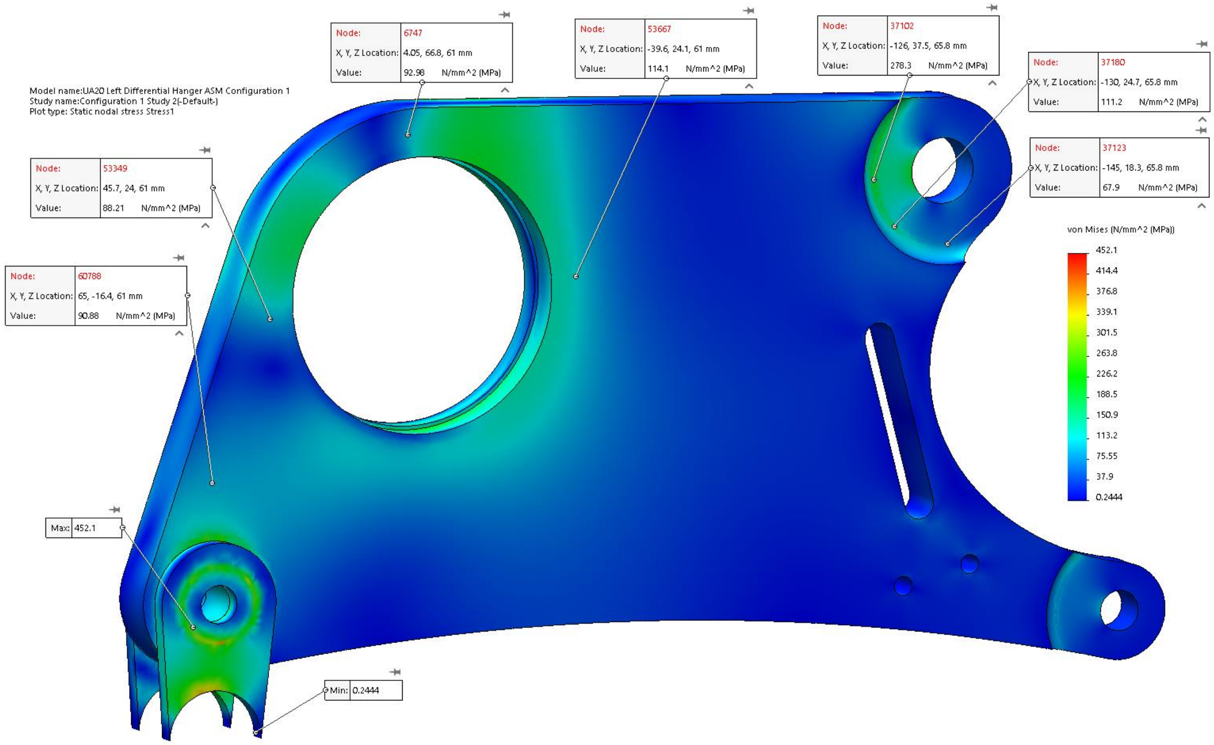
Study name	Configuration 1 Study 2 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Curvature-based mesh
Jacobian points	4 points
Mesh Control	Defined
Max Element Size	2.97553 mm
Min Element Size	0.595106 mm
Mesh quality	High
Total nodes	273079
Total elements	183158
Maximum Aspect Ratio	50.015
Percentage of elements with Aspect Ratio < 3	98.6
Percentage of elements with Aspect Ratio > 10	0.13
% of distorted elements (Jacobian)	0
Remesh failed parts with incompatible mesh	Off
Time to complete mesh(hh:mm:ss)	00:00:09
Computer name	

<u>Figure E11:</u> Mesh 2 with maximum element size of 2.97 mm throughout the differential hanger and a mesh control size of 0.8 mm throughout the engine mounting points.

Curvature-based mesh with 4 Jacobian points is used throughout all static simulations. Mesh 1 is a coarser mesh automatically suggested by SolidWorks with a maximum global element size of 5.95 mm throughout the plate thickness and a mesh control of maximum element size 1.2 mm at the engine mounting points due to the fillets and high model curvature. Mesh 2 halves the maximum global element size of mesh 1 to 2.97 mm and applies a mesh control of 0.8 mm element size at the engine mounting points. Mesh 2 increases the total amount of elements by 377%. The von Mises results for mesh 1 and mesh 2 are shown in figure E12 and E13 respectively.

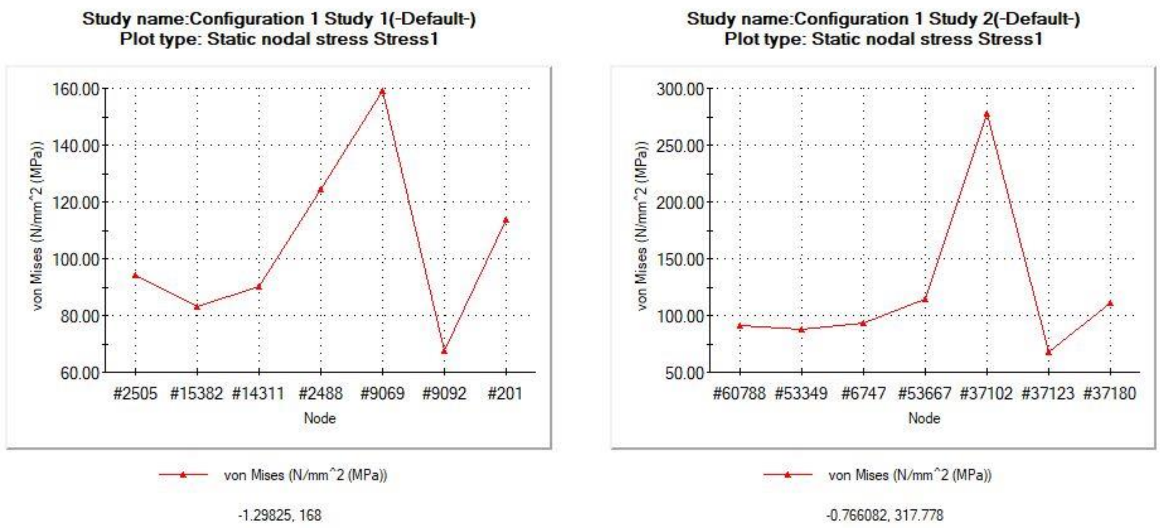


<u>Figure E12:</u> von Mises results for mesh 1 with probed sensor locations shown.



<u>Figure E13:</u> von Mises results for mesh 2 with probed sensor locations shown.

To determine mesh independence, simulation sensors were placed on the differential hanger in critical areas and the probe tool is used in the von Mises results to determine the stress at the nearest node to the sensor as shown in figures E12 and E13. Figure E14 graphs the probed values from the sensors. Table E1 shows the deviations at the probed spots to interpret result differences between mesh 1 and mesh 2.



Mesh 1 **Mesh 2**

Figure E14: Stress vs. sensor locations comparing mesh 1 and mesh 2

Table E1: Stress deviations at probed locations across the left differential hanger.

Sensor	Mesh #1 Stress (MPa)	Mesh #2 Stress (MPa)	Stress Deviation Due to 377% Element Count Increase
1	94.28	90.88	-3.6%
2	83.04	88.21	+6.2%
3	90.44	92.98	+2.8%
4	124.6	141.6	-8.4%
5	159.0	278.3	+75.0%
6	113.8	111.2	-2.2%
7	67.63	67.90	+0.4%

These results suggest that the study is mesh independent due to a marginal increase in stress with nearly 4 times increase in element count. The mesh control around the engine mounting points could possibly be further refined, however sensor #5 with a 75% stress deviation is a single outlier located on the fillet boundary line which could be representing an unrealistic stress concentration. The final design eliminated the need for this fillet and the stress singularity no longer existed. For the proceeding iterative studies, mesh 2 was used.

Furthermore, the increase in the von Mises scale and maximum stress takes place at the differential hanger-rear mounting tab interface on the outside perimeter of the bolt head. Many academic studies suggest that the local stress at bolts and bolt hole locations are unrealistic and a “one-element away” approach should be used for a more reasonable margin of safety calculation.

Section 2 – Iterative Results

7075-T6 Aluminum Yield Strength = 505 MPa (Differential Hanger)
4140 Alloy Steel Yield Strength = 440 MPa (Rear Mounting Tabs)

Iteration 1:

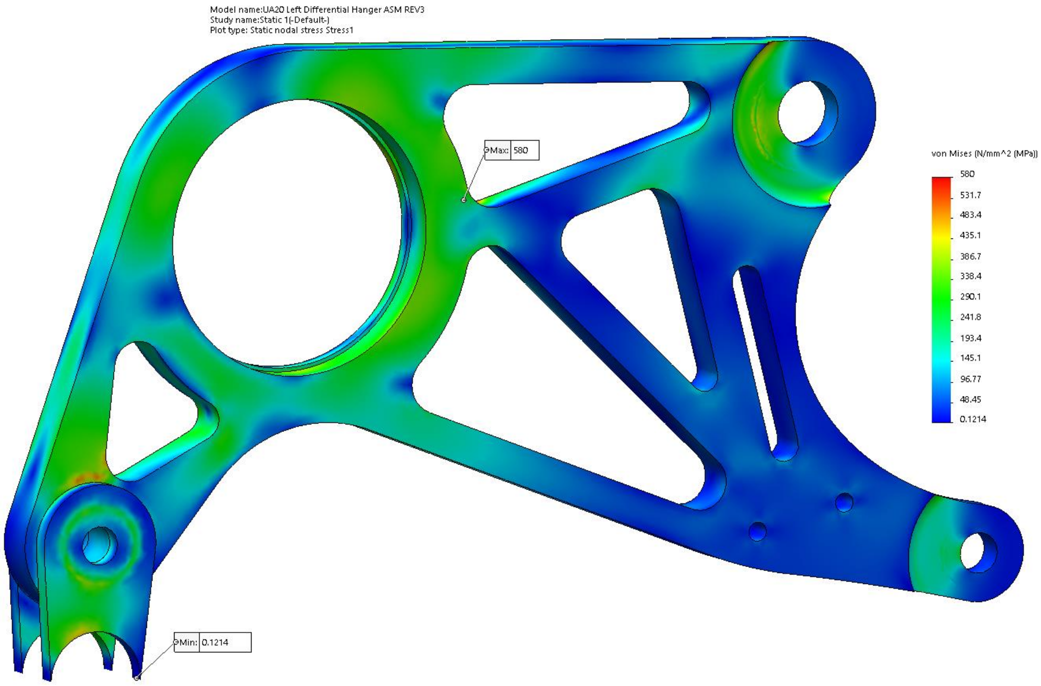


Figure E15: vonMies results for iteration 1 – view #1.

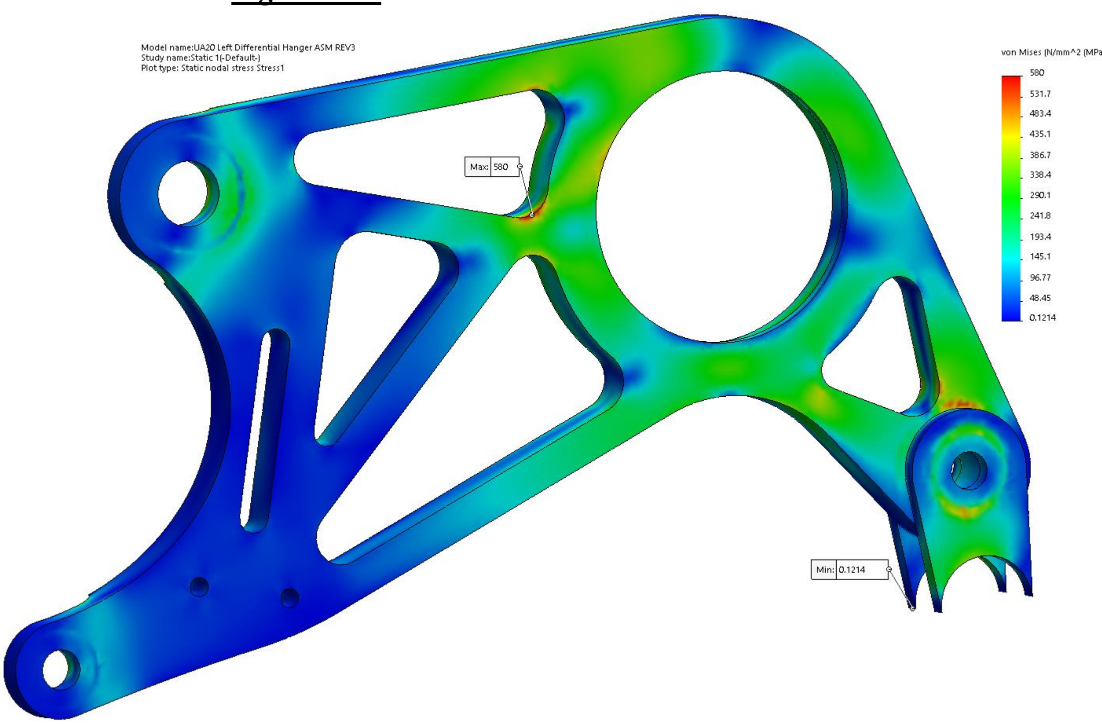


Figure E16: vonMises results for iteration 1 – view #2.

Iteration 2:

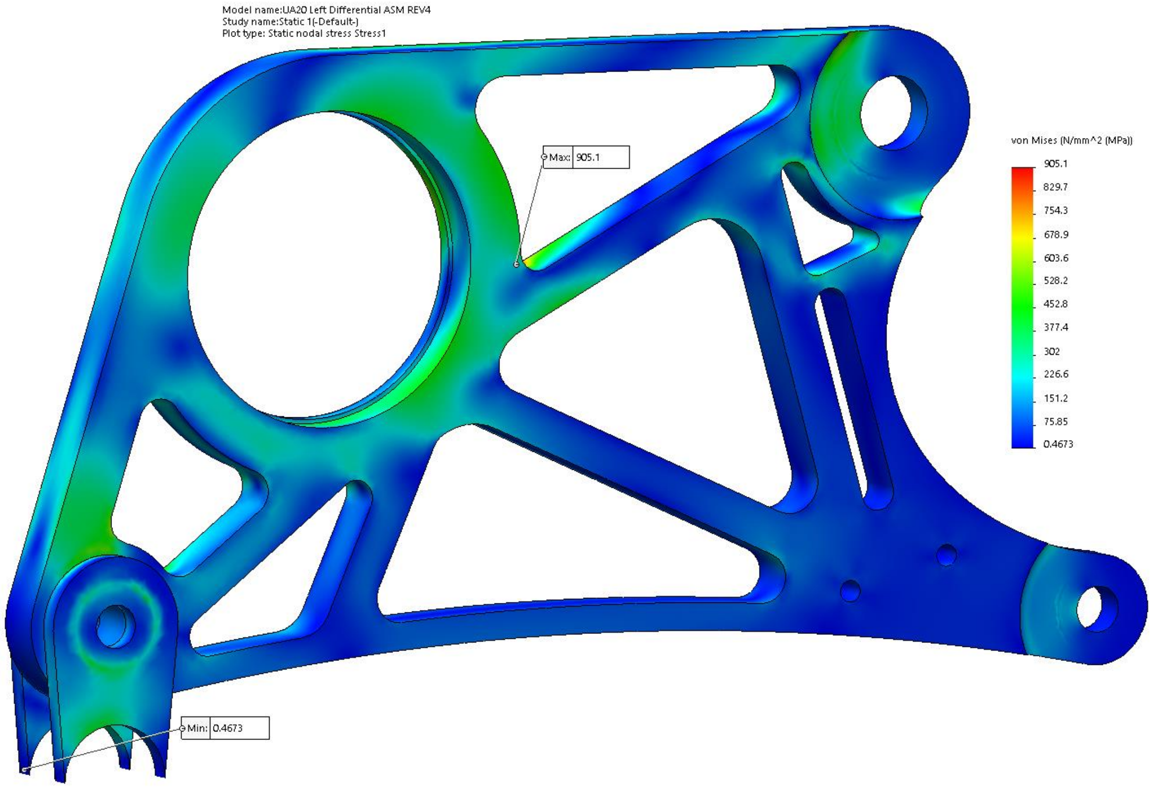


Figure E17: vonMises results for iteration 2 – view #1.

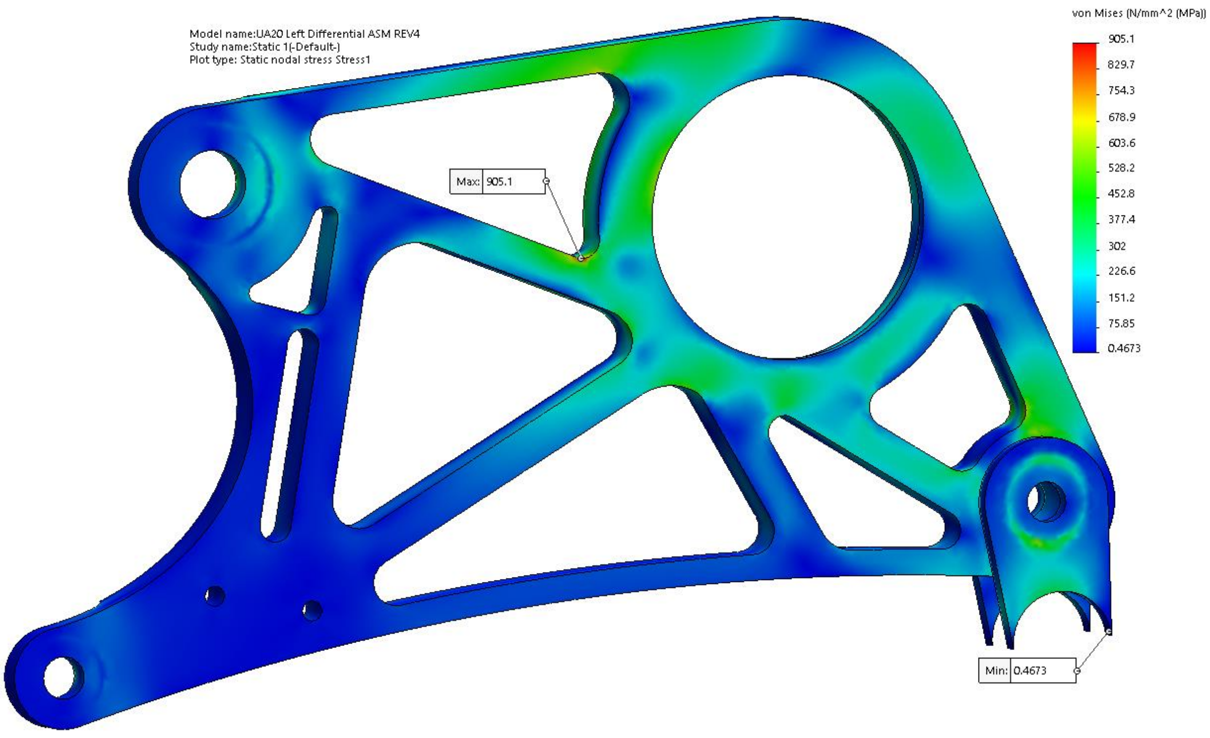


Figure E18: vonMises results for iteration 2 – view #2.

Iteration 3 – Final Design:

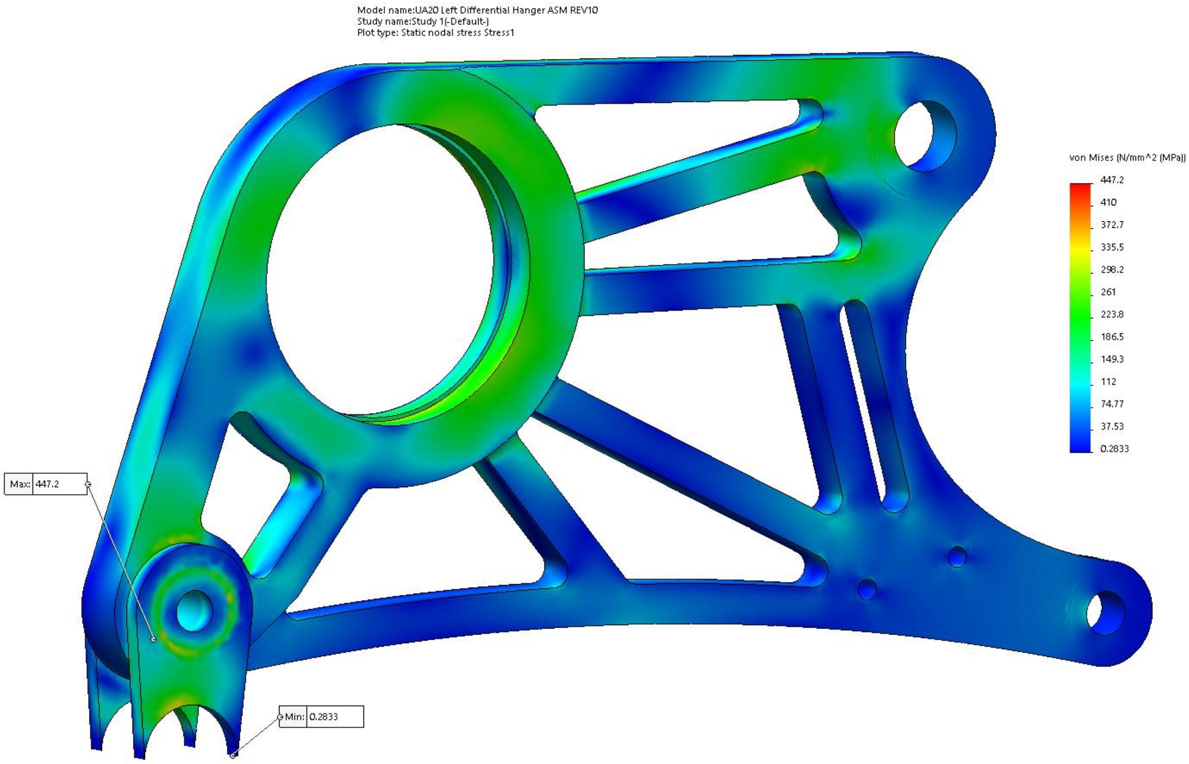


Figure E19: Final design von Mises results – view #1.

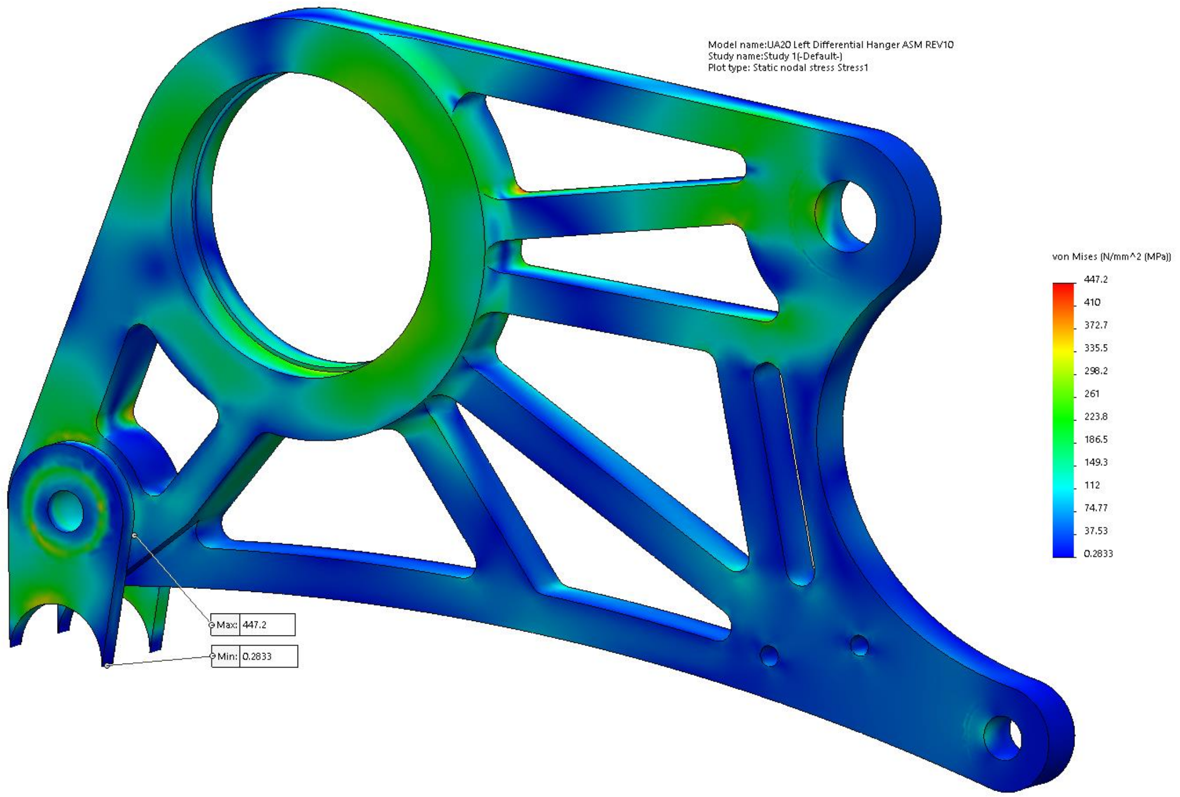


Figure E20: Final design von Mises results – view #2.

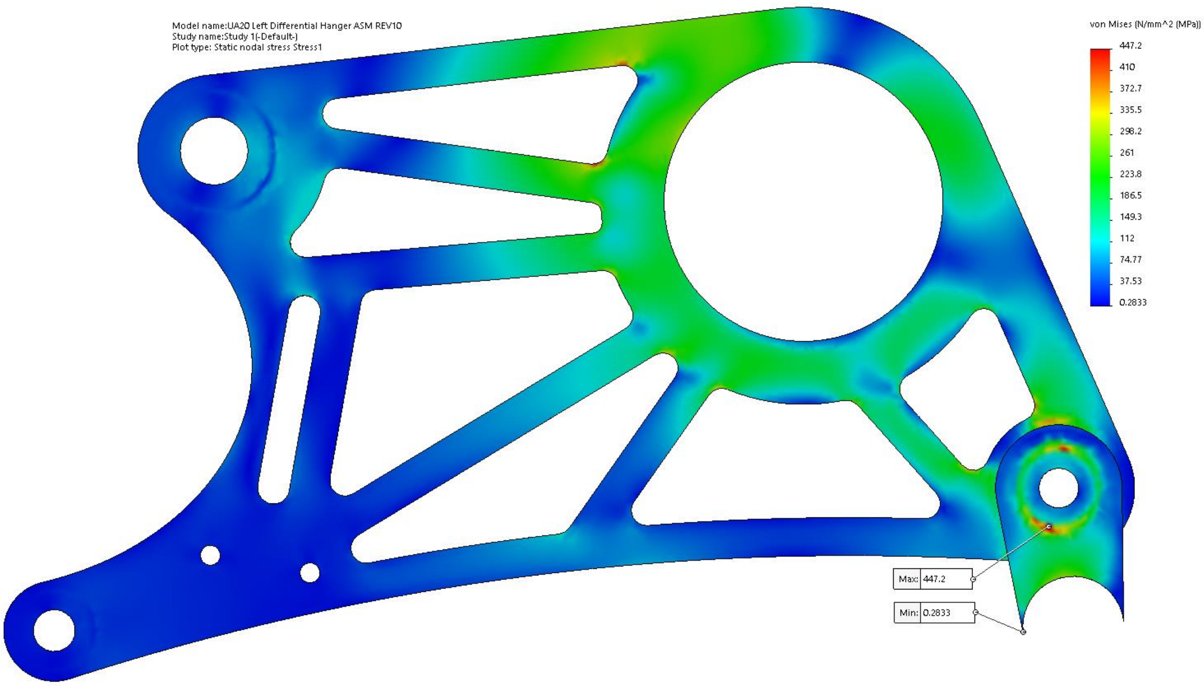


Figure E21: Final design von Mises results – view #3.

Section 3 – Final Design Finite Element Analysis Post-Processing

The mesh used for von Mises results shown in figures E19, E20 and E21 utilized the same mesh parameters as the mesh 2 baseline study shown in figure E11. Figure E22 shows the final designs' mesh details.

Curvature-based mesh was used with 4 Jacobian points with a maximum global element size of 2.97 mm. Mesh control with a maximum element size of 0.8 mm was applied to the engine mounting points and the differential bearing bore fillets. Total number of elements was 118781 and of those elements, 99.3% had an aspect ratio less than 3 which signifies a reduction in the possibility of unrealistic and inaccurate stress concentrations. Figure E23 shows an aspect ratio mesh quality plot. The more blue color found throughout the mesh quality plot, the higher the probability of an accurate mesh.

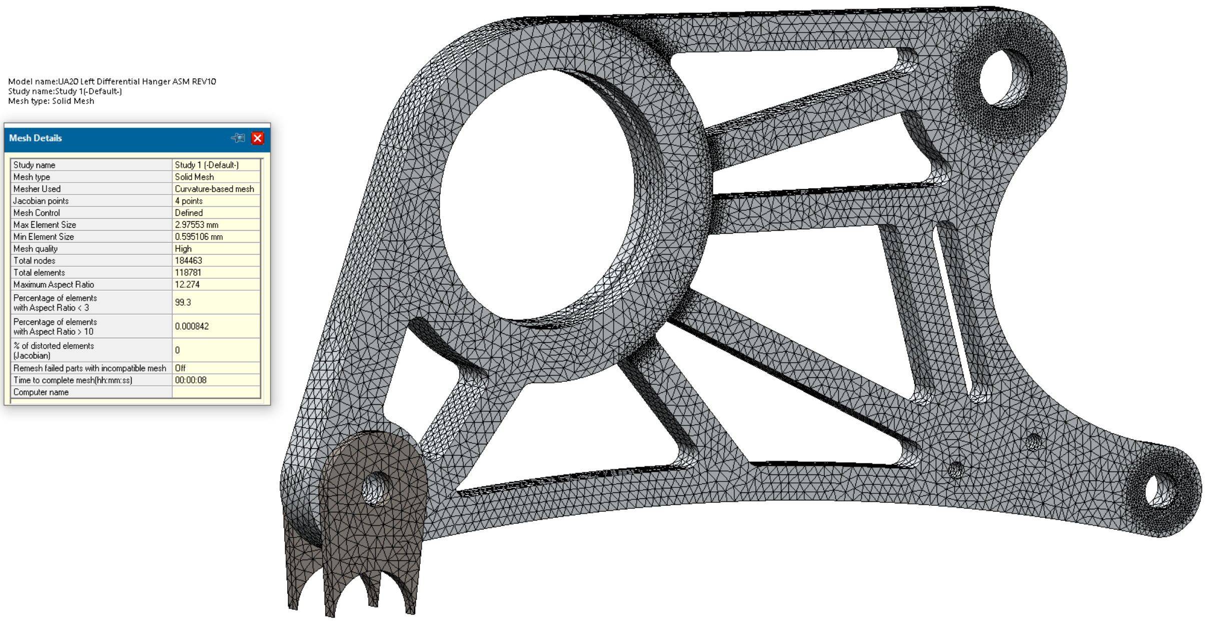


Figure E22: Mesh with maximum global element size of 2.97 mm and mesh control maximum element size of 0.8 mm.

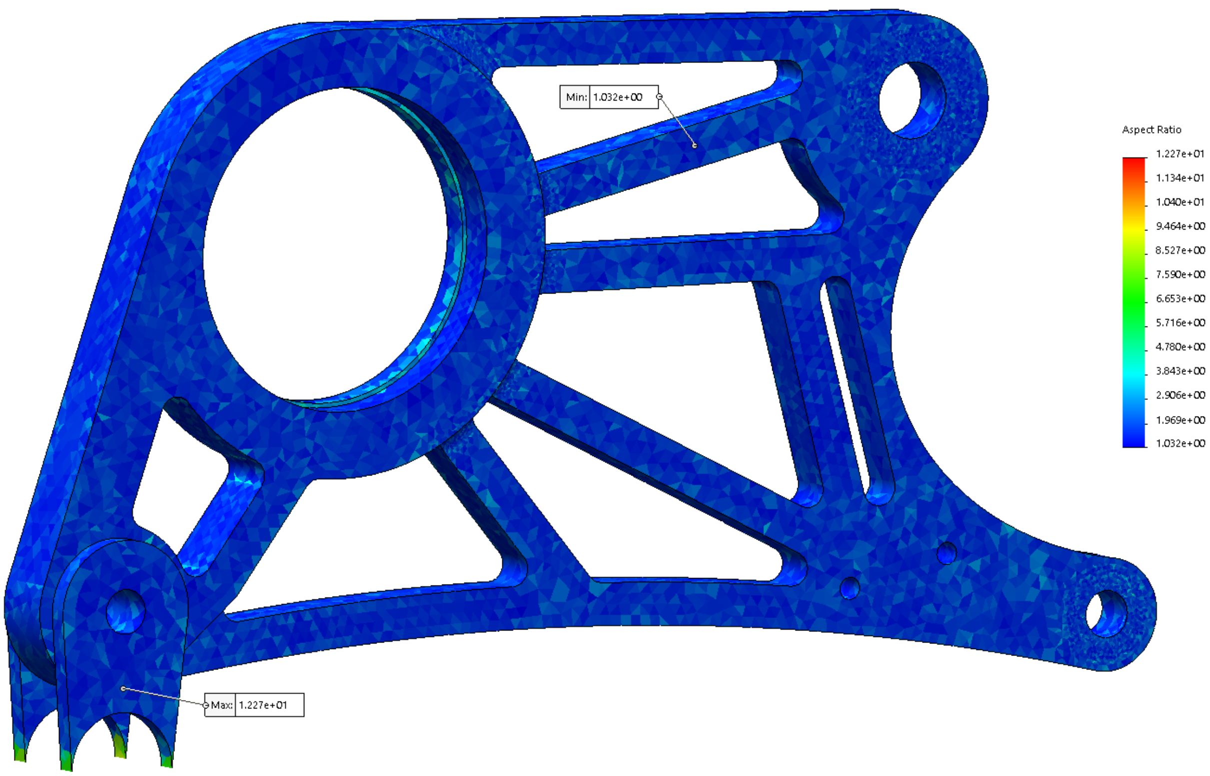


Figure E23: Aspect ratio mesh quality plot.

Displacement results for the final differential hanger design are shown in figure E24. A maximum displacement of 0.886 mm was found which aligns with rigidity predictions calculated in the conceptual phase.

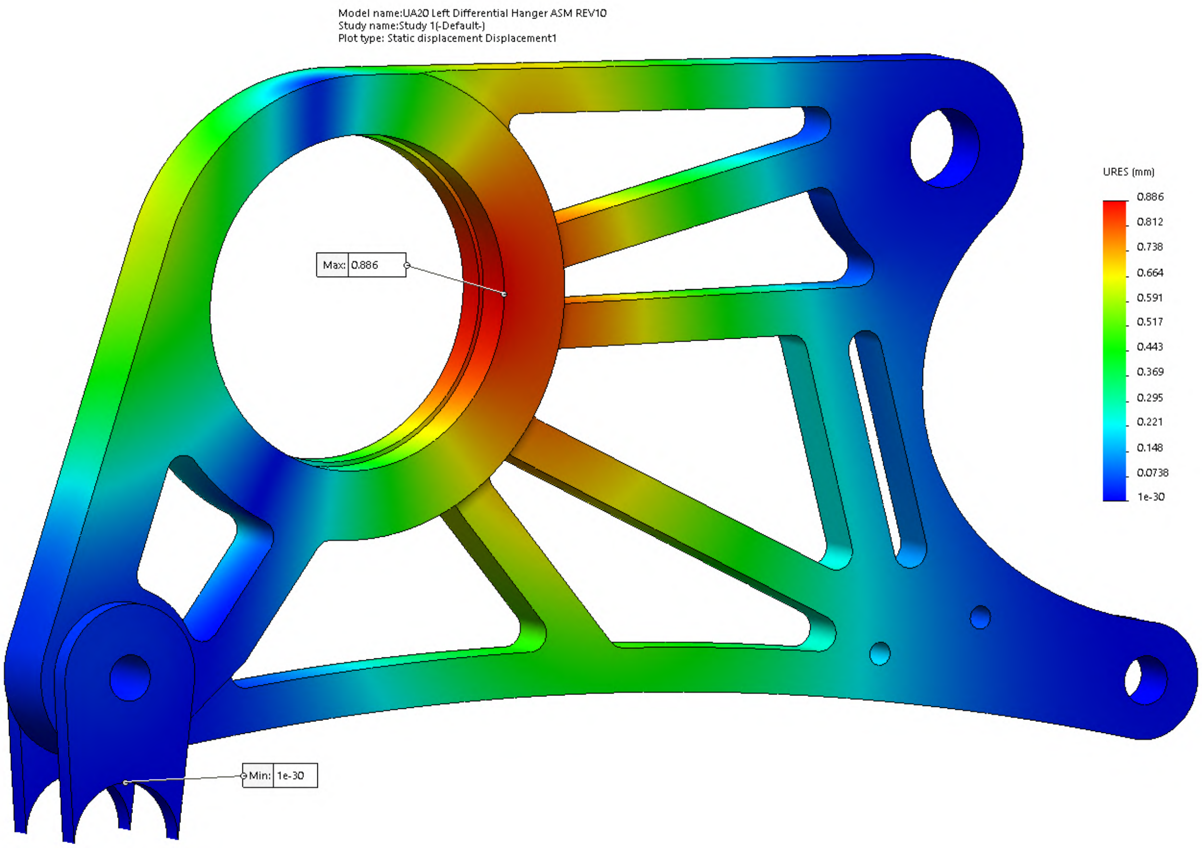
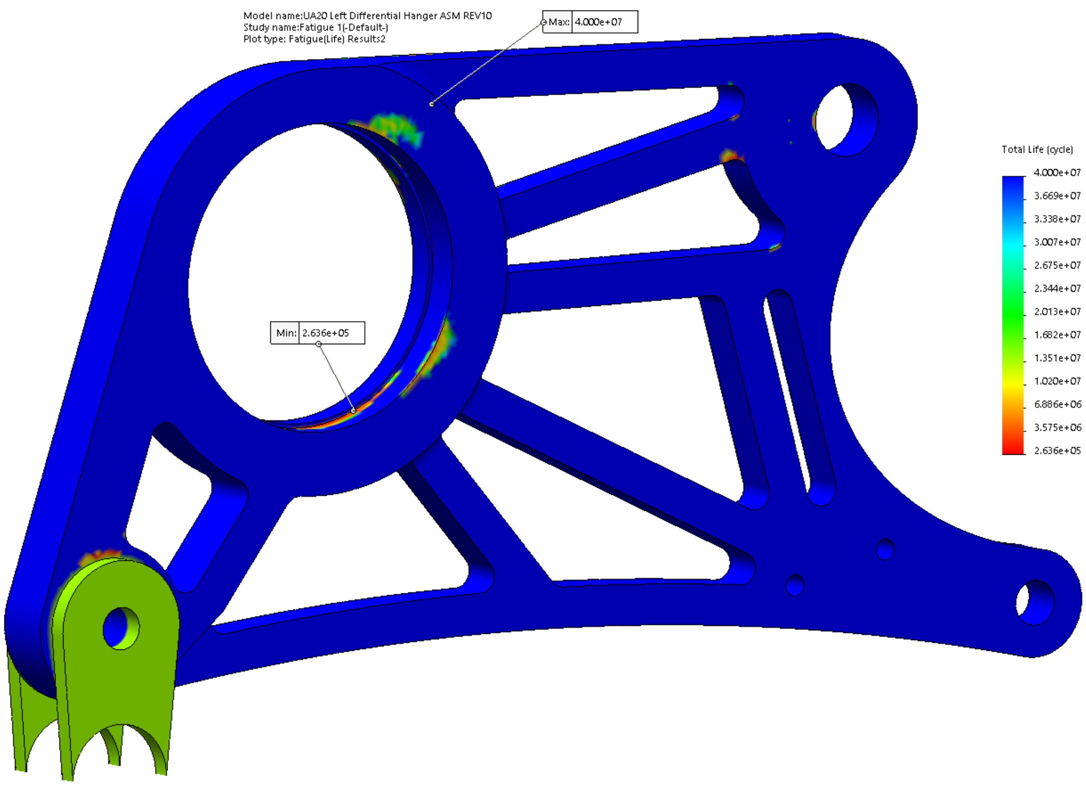


Figure E24: Final design displacement results showing maximum deflection of 0.886 mm.

A fatigue study was conducted based on the final designs' static study loads and constraints. A zero-based loading study was conducted and the minimum number of cycles before damage occurs was calculated to be 263,600 cycles as shown in figure E25. This number of cycles was conservatively calculated to be equivalent to launching the vehicle from rest, up to the maximum chain force (29.8 km/h) and then back to rest consecutively for 3954 km. This is approximately 4 times the predicted lifetime of the vehicle.



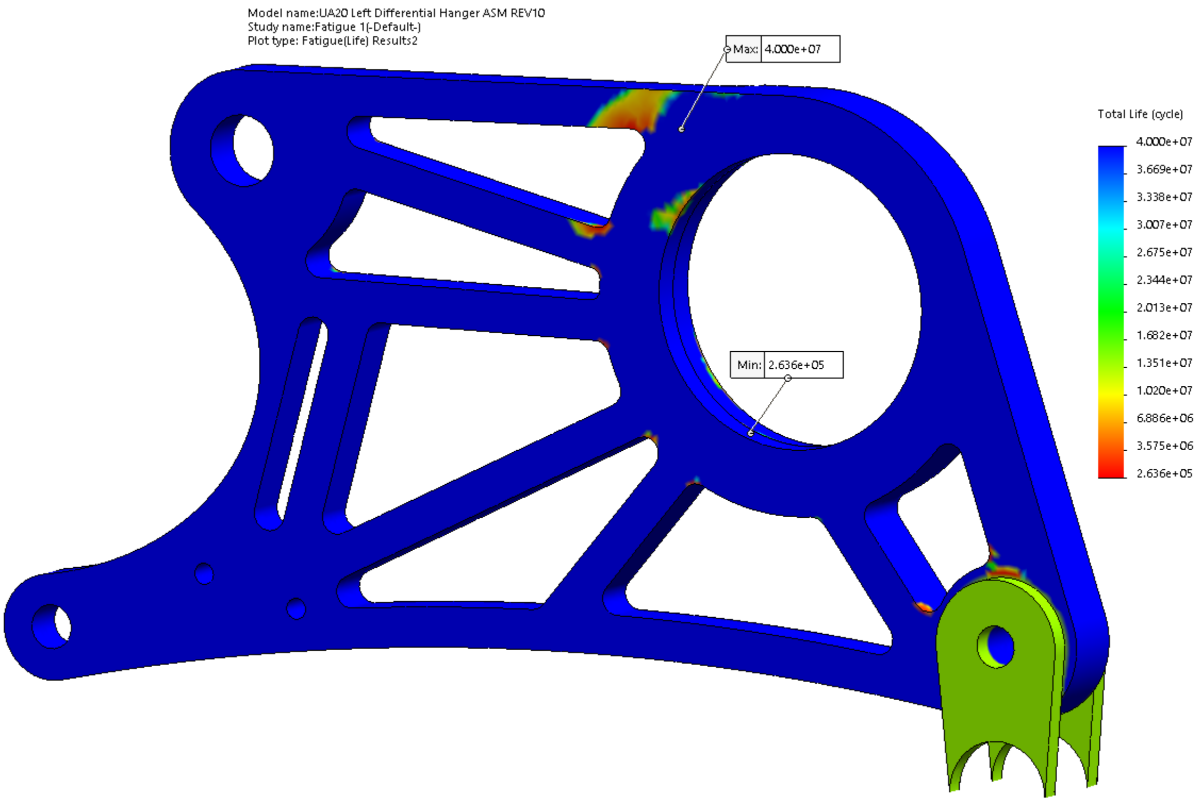


Figure E25: Fatigue study showing minimum number of cycles of 2.636e+05 before damage occurs.

Investigating further the maximum deflection of the differential hangers, a study with the final design geometry discussed above was conducted where the inner differential bearing race was modeled and constrained in the assembly as shown in the figures below.

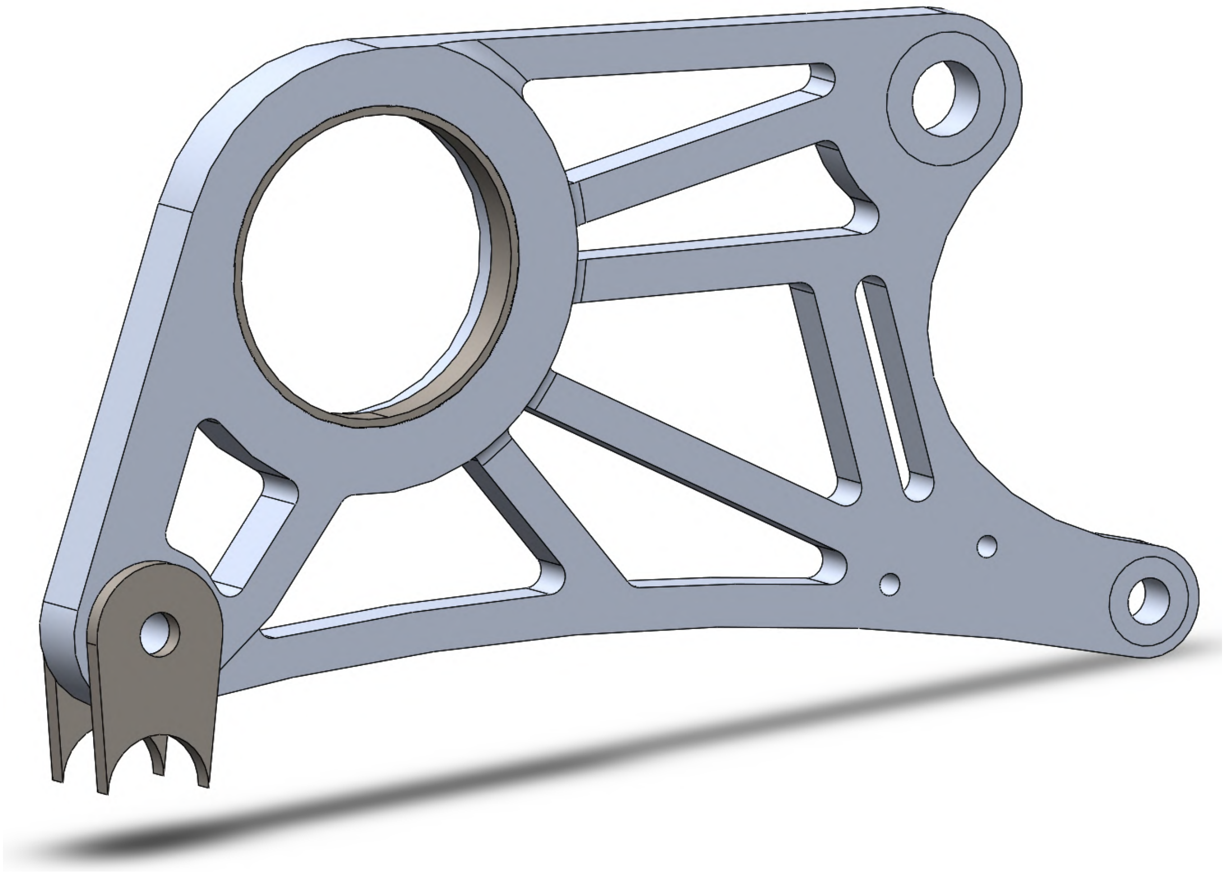


Figure E26: Inner bearing race added to final design static study.

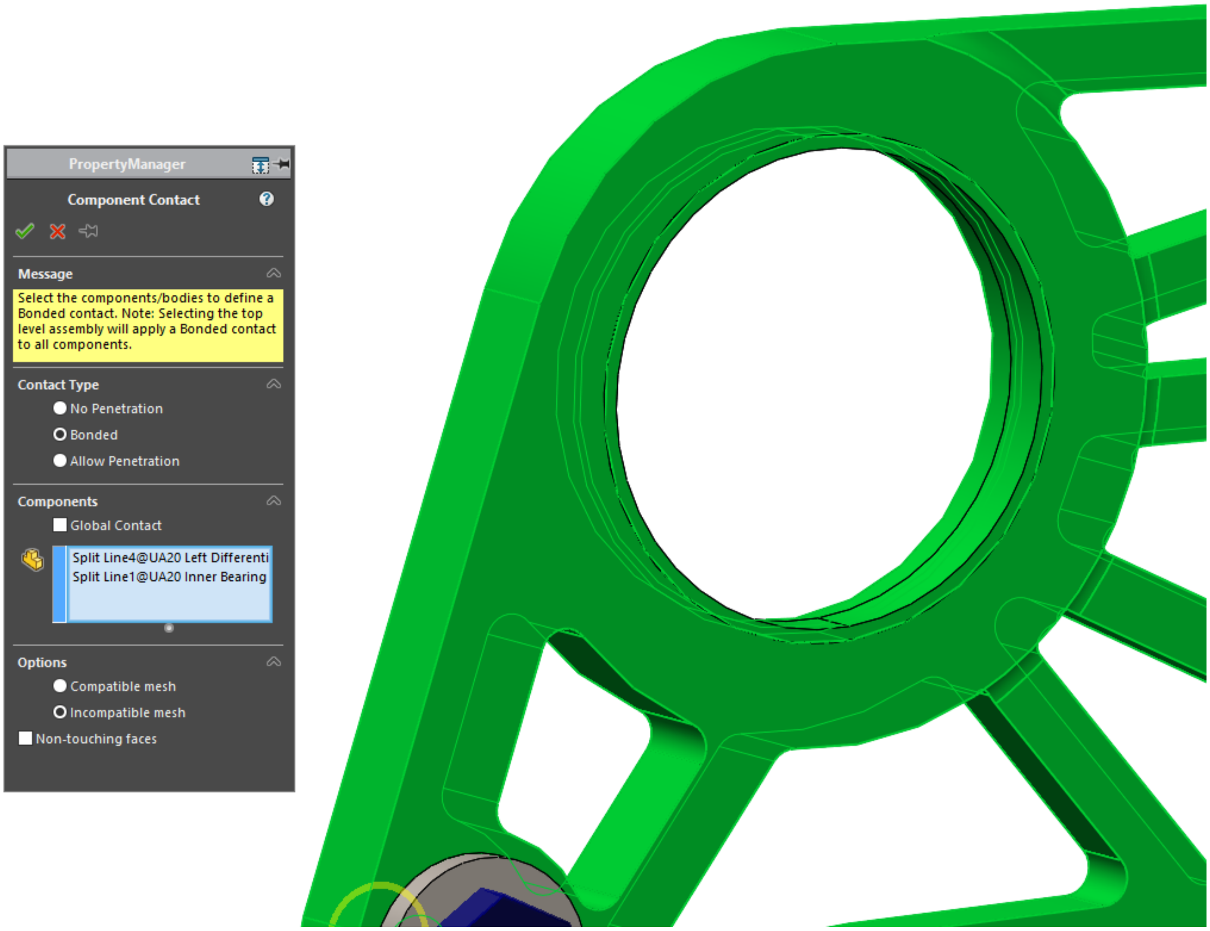


Figure E27: Bonded contact between inner bearing race and differential hanger.

Inner bearing race is set as a rigid component in this study and is constrained by bonding it to the differential hanger. This simulates the assembled state when the differential bearing is press fit into the differential hanger. Maximum displacement was calculated to be 0.598 mm and is shown in figure E28. This confirms speculations that adding a press fit bearing into the differential hanger bearing bore will decrease the maximum deflection as the bearing race acts to maintain the bearing bores cylindrical shape.

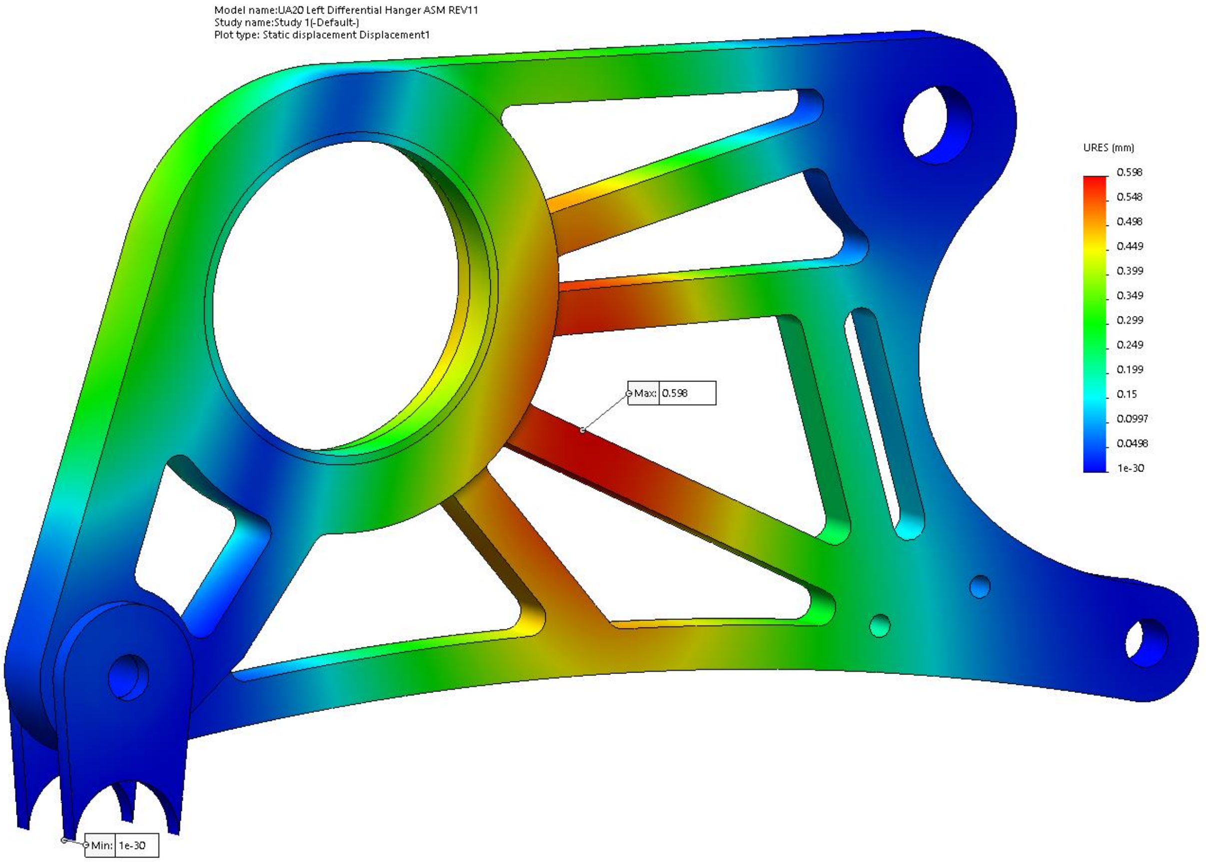


Figure E28: Final design with inner bearing race results showing maximum deflection of 0.598 mm.

Engine Loads

To analyze the loading caused by the engine mounted to the differential hangers, the SolidWorks assembly was imported into Altair Inspire 2019.3 to conduct a finite element analysis study. This was done due to the way SolidWorks simulation handles remote loads. When a rigid remote load is applied to a part in SolidWorks, the area which is attached to the rigid remote load is restricted from deforming. This can produce an inaccurate analysis for the differential hangers as it is expected that the engine mounting holes – the area where the remote load is attached to – will deform and experience some stress. Two studies were conducted, 2 g of longitudinal forces from the engine mass and 2 g of lateral forces from the engine mass. The applied loads are shown below in figure E29 and E30. The differential hanger was constrained in Altair Inspire 2019.3 as close as possible to the SolidWorks constraints outlined in section 1 – finite element analysis pre-processing.

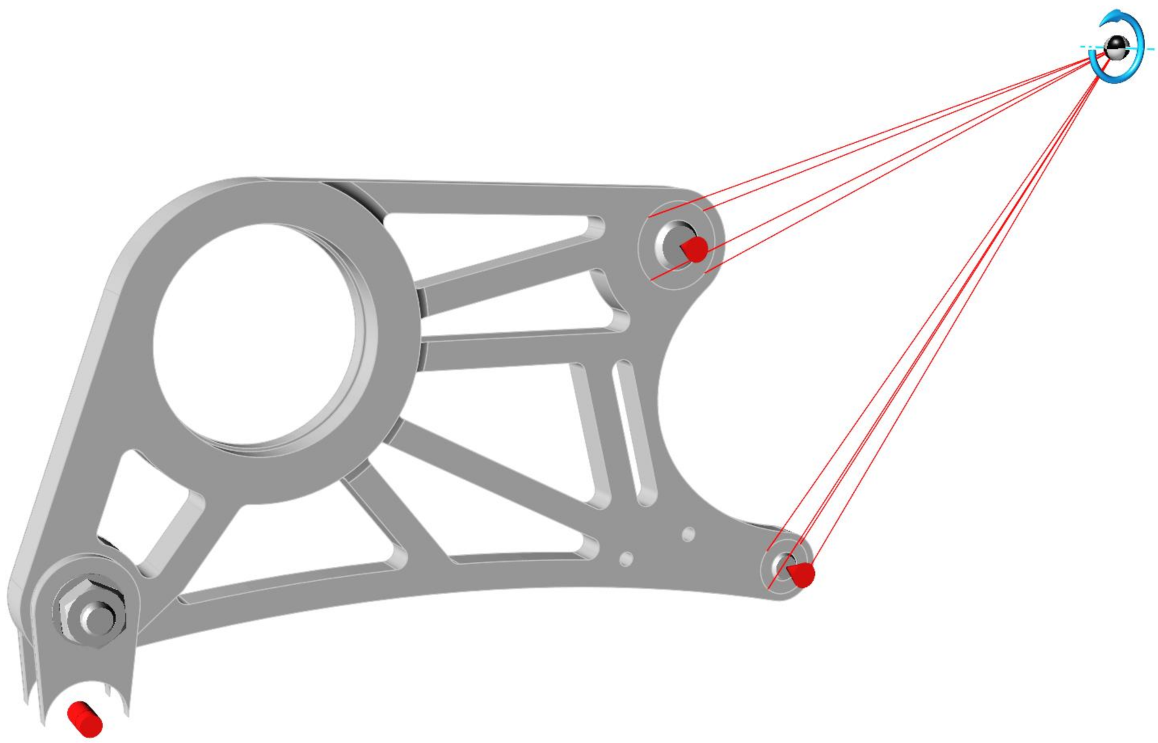


Figure E29: 2 g longitudinal engine mass acceleration applied to the differential hanger in Altair Inspire 2019.3

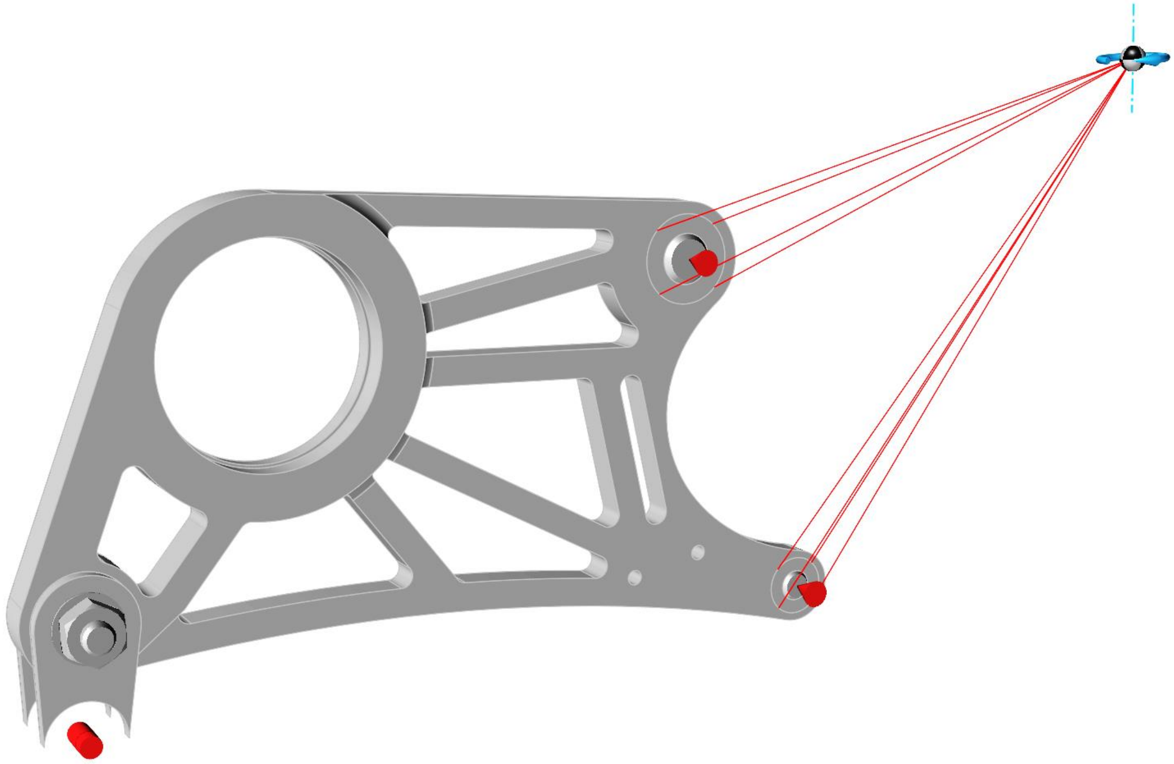


Figure E30: 2 g lateral engine mass acceleration applied to the differential hanger in Altair Inspire 2019.3

A remote mass was placed in space where the center of mass of the KTM 690 would be located when assembled with the differential hangers. The connections between the remote mass and the differential hanger was set to a rigid connection to simulate the engine which can be assumed non-deformable. The wet weight including accessories of the KTM 690 engine is approximately 60 kg, thus the remote mass was constrained to 120kg in both studies to maintain a factor of safety of 2. In both studies, an angular acceleration of 47.16 rad/s^2 was applied to the engine mass to simulate the engines' center of gravity acceleration around the front engine mounts (a radius of 0.208 m from the engines center of gravity) under longitudinal and lateral accelerations.

Altair Inspire 2019.3 uses its own background meshing software called OptiStruct which allows you to set the average element size and apply mesh controls if needed. The global average element size was set to 1.5875 mm and mesh controls with element size of 0.8 mm were applied to the model fillets and the engine mounting holes as done in SolidWorks. The mesh used for both load cases can be seen below in figure E31.

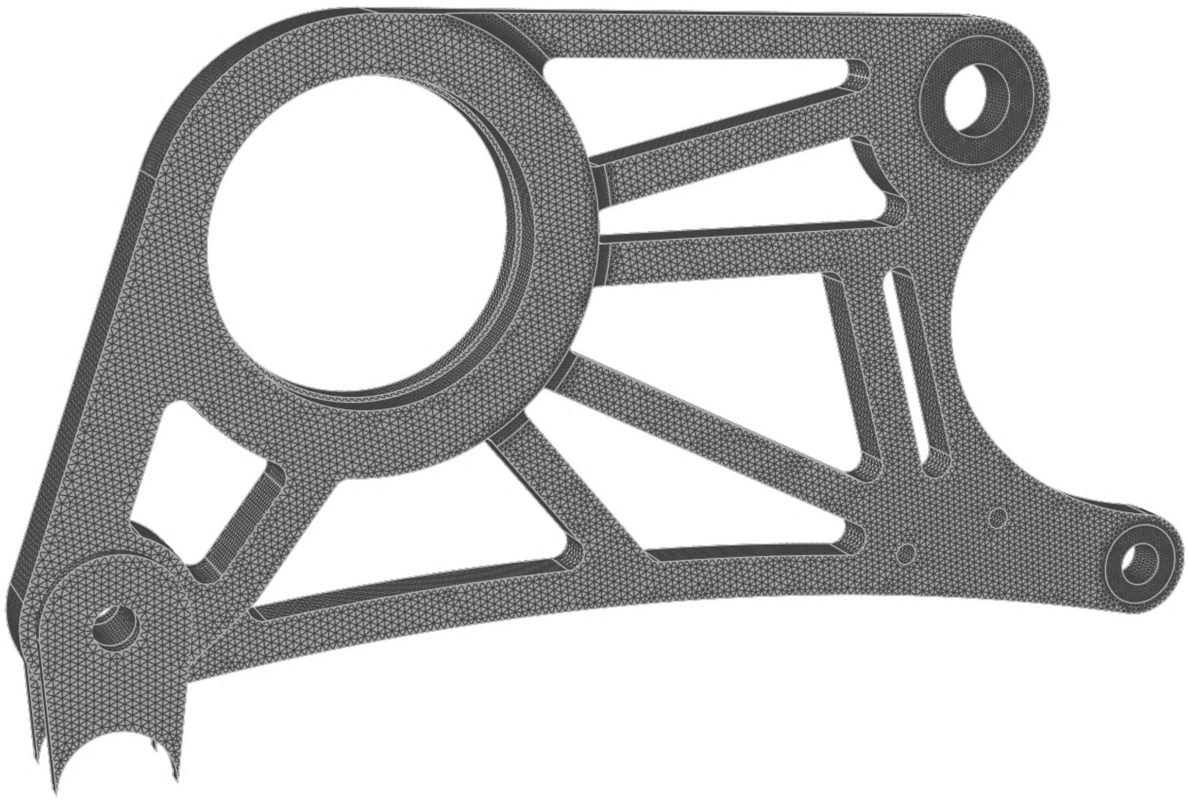


Figure E31: Altair Inspire 2019.3 mesh with average global element size of 1.5875 mm and a mesh control of element size 0.8 mm.

The vonMises results for both engine loading cases are shown below in figures E32 and E33.

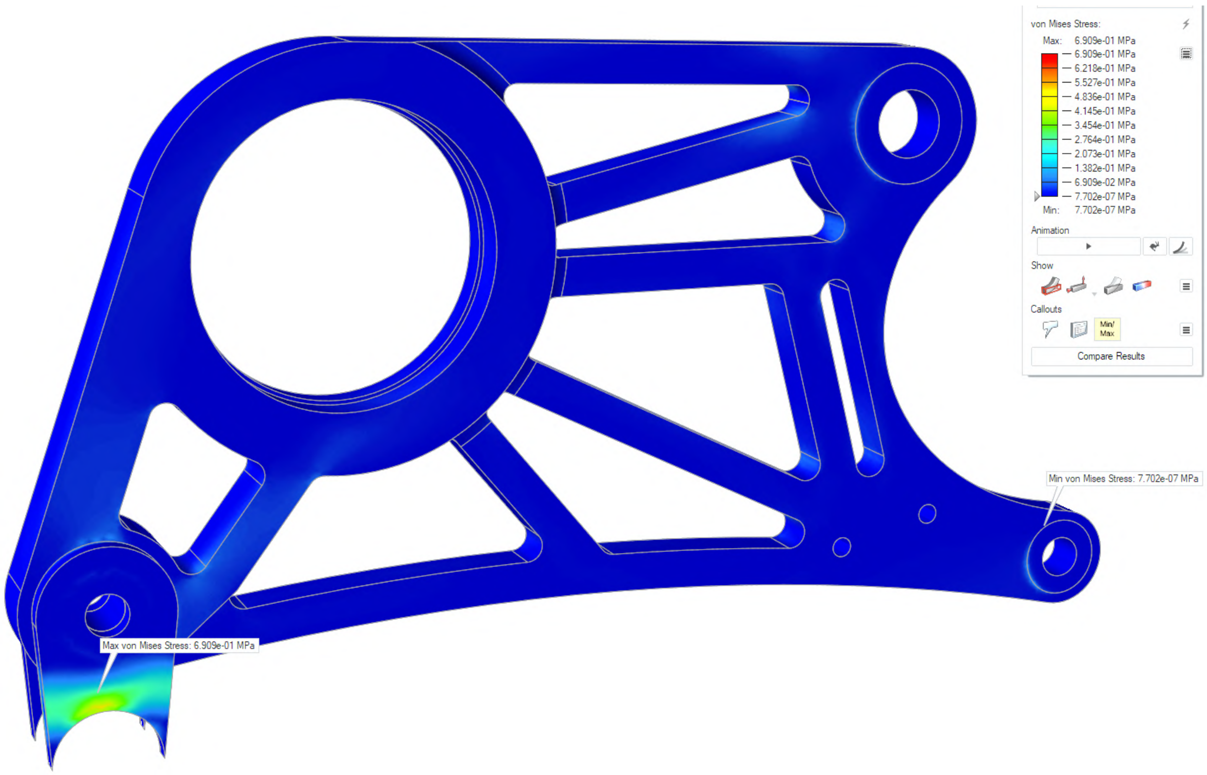


Figure E32: vonMises results for 2 g longitudinal engine mass acceleration.

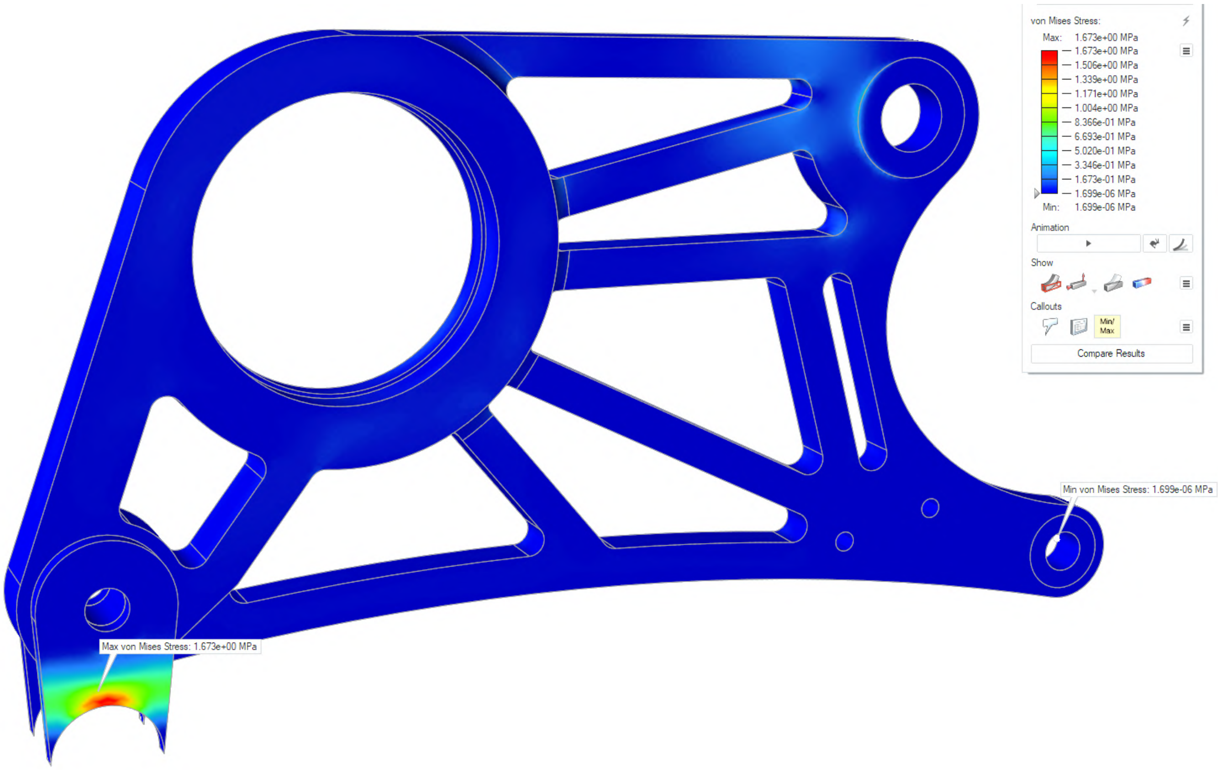


Figure E33: vonMises results for 2 g lateral engine mass acceleration.

The results shown above suggest that the engine loading is negligible in the differential hanger due to the small mass of the engine. The engine loading affects the mounting tabs more than anything, especially during lateral acceleration which is to be expected. To counter the stress shown in the mounting tabs near the weld locations, cross bracing between the two differential hangers should be considered to spread out and absorb more of the lateral engine loading.

Section 4 – Finite Element Analysis Validation

The following hand calculations calculate the maximum stress in a single web of the left differential hanger highlighted in figure E34 using the maximum chain force to validate FEA results.

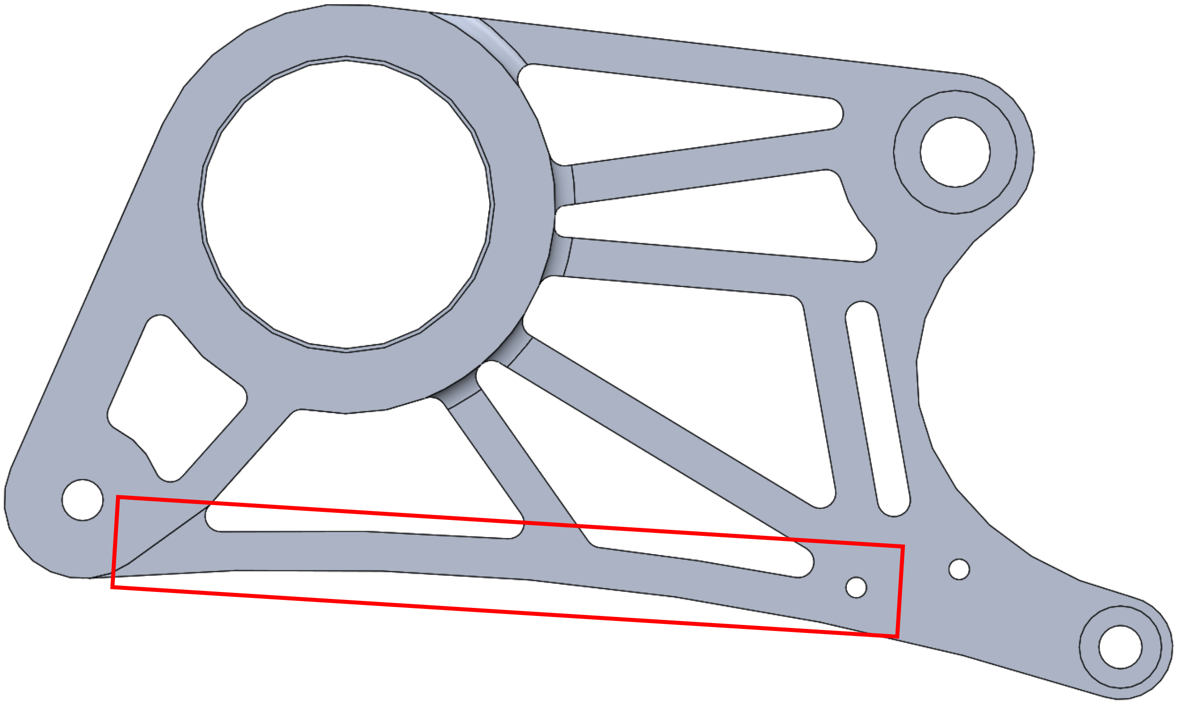
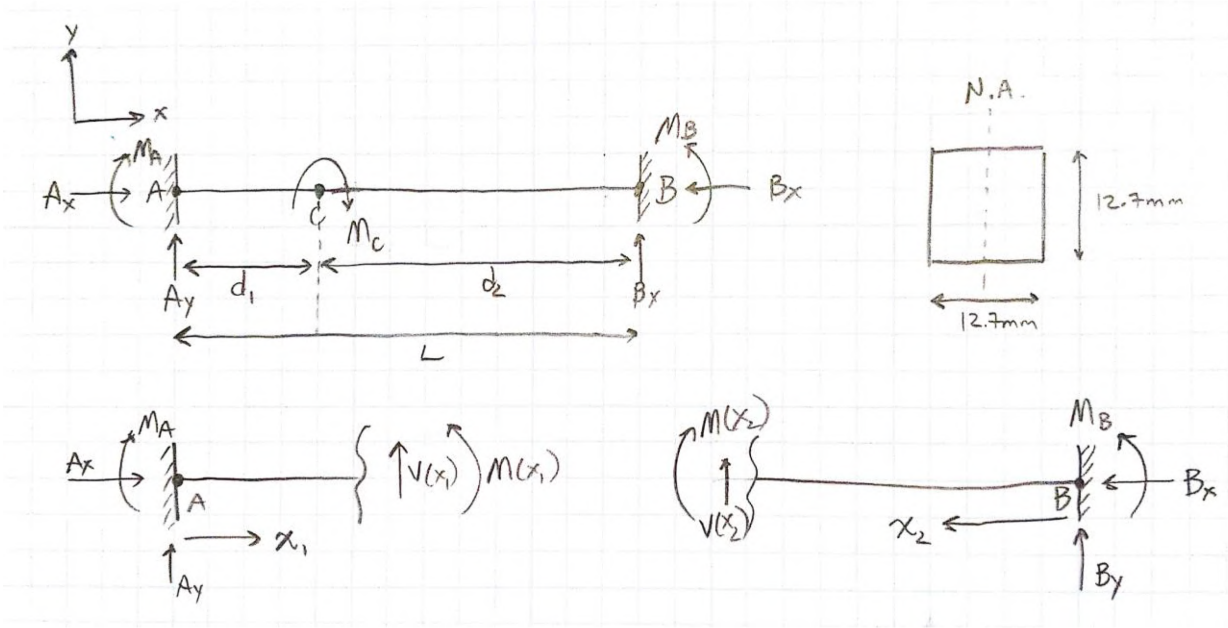


Figure E34: Left differential hanger highlighting single web being calculated for maximum stress.



$M_C = 498\ N \cdot m$ (maximum moment caused by chain force)
$L = 0.252\ m$
$d_1 = 0.064\ m$
$d_2 = 0.188\ m$

$$\sum F_y = 0 = A_y + B_y \rightarrow\ A_y = -B_y\ \ (\mathbf{1})$$
$$\sum M_A = 0 = M_A + M_C - M_B - B_y(L)\ \ (\mathbf{2})$$

$$\sum M(x_1) = 0 = M_A + A_y(x_1)$$
$$M(x_1) = M_A + A_y(x_1);\ \ 0 < x_1 < d_1$$

$$\sum M(x_2) = 0 = M_B + B_y(x_2)$$
$$M(x_2) = M_B + B_y(x_2);\ \ 0 < x_2 < d_2$$

$$EI\frac{d^2 v_1}{d{x_1}^2} = M_A + A_y(x_1) \quad (\mathbf{3})$$
$$EI\frac{dv_1}{dx_1} = M_A(x_1) + \frac{A_y(x_1)^2}{2} + C_1 \quad (\mathbf{4})$$
$$EIv_1 = \frac{M_A(x_1)^2}{2} + \frac{A_y(x_1)^3}{6} + C_1(x_1) + C_2 \quad (\mathbf{5})$$

$$EI\frac{d^2v_2}{d{x_2}^2} = M_B + B_y(x_2) \quad \mathbf{(4)}$$

$$EI\frac{dv_2}{dx_2} = M_B(x_2) + \frac{B_y(x_2)^2}{2} + C_3 \quad \mathbf{(6)}$$

$$EIv_2 = \frac{M_b(x_2)^2}{2} + \frac{B_y(x_2)^3}{6} + C_3(x_2) + C_4 \quad \mathbf{(7)}$$

$$@\ x_1 = 0:\ \frac{dv_1}{dx_1} = 0;\ v_1 = 0$$

$$@\ x_2 = 0:\ \frac{dv_2}{dx_2} = 0;\ v_2 = 0$$

$$\therefore C_1 = C_2 = C_3 = C_4 = 0$$

$$@\ x_1 = d_1, x_2 = d_2 \quad \rightarrow \quad \frac{dv_1}{dx_1} = \frac{dv_2}{dx_2}$$

$$\therefore M_A(d_1) + \frac{A_y(d_1)^2}{2} = M_B(d_2) + \frac{B_y(d_2)^2}{2}$$

$$\rightarrow\ M_A(d_1) + \frac{A_y(d_1)^2}{2} - M_B(d_2) - \frac{B_y(d_2)^2}{2} = 0 \quad \mathbf{(8)}$$

$$@\ x_1 = d_1, x_2 = d_2 \quad \rightarrow \quad v_1 = v_2$$

$$\therefore\ \frac{M_A(d_1)^2}{2} + \frac{A_y(d_1)^3}{6} = \frac{M_b(d_2)^2}{2} + \frac{B_y(d_2)^3}{6}$$

$$\rightarrow\ \frac{M_A(d_1)^2}{2} + \frac{A_y(d_1)^3}{6} - \frac{M_b(d_2)^2}{2} - \frac{B_y(d_2)^3}{6} = 0 \quad \mathbf{(9)}$$

Solving equations **(1), (2), (8) and (9):**

$$A_y = -5204.34\ N$$

$$B_y = 5204.34\ N$$

$$M_A = 405.70\ N \cdot m$$

$$M_B = -407.79\ N \cdot m$$

The respective shear force and bending moment diagram is shown below in figure E35.

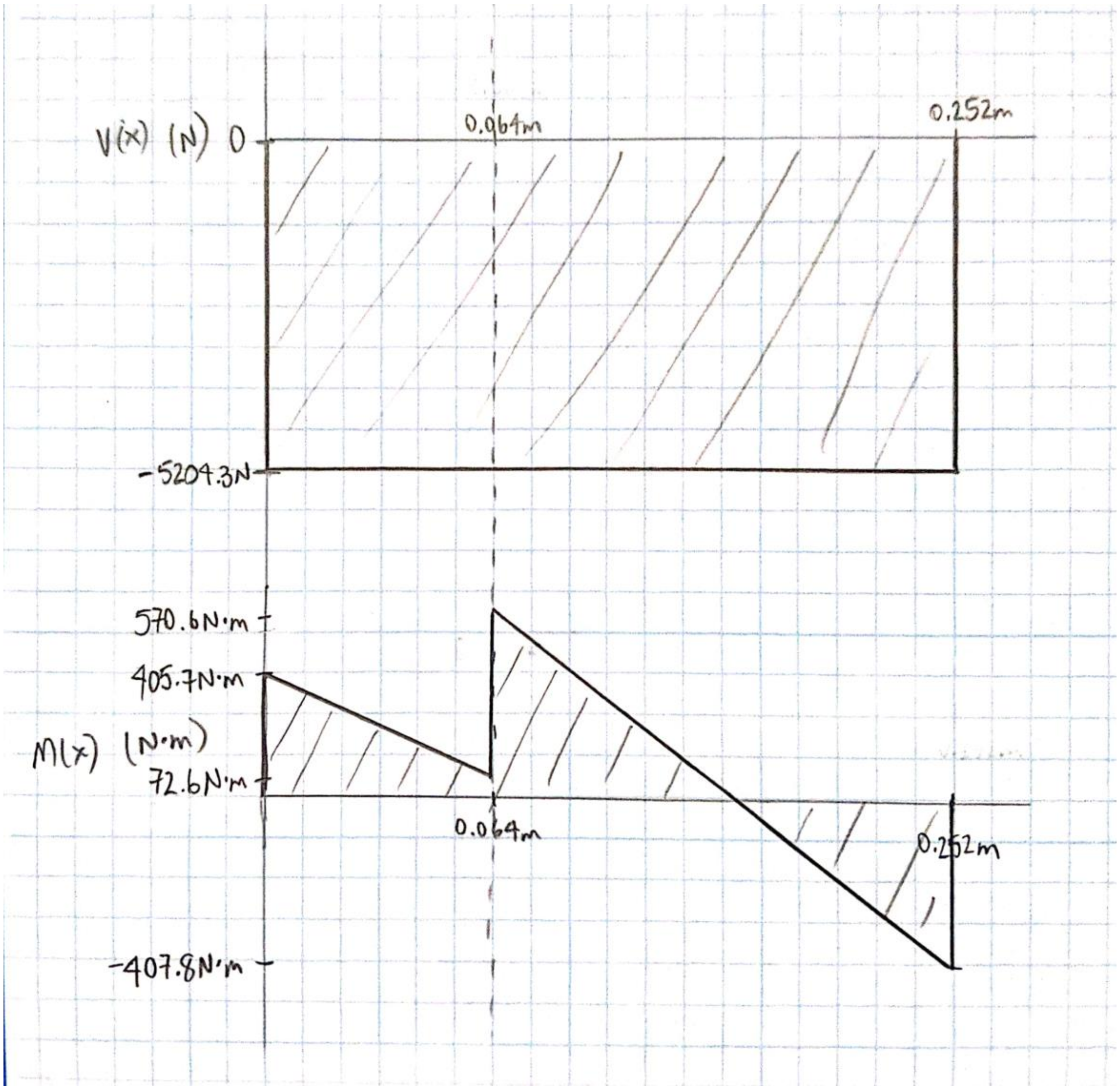


Figure E35: Shear force and bending moment diagram for calculated differential hanger web.

$$I = \frac{1}{12}(12.7\,mm)^4 = 2167.87\,mm^4$$

$$\sigma_{max=} \frac{M_{max} \cdot c}{I} = \frac{\left(570600\,\frac{N}{mm}\right) \cdot (6.35\,mm)}{2167.87\,mm^4} = 1671.37\,MPa$$

Based on the calculations shown above, the max stress if only a single 0.5"x0.5" web existed on the differential hanger is 1671.3 MPa – 3 times the yield strength of 7075-T6 aluminum. However, as shown in figure E34, the differential hanger has 5 webs of the same cross-sectional area acting in the same direction which all act to absorb some of the stress calculated above.

Appendix F
Differential Bearing Lifespan Calculations

Requirements for the differential bearings is that they are sealed on both sides due to being exposed to harsh environments, the inner bore must fit the differential and that they must be small and compact to reduce additional rotating mass.

Because the car never uses 5th or 6th gear from the KTM 690 transmission, an average chain force across all RPM ranges through the usable gears is calculated from appendix D, section 1, table D2 to be 6894.5 N, or 1549.9 lbf. A free body diagram is shown below in figure F1 and the bearing reaction force calculations follow.

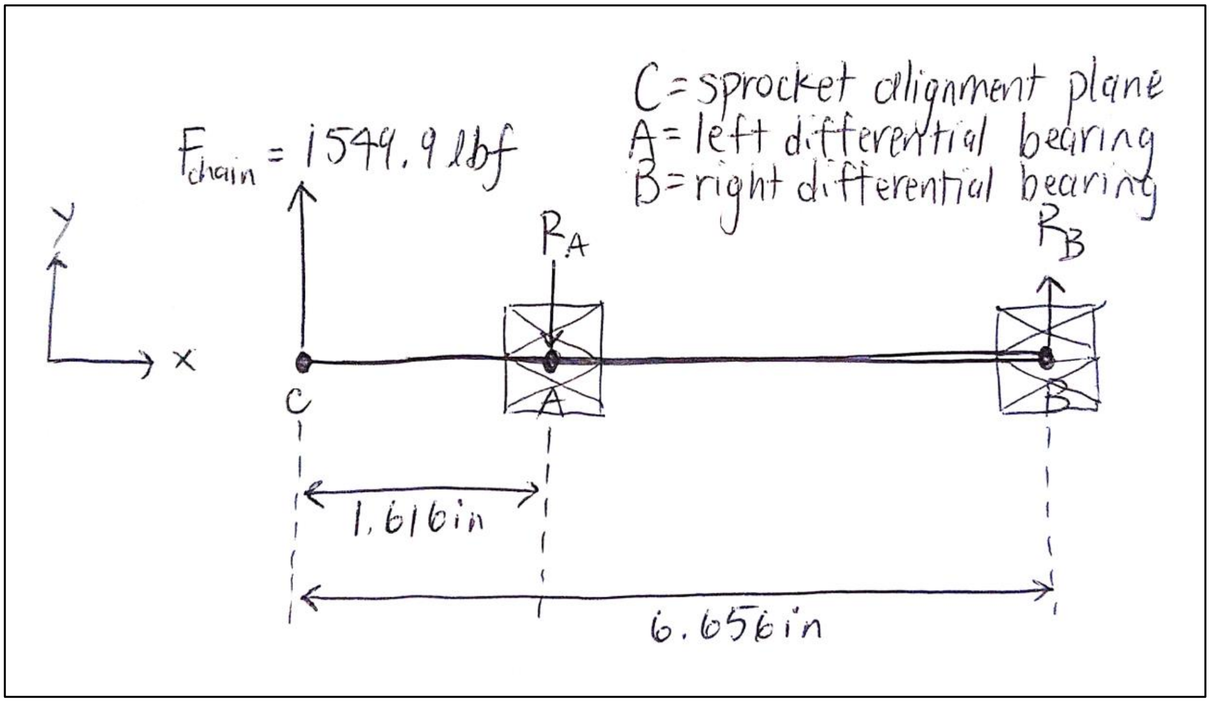


Figure F1: Differential bearing reaction forces.

$$\sum F_y = 0 = 1549.9\ lbf - R_A - R_B \qquad (1)$$

$$\sum M_B = 0 = (1549.9\ lbf) \cdot (6.656\ in) - R_A \cdot (5.040\ in) \qquad (2)$$

Solving (1) and (2):

$$\boxed{\begin{array}{c} R_A = 2047.02\ lbf \\ R_B = 497.08\ lbf \end{array}}$$

Since the differential approximately rotates at the same rate as the rear wheels, the total distance travelled by the UA-20 vehicle per revolution of the differential can be calculated. This calculation is shown below.

$$Tire\ Radius = 0.26045\ m$$

$$Distance\ travelled\ per\ differential\ rotation = 2 \cdot \pi \cdot (0.26035\ m) = 1.635\ m$$

Historical data from the University of Alberta Formula Racing Team shows that a car drives at most 1000 km per season, 250 km total at competitions and the remainder during testing sessions. Applying a factor of safety of 2, we can confidently assume the UA-20 vehicle will not exceed 2000 km of total driving throughout the season.

$$2000\ km = 2 \cdot 10^6\ m$$
$$L_P = Total\ bearing\ cycles = \frac{2 \cdot 10^6\ m}{1.635\ m} = 1.223 \cdot 10^6$$

Typical ball bearing life is calculated with a 90% reliability factor. Calculations for the required dynamic load of the left and right deep groove ball bearings is shown below.

$$K_R = reliability\ factor = 1$$
$$C_{R,A} = R_A \cdot \left(\frac{L_P}{K_R}\right)^{\frac{1}{3}} = (2047.02\ lbf) \cdot \left(\frac{1.223}{1}\right)^{\frac{1}{3}} = 2189.1\ lbf$$
$$C_{R,B} = R_B \cdot \left(\frac{L_P}{K_R}\right)^{\frac{1}{3}} = (497.08\ lbf) \cdot \left(\frac{1.223}{1}\right)^{\frac{1}{3}} = 531.6\ lbf$$

Based on the dynamic loads required, left and right differential bearings were chosen based on local availability, and the requirements stated earlier. The selected bearings are listed in Table F1.

Table F1: Selected Differential Bearings

Ball Bearing Model	Inner Bore Diameter (mm)	Dynamic Load Rating (lbf)
61811-2RSR-Y	55	2158.2
61810-2RSR	50	1618.6

Total number of driven kilometers before bearing fatigue begins can be calculated from the information above and the calculations are shown below.

$$L_{P,A} = K_R \cdot \left(\frac{C_{R,A}}{R_A}\right)^3 = (1) \cdot \left(\frac{2158.2\ lbf}{2047.02\ lbf}\right)^3 = 1.172 \cdot 10^6\ cycles = 1916\ km$$
$$L_{P,B} = K_R \cdot \left(\frac{C_{R,B}}{B}\right)^3 = (1) \cdot \left(\frac{1619.6\ lbf}{497.08\ lbf}\right)^3 = 34.599 \cdot 10^6\ cycles = 56{,}569\ km$$

After driving the UA-20 vehicle 1000 km, the left differential bearing should be inspected for unusual wear and increased friction, and if needed it will be replaced.

Appendix G
Cost and Manufacturing Analysis

To analyze the cost of manufactured components, the FSAE competition utilizes their own cost system that requires the input of material and manufacturing operations in order to set a standardized cost for a part which can be compared across vehicles. Only parts that differ from the UA-19 and UA-20 car are analyzed in this section. Machining costs utilize a quantity defined by the volume of material removed and the corresponding machining operation. The figures below show the manufacturing operations for each UA-20 part that differs from the UA-19 vehicle. The figures also show FSAE competitions' standardized cost for each part which will be displayed in the teams cost report.

	Part	Part #	Op Num	Part Cost	Quantity	Subtotal
1	Left Differential Hanger	-	1	36.43	1	36.43
2	Right Differential Hanger	-	2	35.16	1	35.16
3	Idler Sprocket Shaft	-	3	6.32	1	6.32
4	Idler Sprocket	-	4	4.43	1	4.43
						Subtotal $82.34

Add Part

Figure G1: Total cost of UA-20 drivetrain parts that differ from the UA-19 vehicle.

	Material	Use	Op Num	Size 1	Size 2	Area Name	Area	Length	Density	Quantity	Unit Cost	Subtotal
1	Aluminum, Premium (by Dimensions)	stock hanger material	1			Plate	101.526 in^2	0.625 in	0.098 lb/in^3	1	11.8467	11.85

Subtotal $11.85

Add Material

Processes

	Process	Use	Op Num	Quantity	Multiplier	Mult. Val.	Unit Cost	Subtotal
1	Machining Setup, Install and remove	prepare waterjet	1	1		1	1.3	1.30
2	Waterjet Cut	waterjet profile		181.532	Material - Aluminum	1	0.01	1.82
3	Machining Setup, Change	move to CNC	2	1		1	0.65	0.65
4	Machining	differential hole bore	2	62.21	Material - Aluminum	1	0.04	2.49
5	Machining Setup, Change	bit change	3	1		1	0.65	0.65
6	Machining	face down hanger jog	3	418.47	Material - Aluminum	1	0.04	16.74
7	Machining Setup, Change	bit change	4	1		1	0.65	0.65
8	Machining	chamfers and fillets	4	7	Material - Aluminum	1	0.04	0.28

Subtotal $24.58

Add Process

Figure G2: Material cost and manufacturing operations for UA-20 left differential hanger.

	Material	Use	Op Num	Size 1	Size 2	Area Name	Area	Length	Density	Quantity	Unit Cost	Subtotal
1	Aluminum, Premium (by Dimensions)	stock hanger material	1			Plate	101.526 in^2	0.625 in	0.098 lb/in^3	1	11.8467	11.85

Add Material

Subtotal $11.85

Processes

	Process	Use	Op Num	Quantity	Multiplier	Mult. Val.	Unit Cost	Subtotal
1	Machining Setup, Change	prepare waterjet	1	1		1	0.65	0.65
2	Waterjet Cut	waterjet profile	1	181.532	Material - Aluminum	1	0.01	1.82
3	Machining Setup, Change	move to CNC	2	1		1	0.65	0.65
4	Machining	differential hole bore	2	46.75	Material - Aluminum	1	0.04	1.87
5	Machining Setup, Change	bit change	3	1		1	0.65	0.65
6	Machining	face down hanger jog	3	418.47	Material - Aluminum	1	0.04	16.74
7	Machining Setup, Change	bit change	4	1		1	0.65	0.65
8	Machining	chamfers and fillets	4	7	Material - Aluminum	1	0.04	0.28

Add Process

Subtotal $23.31

Figure G3: Material cost and manufacturing operations for UA-20 left differential hanger.

	Material	Use	Op Num	Size 1	Size 2	Area Name	Area	Length	Density	Quantity	Unit Cost	Subtotal
1	Steel, Alloy (by Dimensions)	raw material for shaft	1			hex bar	139.76 mm^2	66.42 mm	7850 kg/m^3	1	0.164	0.16

Subtotal $0.16

Add Material

Processes

	Process	Use	Op Num	Quantity	Multiplier	Mult. Val.	Unit Cost	Subtotal
1	Machining Setup, Install and remove	setup lathe	1	1		1	1.3	1.30
2	Machining	machine shaft steps	1	1.38	Material - Steel	3	0.04	0.17
3	Machining Setup, Install and remove	flip shaft in lathe	2	1		1	1.3	1.30
4	Machining	machine shaft steps	2	1.6296	Material - Steel	3	0.04	0.20
5	Machining Setup, Install and remove	move to mill	3	1		1	1.3	1.30
6	Machining	detent and face shaft	3	0.246	Material - Steel	3	0.04	0.03
7	Machining Setup, Install and remove	flip shaft in mill	4	1		1	1.3	1.30
8	Machining	face shaft	4	0.144	Material - Steel	3	0.04	0.02
9	Threading, External (machining)	thread both sides of shaft	5	1.8	Material - Steel	3	0.1	0.54

Subtotal $6.16

Add Process

Figure G4: Material cost and manufacturing operations for UA-20 idler sprocket shaft.

	Material	Use	Op Num	Size 1	Size 2	Area Name	Area	Length	Density	Quantity	Unit Cost	Subtotal
1	Aluminum, Premium (by Dimensions)	stocket material for sprocket	1			plate	2560.64 mm^2	7.13 mm	2712 kg/m^3	1	0.208	0.21

Subtotal $0.21

Add Material

Processes

	Process	Use	Op Num	Quantity	Multiplier	Mult. Val.	Unit Cost	Subtotal
1	Machining Setup, Install and remove	waterjet profile	1	1		1	1.3	1.30
2	Waterjet Cut	waterjet profile	1	12.613	Material - Aluminum	1	0.01	0.13
3	Machining Setup, Install and remove	move to lathe	2	1		1	1.3	1.30
4	Machining	face and bore hole	2	3.7422	Material - Aluminum	1	0.04	0.15
5	Machining Setup, Install and remove	flip in lathe	3	1		1	1.3	1.30
6	Machining	face	3	0.9733	Material - Aluminum	1	0.04	0.04

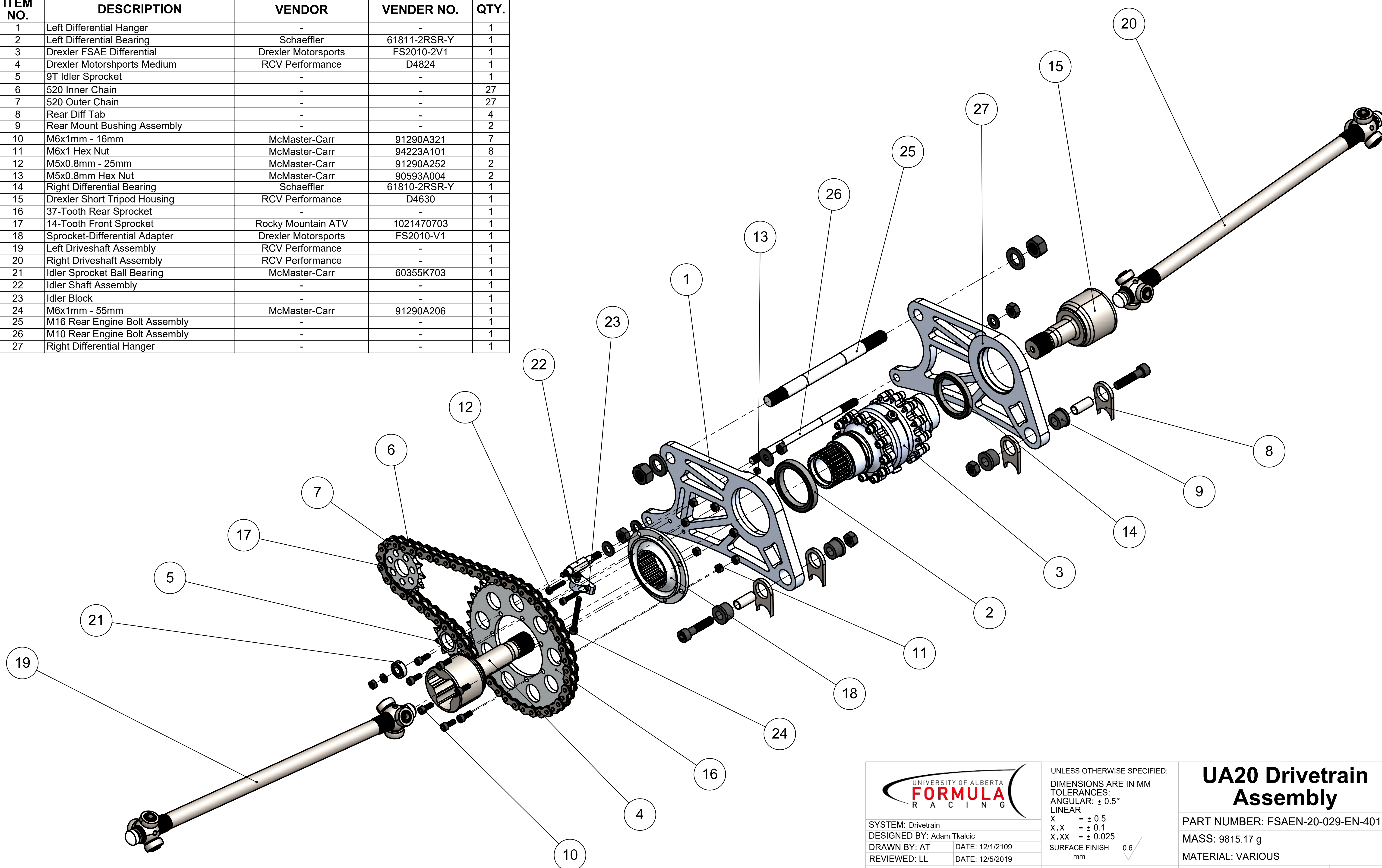
Subtotal $4.22

Add Process

Figure G5: Material cost and manufacturing operations for UA-20 idler sprocket.

Appendix H
Detailed Engineering Drawings

ITEM NO.	DESCRIPTION	VENDOR	VENDER NO.	QTY.
1	Left Differential Hanger	-	-	1
2	Left Differential Bearing	Schaeffler	61811-2RSR-Y	1
3	Drexler FSAE Differential	Drexler Motorsports	FS2010-2V1	1
4	Drexler Motorshports Medium	RCV Performance	D4824	1
5	9T Idler Sprocket	-	-	1
6	520 Inner Chain	-	-	27
7	520 Outer Chain	-	-	27
8	Rear Diff Tab	-	-	4
9	Rear Mount Bushing Assembly	-	-	2
10	M6x1mm - 16mm	McMaster-Carr	91290A321	7
11	M6x1 Hex Nut	McMaster-Carr	94223A101	8
12	M5x0.8mm - 25mm	McMaster-Carr	91290A252	2
13	M5x0.8mm Hex Nut	McMaster-Carr	90593A004	2
14	Right Differential Bearing	Schaeffler	61810-2RSR-Y	1
15	Drexler Short Tripod Housing	RCV Performance	D4630	1
16	37-Tooth Rear Sprocket	-	-	1
17	14-Tooth Front Sprocket	Rocky Mountain ATV	1021470703	1
18	Sprocket-Differential Adapter	Drexler Motorsports	FS2010-V1	1
19	Left Driveshaft Assembly	RCV Performance	-	1
20	Right Driveshaft Assembly	RCV Performance	-	1
21	Idler Sprocket Ball Bearing	McMaster-Carr	60355K703	1
22	Idler Shaft Assembly	-	-	1
23	Idler Block	-	-	1
24	M6x1mm - 55mm	McMaster-Carr	91290A206	1
25	M16 Rear Engine Bolt Assembly	-	-	1
26	M10 Rear Engine Bolt Assembly	-	-	1
27	Right Differential Hanger	-	-	1



UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 12/1/2109
REVIEWED: LL — DATE: 12/5/2019
COMMENTS: None.

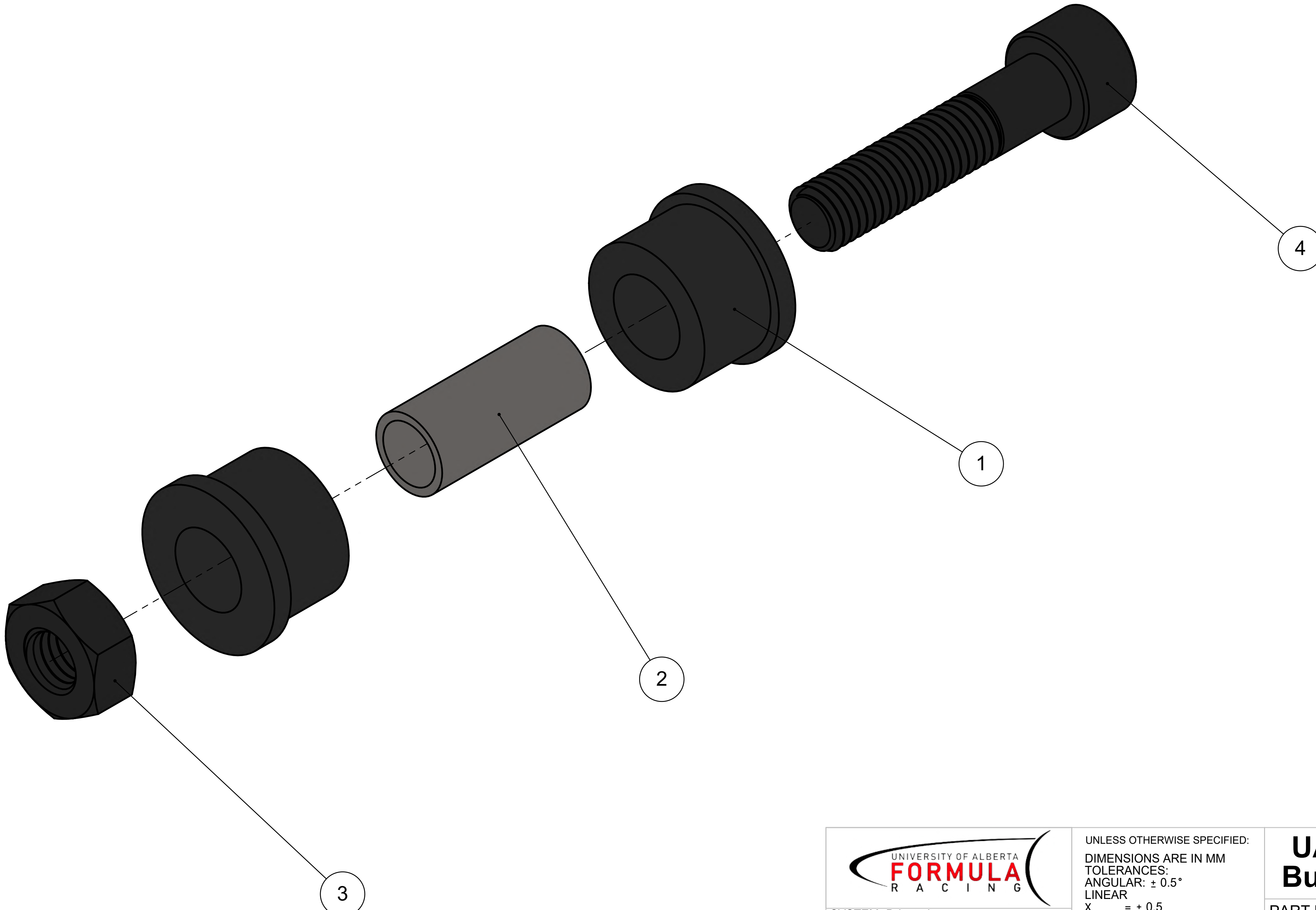
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 Drivetrain Assembly

REV: AA

PART NUMBER: FSAEN-20-029-EN-4011
MASS: 9815.17 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 7:32
DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	Universal Flange Bushing	Energy Suspension Parts	9.9176	2
2	Bushing Sleeve	McMaster-Carr	9922K18	1
3	M10x1.5mm Hex Nut	McMaster-Carr	94223A103	1
4	M10x1.5mm - 45mm	McMaster-Carr	91290A530	1



UA20 Rear Mount Bushing Assembly

REV: AA

UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain

DESIGNED BY: Adam Tkalcic

DRAWN BY: AT — DATE: 12/1/2019

REVIEWED: LL — DATE: 12/5/2019

COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

PART NUMBER:

MASS: 28.30 g

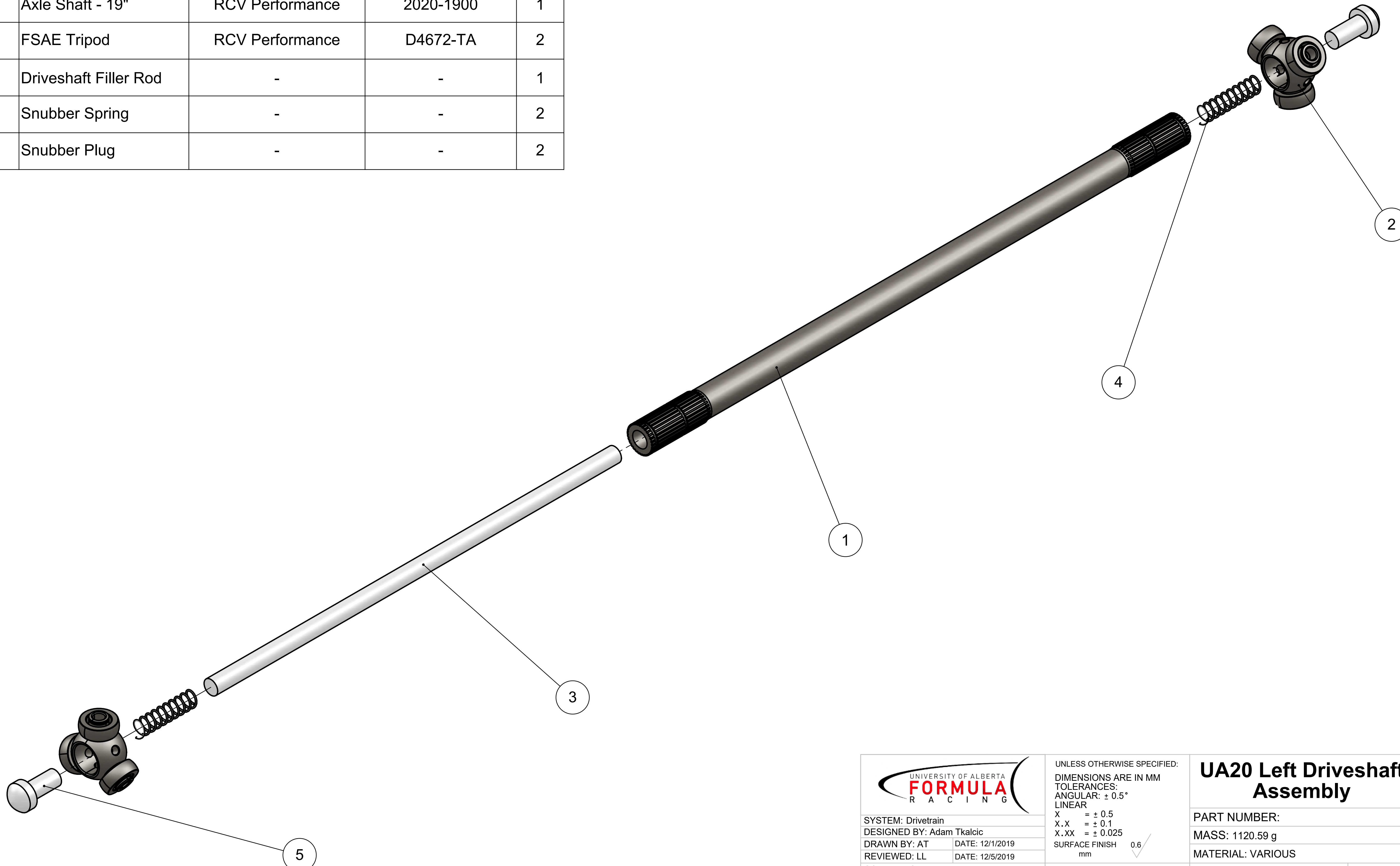
MATERIAL: VARIOUS

SHEET SIZE: B

SCALE: 3:2

DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	Axle Shaft - 19"	RCV Performance	2020-1900	1
2	FSAE Tripod	RCV Performance	D4672-TA	2
3	Driveshaft Filler Rod	-	-	1
4	Snubber Spring	-	-	2
5	Snubber Plug	-	-	2



UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT DATE: 12/1/2019
REVIEWED: LL DATE: 12/5/2019
COMMENTS:
None.

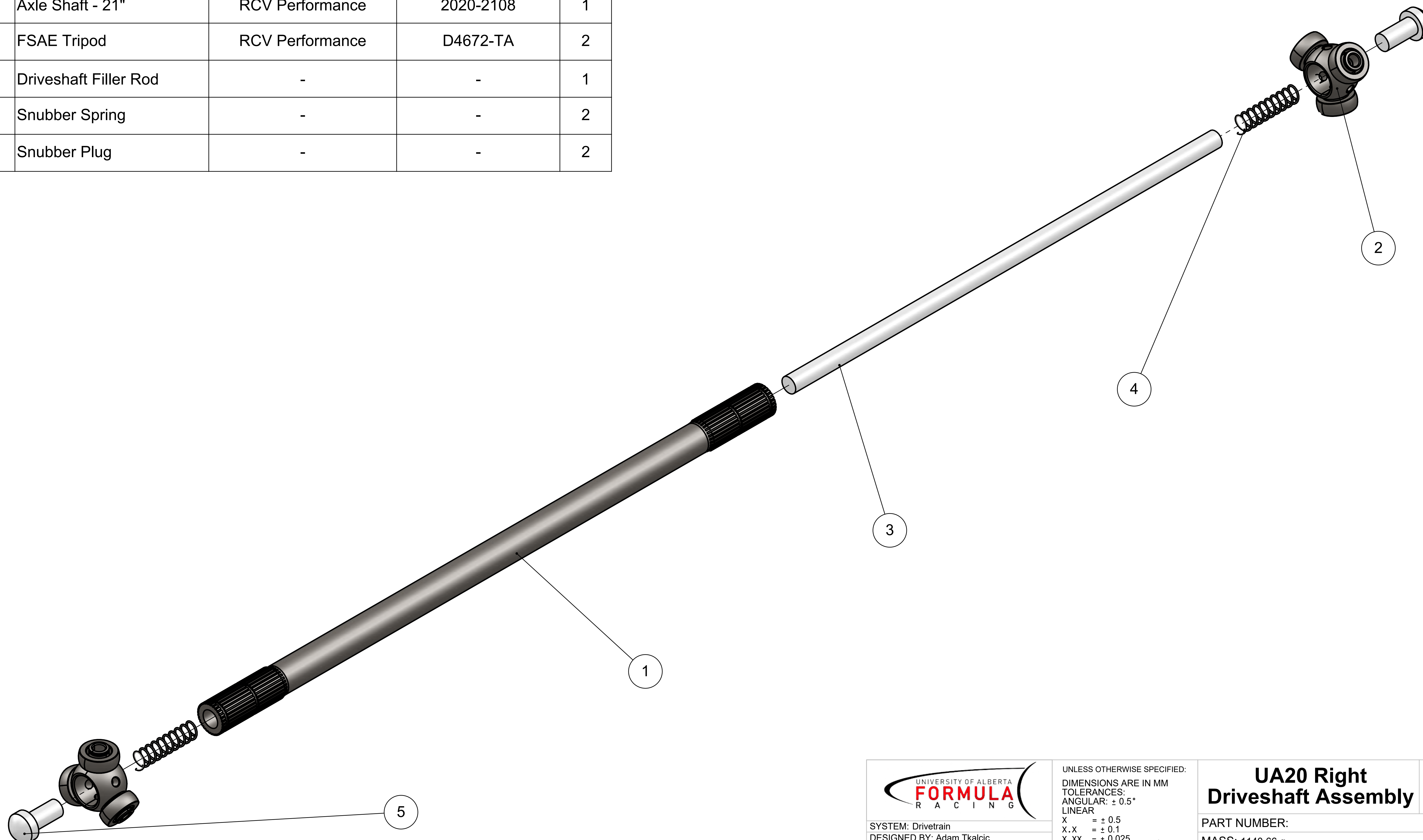
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 Left Driveshaft Assembly

REV: AA

PART NUMBER:
MASS: 1120.59 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 7:16
DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	Axle Shaft - 21"	RCV Performance	2020-2108	1
2	FSAE Tripod	RCV Performance	D4672-TA	2
3	Driveshaft Filler Rod	-	-	1
4	Snubber Spring	-	-	2
5	Snubber Plug	-	-	2



UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT DATE: 12/1/2019
REVIEWED: LL DATE: 12/5/2019
COMMENTS:
None.

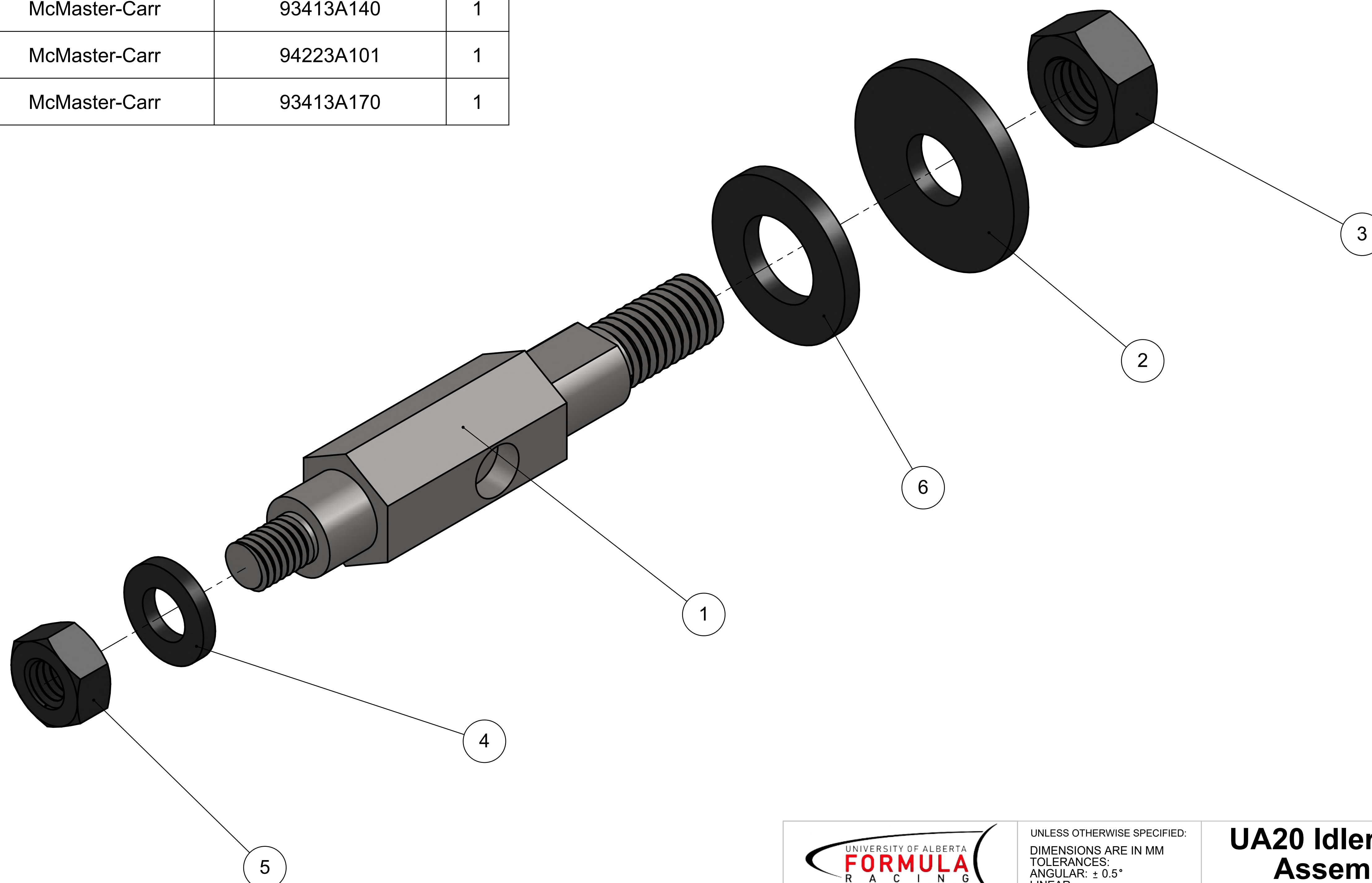
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH mm 0.6

UA20 Right Driveshaft Assembly

REV: AA

PART NUMBER:
MASS: 1140.66 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 7:16
DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	Idler Shaft	-	-	1
2	M8 Oversized Washer	McMaster-Carr	91116A380	1
3	M8x1.25 Hex Nut	McMaster-Carr	94223A102	1
4	M6 Washer	McMaster-Carr	93413A140	1
5	M6x1 Hex Nut	McMaster-Carr	94223A101	1
6	M10 Washer	McMaster-Carr	93413A170	1



UNIVERSITY OF ALBERTA FORMULA RACING

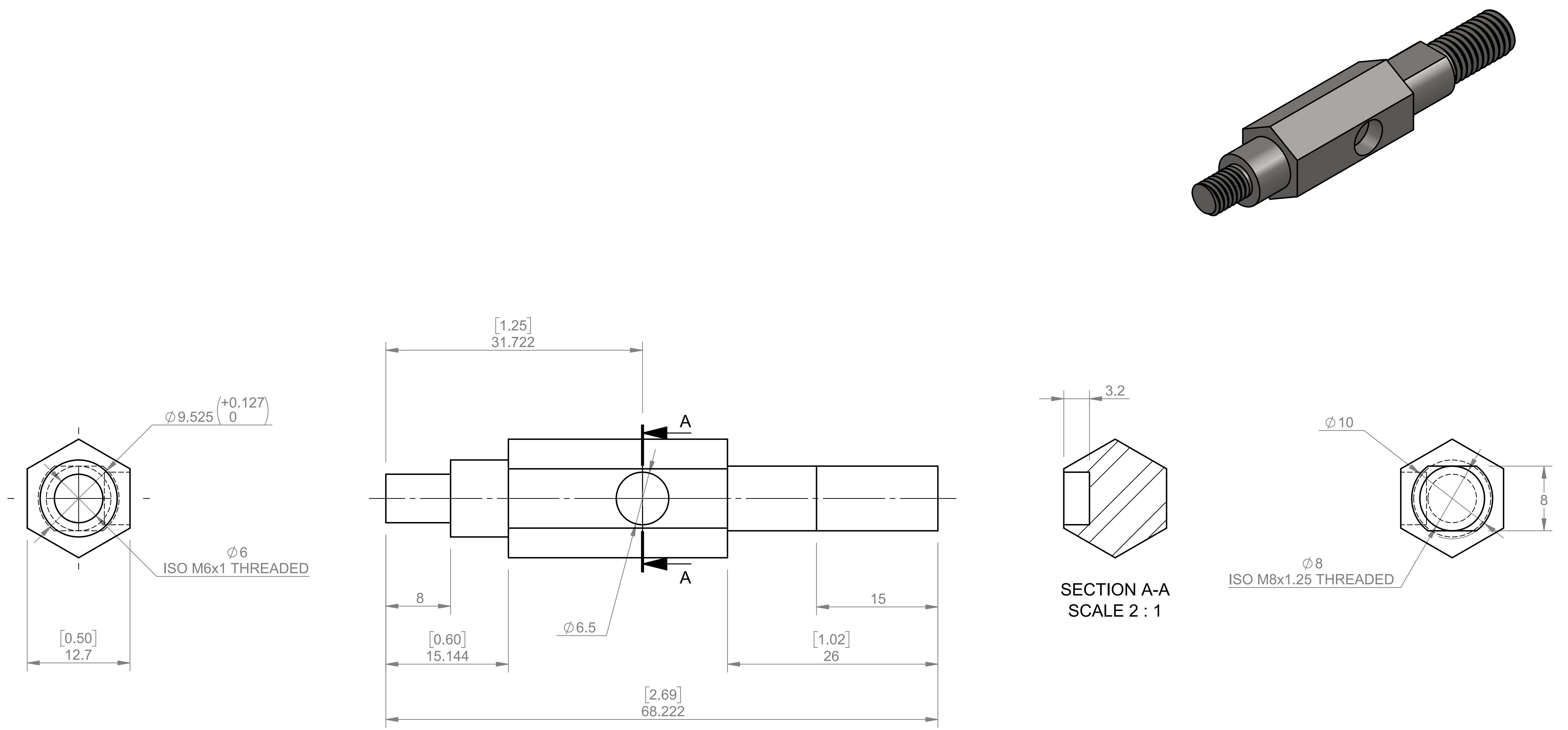
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT DATE: 12/1/2019
REVIEWED: LL DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 Idler Shaft Assembly

REV: AA

PART NUMBER:
MASS: 47.83 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 2:1
DO NOT SCALE DRAWING



UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 12/2/2019
REVIEWED: LL — DATE: 12/5/2019
COMMENTS:
None.

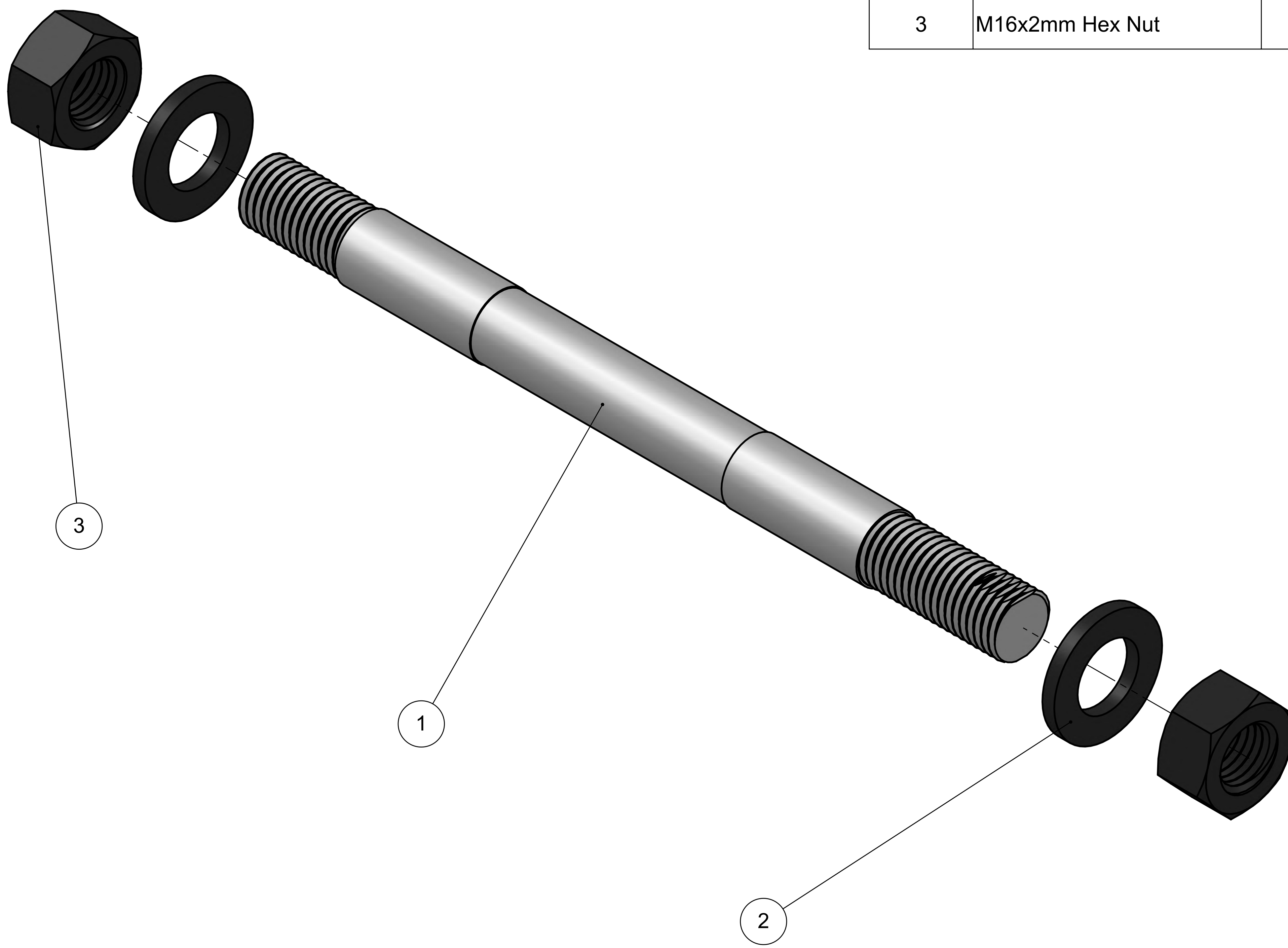
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 Idler Shaft

REV: AA

PART NUMBER:
MASS: 46.63 g
MATERIAL: AISI 4140 STEEL, HEX BAR
SHEET SIZE: B
SCALE: 2:1
DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	M16 Engine Bolt	-	-	1
2	M16 Washer	McMaster-Carr	98040A109	2
3	M16x2mm Hex Nut	McMaster-Carr	94223A105	2



UNIVERSITY OF ALBERTA FORMULA RACING

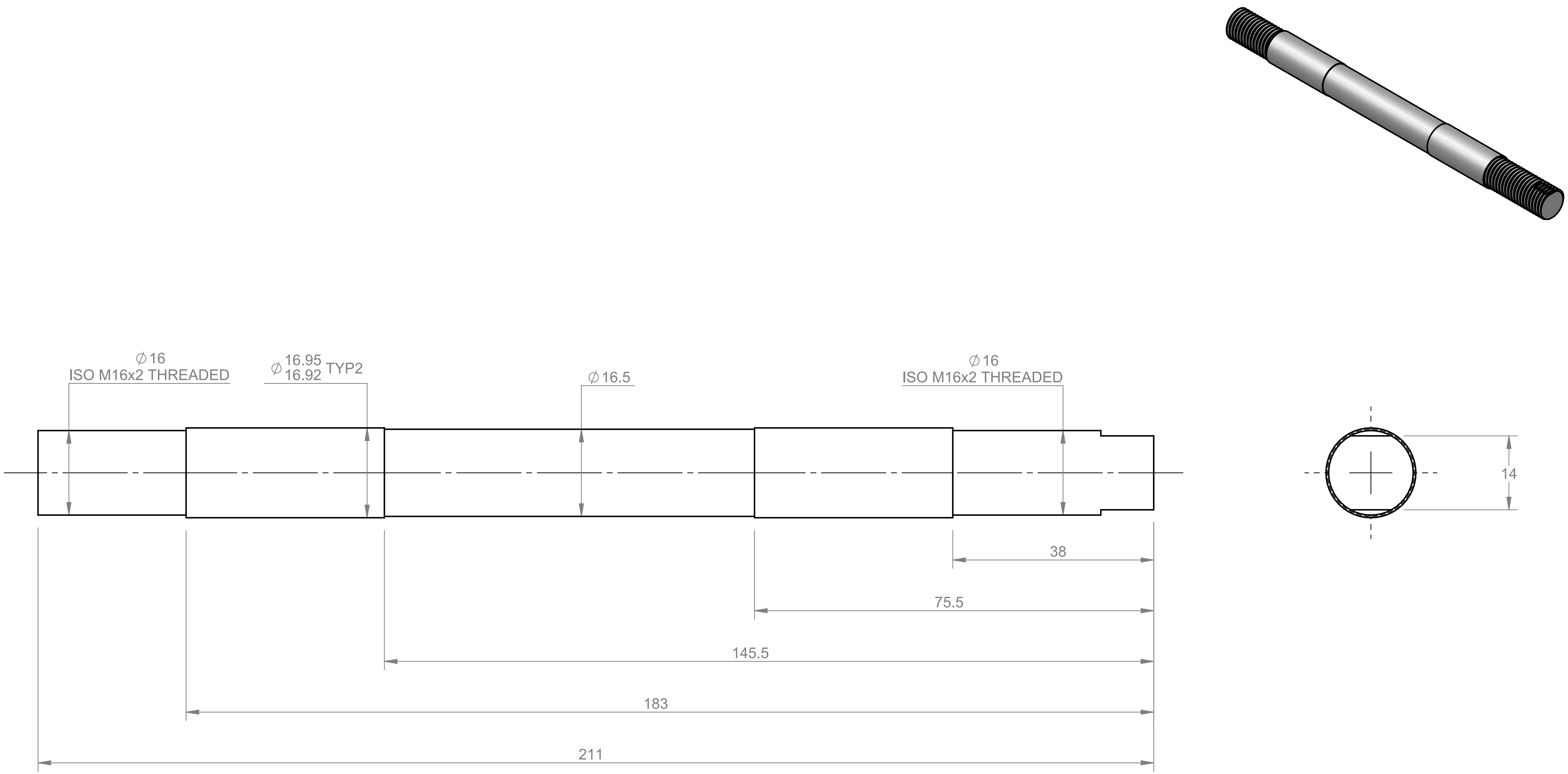
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT DATE: 12/1/2019
REVIEWED: LL DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6
mm

UA20 M16 Engine Bolt Assembly

REV: AA

PART NUMBER:
MASS: 351.20 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 1:1
DO NOT SCALE DRAWING



UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT DATE: 12/2/2019
REVIEWED: LL DATE: 12/5/2019
COMMENTS:
None.

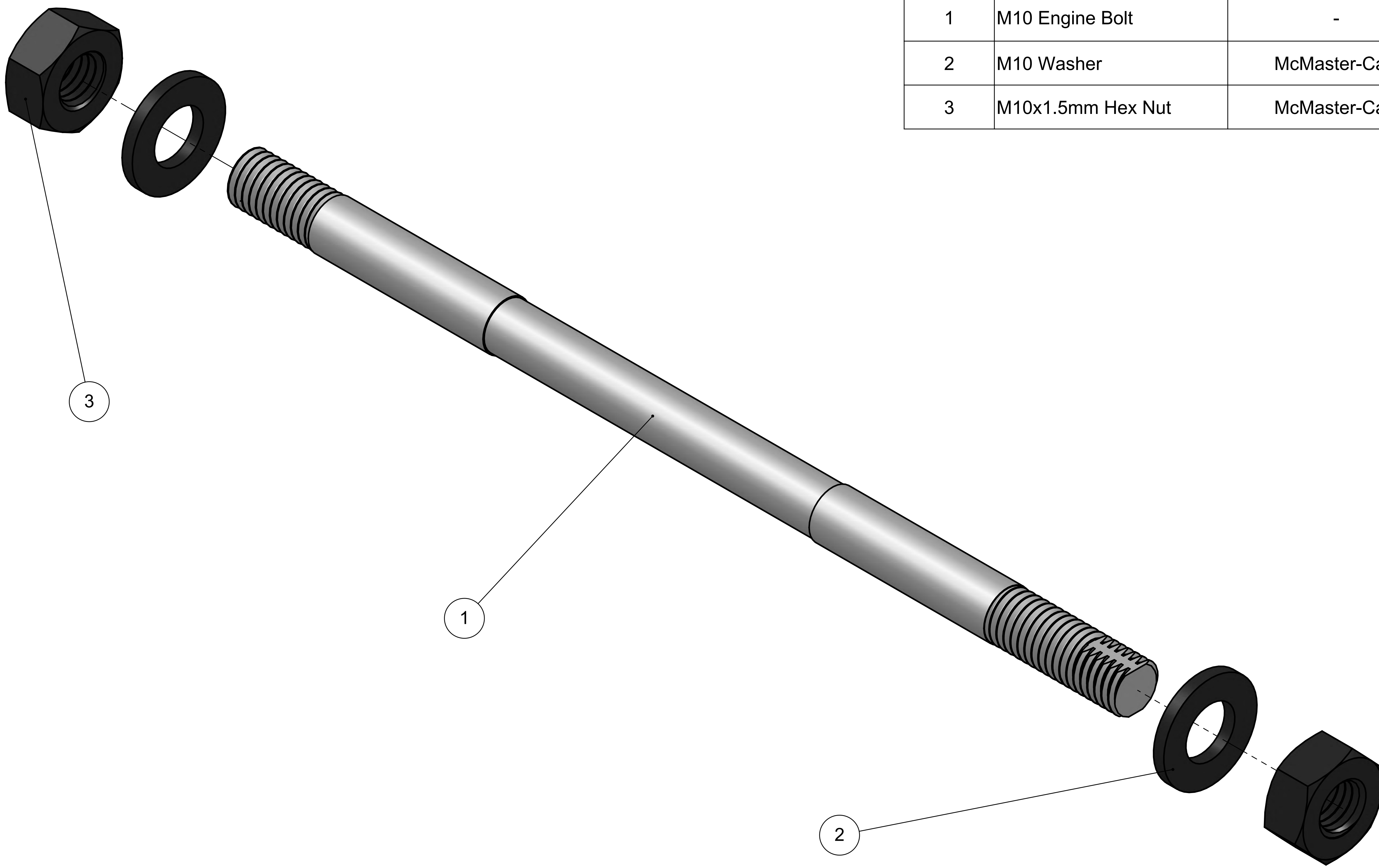
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 M16 Engine Bolt

REV:

PART NUMBER:
MASS: 354.48 g
MATERIAL: AISI 4140 STEEL
SHEET SIZE: B
SCALE: 5:4
DO NOT SCALE DRAWING

ITEM NO.	DESCRIPTION	VENDOR	VENDOR NO.	QTY.
1	M10 Engine Bolt	-	-	1
2	M10 Washer	McMaster-Carr	93413A170	2
3	M10x1.5mm Hex Nut	McMaster-Carr	94223A103	2



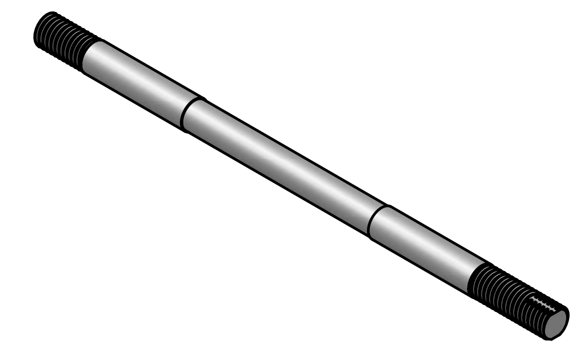
UA20 M10 Engine Bolt Assembly

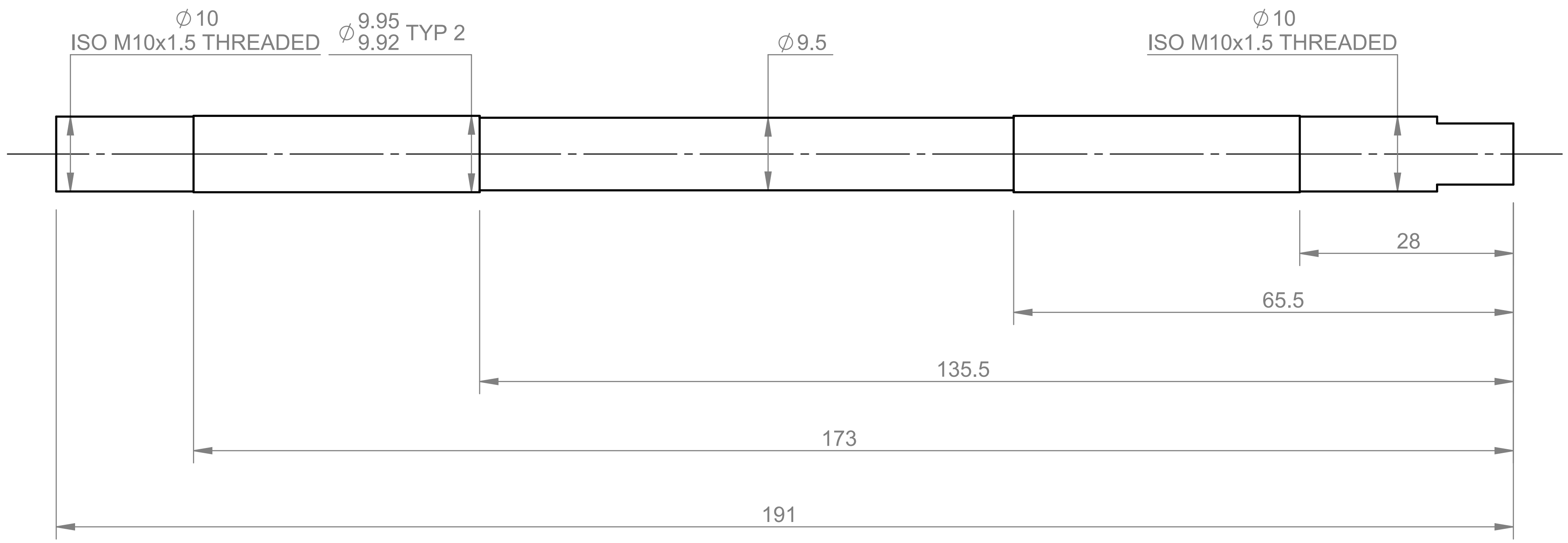
REV: AA

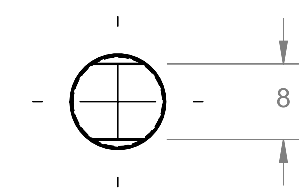
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 12/1/2019
REVIEWED: LL — DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

PART NUMBER:
MASS: 111.27 g
MATERIAL: VARIOUS
SHEET SIZE: B
SCALE: 3:2
DO NOT SCALE DRAWING

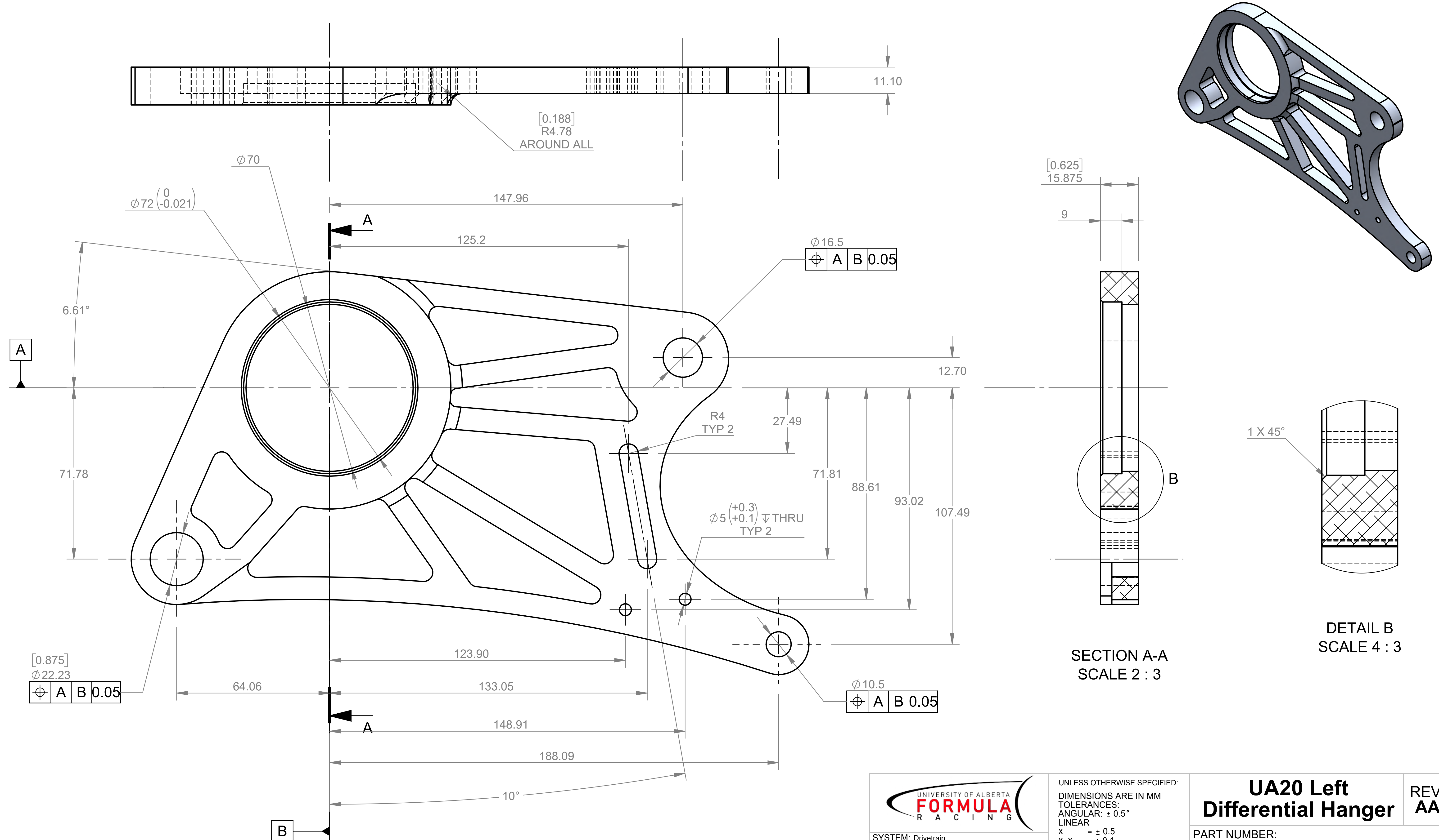






UA20 M10 Engine Bolt

UNIVERSITY OF ALBERTA FORMULA RACING	UNLESS OTHERWISE SPECIFIED:	REV:
SYSTEM: Drivetrain	DIMENSIONS ARE IN MM	PART NUMBER:
DESIGNED BY: Adam Tkalcic	TOLERANCES: ANGULAR: ± 0.5°	MASS: 111.88 g
DRAWN BY: AT — DATE: 12/2/2019	LINEAR X = ± 0.5; X.X = ± 0.1; X.XX = ± 0.025	MATERIAL: AISI 4140 STEEL
REVIEWED: LL — DATE: 12/5/2019	SURFACE FINISH 0.6 mm	SHEET SIZE: B
COMMENTS: None.		SCALE: 5:4 DO NOT SCALE DRAWING



NOTE: PART PROFILE TO BE WATERJET CUT OR CNC MILLED. ONLY CRITICAL DIMENSIONS SHOWN.

UNIVERSITY OF ALBERTA FORMULA RACING

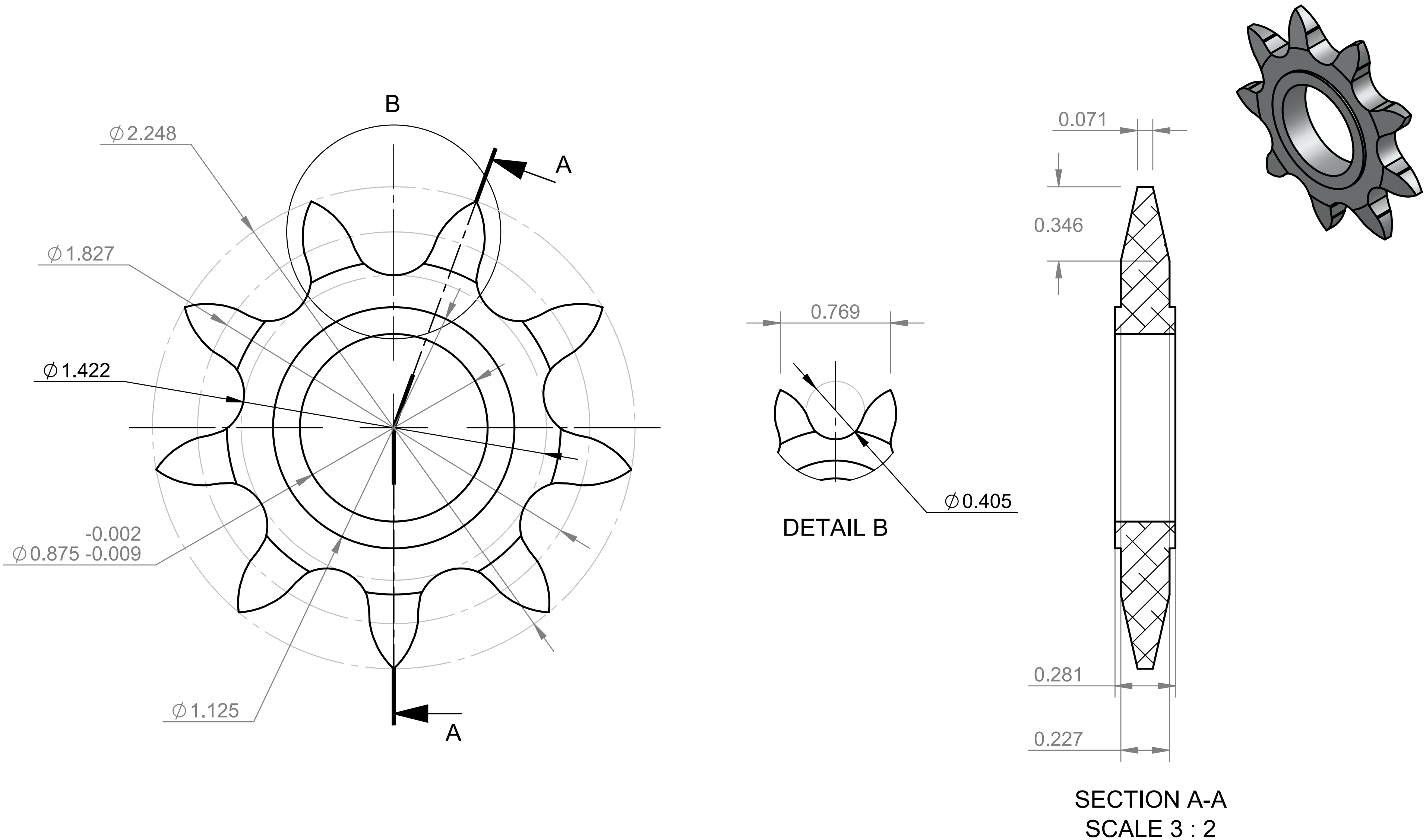
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 12/2/2019
REVIEWED: LL — DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6 mm

UA20 Left Differential Hanger

REV: AA

PART NUMBER:
MASS: 586.20 g
MATERIAL: 7075-T6 (SN)
SHEET SIZE: B
SCALE: 2:3
DO NOT SCALE DRAWING



SPECIFICATIONS:
520 SERIES CHAIN
0.625" PITCH
0.250" ROLLER WIDTH
0.400" ROLLER DIAMETER

ONLY CRITICAL DIMENSIONS SHOWN

UNIVERSITY OF ALBERTA FORMULA RACING

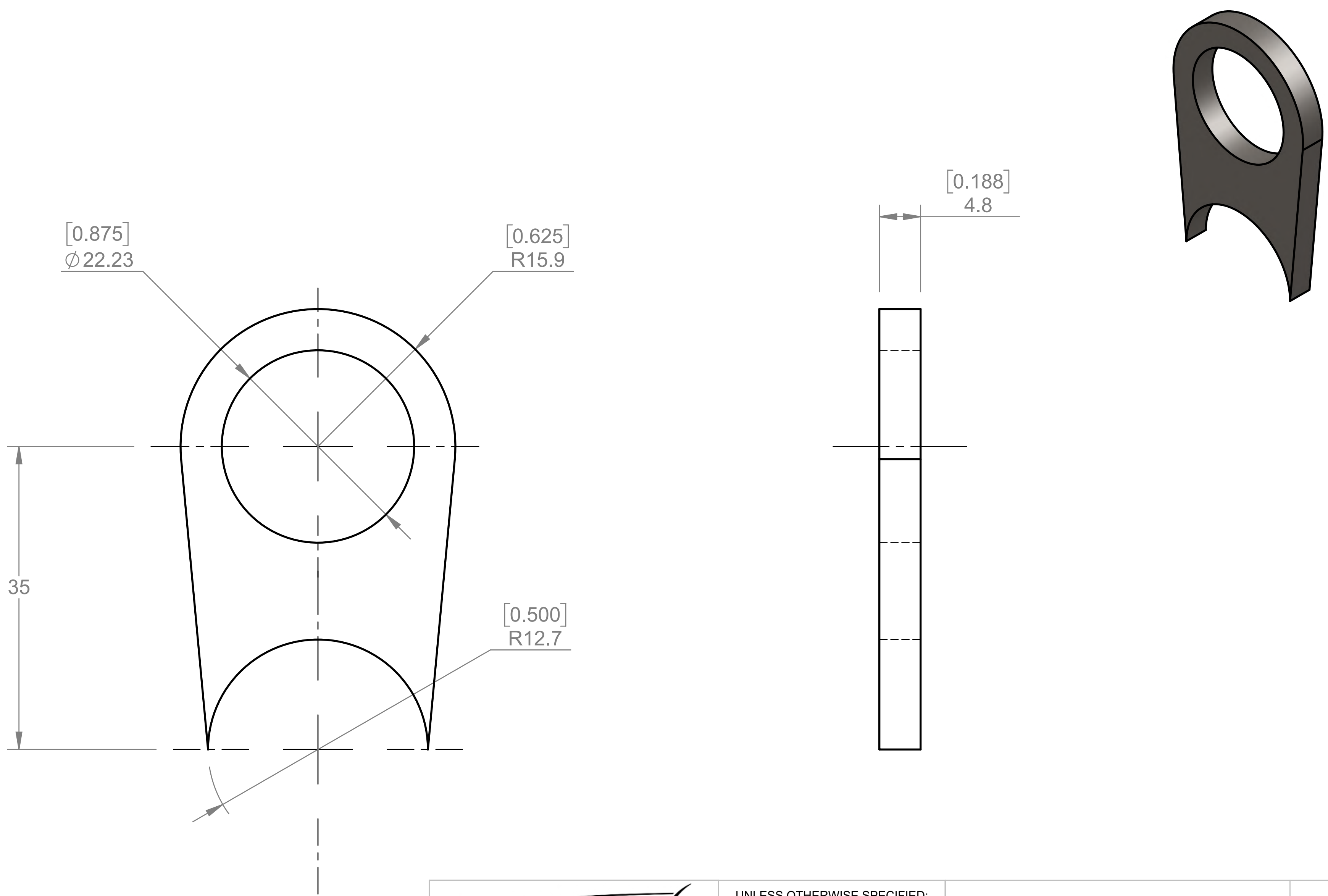
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT | DATE: 11/11/2019
REVIEWED: LL | DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.01
X.X = ± 0.005
X.XX = ± 0.001
SURFACE FINISH 32 µin

9T Idler Sprocket

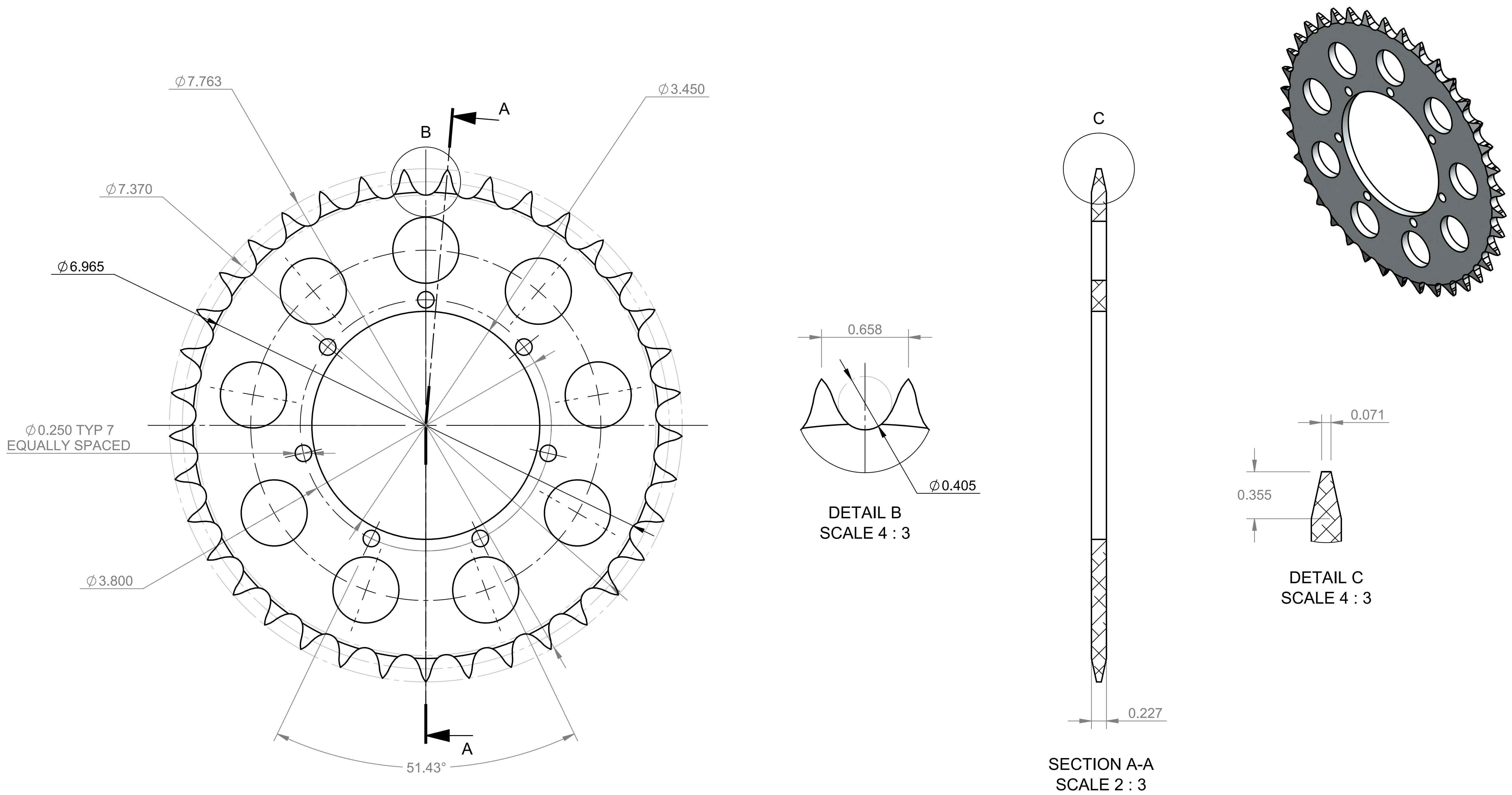
REV: AB

PART NUMBER:
MASS: 0.039 g
MATERIAL: 7075-T6 (SN)
SHEET SIZE: A
SCALE: 1:1
DO NOT SCALE DRAWING



NOTE: QTY. 4

UNIVERSITY OF ALBERTA FORMULA RACING	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH mm 0.6	**UA20 Rear Drivetrain Mounting Tab**	REV: **AA**
SYSTEM: Drivetrain		PART NUMBER:	
DESIGNED BY: Adam Tkalcic		MASS: 28.30 g	
DRAWN BY: AT — DATE: 12/2/2019		MATERIAL: AISI 4140 STEEL	
REVIEWED: LL — DATE: 12/5/2019			
COMMENTS: None.		SHEET SIZE: A	SCALE: 2:1 DO NOT SCALE DRAWING



SPECIFICATIONS:
520 SERIES CHAIN
0.625" PITCH
0.250" ROLLER WIDTH
0.400" ROLLER DIAMETER

PART TO BE CUT FROM 1/4" 7075 AL, FACED DOWN TO 0.227"
ONLY CRITICAL DIMENSIONS SHOWN

UNIVERSITY OF ALBERTA FORMULA RACING

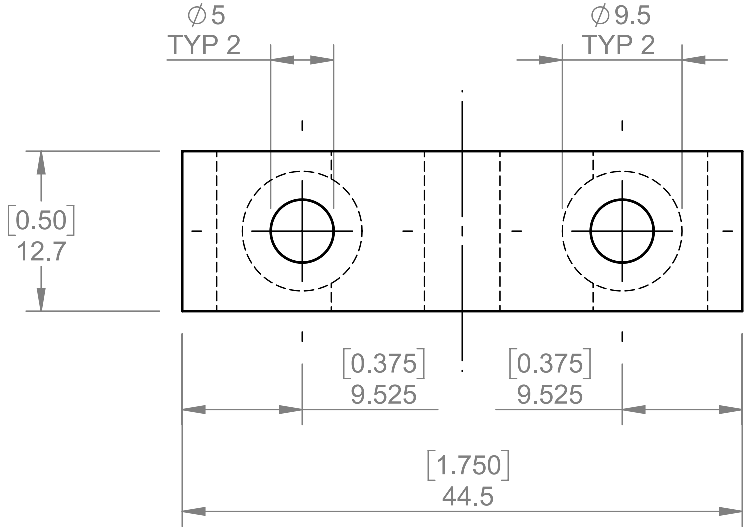
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 11/11/2019
REVIEWED: LL — DATE: 12/5/2019
COMMENTS:
None.

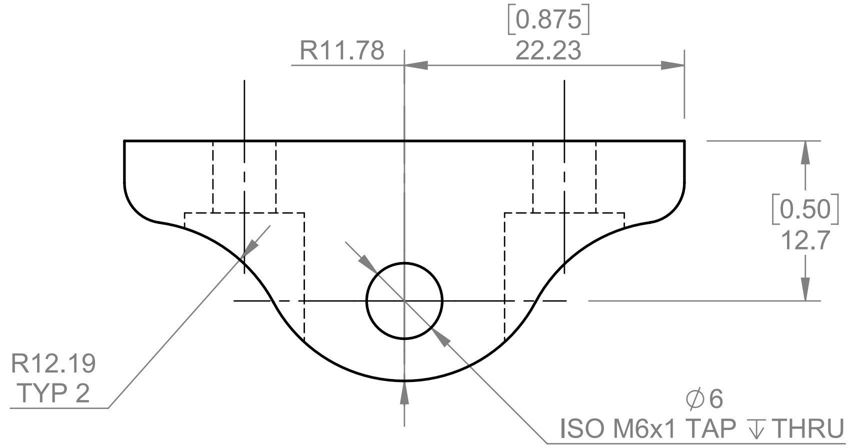
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.01
X.X = ± 0.005
X.XX = ± 0.001
SURFACE FINISH 32 μin

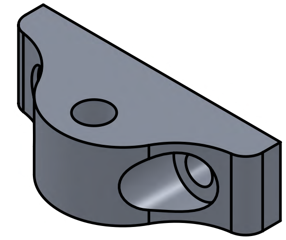
37T Rear Sprocket

REV: AA

PART NUMBER:
MASS: 0.546 g
MATERIAL: 7075-T6 (SN)
SHEET SIZE: B
SCALE: 1:2
DO NOT SCALE DRAWING







UNIVERSITY OF ALBERTA FORMULA RACING

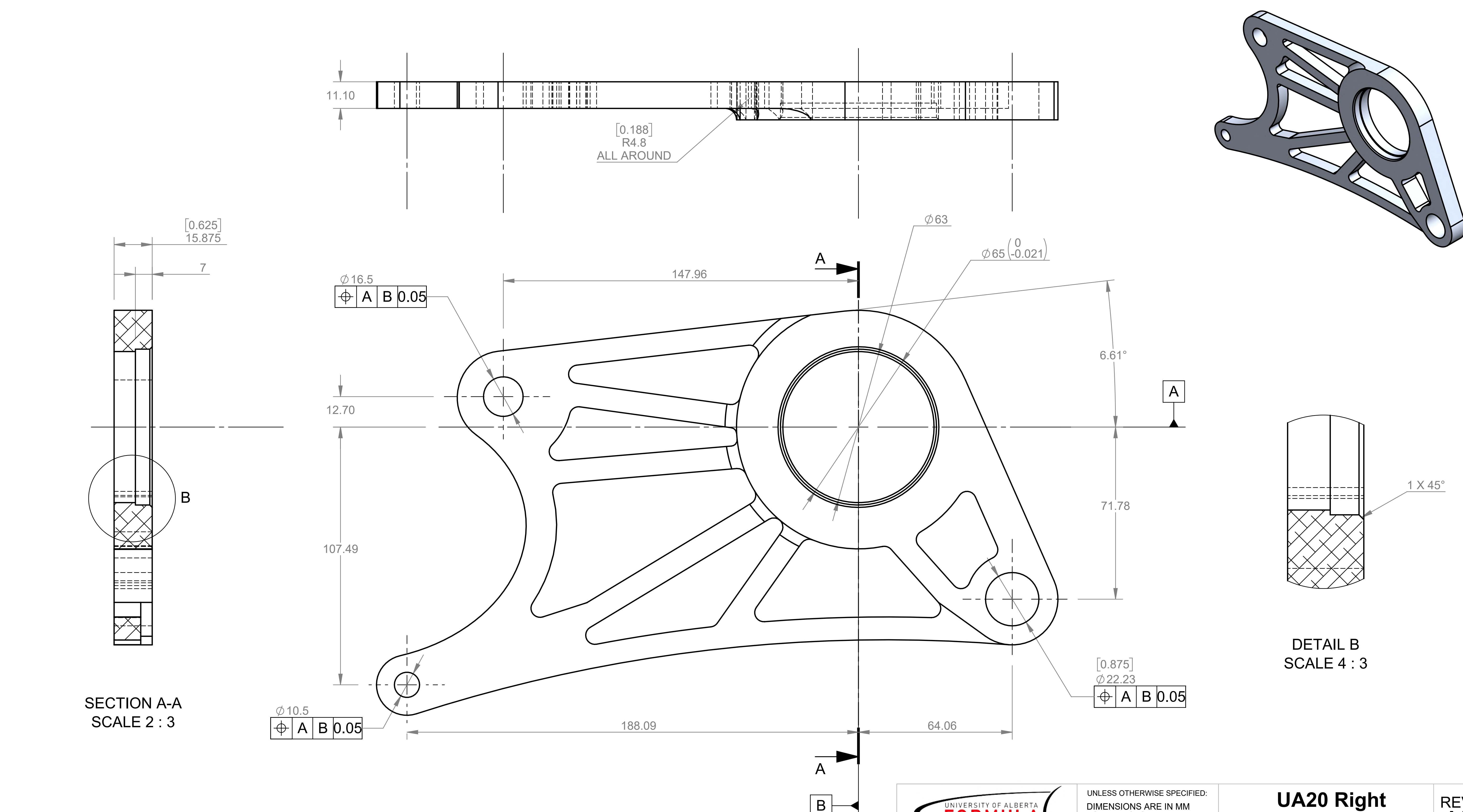
SYSTEM: Drivetrain
DESIGNED BY: Adam Tkalcic
DRAWN BY: AT — DATE: 12/2/2019
REVIEWED: LL — DATE: 12/5/2019
COMMENTS:
None.

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH 0.6
mm

UA20 Idler Block

REV: AA

PART NUMBER:
MASS: 15.98 g
MATERIAL: 7075-T6 (SN)
SHEET SIZE: A
SCALE: 5:3
DO NOT SCALE DRAWING



NOTE: PART PROFILE TO BE WATERJET CUT OR CNC MILLED. ONLY CRITICAL DIMENSIONS SHOWN.

UNIVERSITY OF ALBERTA FORMULA RACING

SYSTEM: Drivetrain		UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH 0.6 mm	**UA20 Right Differential Hanger** — REV: **AA**
DESIGNED BY: Adam Tkalcic			PART NUMBER:
DRAWN BY: AT	DATE: 12/2/2019		MASS: 611.73 g
REVIEWED: LL	DATE: 12/5/2019		MATERIAL: 7075-T6 (SN)
COMMENTS: None.			SHEET SIZE: B — SCALE: 2:3 DO NOT SCALE DRAWING

Appendix I

Design Authorities

a. The FMEA study must contain a detailed description of all the potential failure modes that can occur, the strategy that is used to detect these failures and the tests that have been conducted to prove that the detection strategy works.

b. The failures modes must include but are not limited to the failure of the sensor, sensor signals being out of range, corruption of the message and loss of messages and the associated time outs.

c. In all cases a sensor failure must immediately shutdown power to the motor(s).

T.5 POWERTRAIN

T.5.1 Transmission and Drive

Any transmission and drivetrain may be used.

T.5.2 Drivetrain Shields and Guards

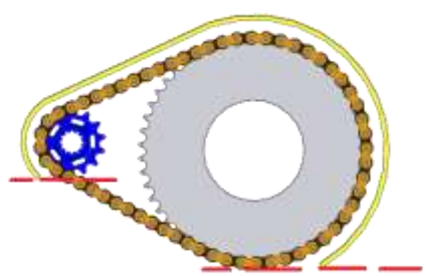
T.5.2.1 Exposed high speed final drivetrain equipment such as Continuously Variable Transmissions (CVTs), sprockets, gears, pulleys, torque converters, clutches, belt drives, clutch drives and electric motors, must be fitted with scatter shields intended to contain drivetrain parts in case of failure.

T.5.2.2 The final drivetrain shield must:

a. Be made with solid material (not perforated)

b. Cover the chain or belt from the drive sprocket to the driven sprocket/chain wheel/belt or pulley.

c. Start and end no higher than parallel to the lowest point of the chain wheel/belt/pulley:



T.5.2.3 Body panels or other existing covers are not acceptable unless constructed per **T.5.2.7** / **T.5.2.8**

T.5.2.4 Frame members or existing components that exceed the scatter shield material requirements may be used as part of the shield.

T.5.2.5 Scatter shields may be composed of multiple segments. Any gaps must be small (< 3 mm)

T.5.2.6 If equipped, the engine drive sprocket cover may be used as part of the scatter shield system.

T.5.2.7 Chain Drive - Scatter shields for chains must:

a. Be made of 2.66 mm (0.105 inch) minimum thickness steel (no alternatives are allowed)

b. Have a minimum width equal to three times the width of the chain

c. Be centered on the center line of the chain

d. Remain aligned with the chain under all conditions

T.5.2.8 Non-metallic Belt Drive - Scatter shields for belts must:

a. Be made from 3.0 mm minimum thickness aluminum alloy 6061-T6

b. Have a minimum width that is equal to 1.7 times the width of the belt.

c. Be centered on the center line of the belt

d. Remain aligned with the belt under all conditions.

T.5.2.9 Attachment Fasteners - All fasteners attaching scatter shields and guards must be 6mm or 1/4" minimum diameter **Critical Fasteners**, see **T.8.2**

T.5.2.10 Finger Guards

a. Must cover any drivetrain parts that spin while the vehicle is stationary with the engine running.

b. Must be made of material sufficient to resist finger forces.

c. Mesh or perforated material may be used but must prevent the passage of a 12 mm diameter object through the guard.

T.5.3 Coolant Fluid

T.5.3.1 Water cooled engines must use only plain water with no additives of any kind.

T.5.3.2 Coolant for electric motors, accumulators or HV electronics must be one of:

- plain water with no additives
- oil

T.5.4 System Sealing

T.5.4.1 Any cooling or lubrication system must be sealed to prevent leakage.

T.5.4.2 The vehicle must be capable of being tilted to a 45° angle without leaking fluid of any type.

T.5.4.3 Flammable liquid leaks must not be allowed to accumulate.

T.5.4.4 At least 2 holes of minimum diameter 25 mm each must be provided in the lowest part of the structure or belly pan in such a way as to prevent accumulation of liquids and/or vapors.

T.5.4.5 Absorbent material and open collection devices (regardless of material) are prohibited in compartments containing engine, drivetrain, exhaust and fuel systems below the highest point on the exhaust system.

T.5.5 Catch Cans

T.5.5.1 Separate catch cans must be employed to retain fluids from any vents for the engine coolant system and engine lubrication system.

Each catch can must have a minimum volume of 10% of the fluid being contained or 0.9 liter, whichever is greater.

T.5.5.2 Any vent on other systems containing liquid lubricant or coolant, including a differential, gearbox, or electric motor, must have a catch can with a minimum volume of 10% of the fluid being contained or 0.5 liter, whichever is greater.

T.5.5.3 Catch cans must be:

a. Capable of containing boiling water without deformation

b. Located rearwards of the firewall below the driver's shoulder level

c. Positively retained, using no tie wraps or tape

T.5.5.4 Any catch can on the cooling system must vent through a hose with a minimum internal diameter of 3 mm down to the bottom levels of the Frame.

T.7.3 Front Mounted

T.7.3.1 In plan view, any part of any Aerodynamic Device must be:

a. No more than 700 mm forward of the fronts of the front tires

b. Within a vertical plane parallel to the centerline of the chassis touching the outside of the front tires at the height of the hubs.

T.7.3.2 When viewed from the front of the vehicle, the part of the front wheels/tires that are more than 250 mm above ground level must be unobstructed when measured without a driver in the vehicle.

T.7.4 Rear Mounted

T.7.4.1 In plan view, any part of any Aerodynamic Device must be:

a. No more than 250 mm rearward of the rear of the rear tires

b. No further forward than a vertical plane through the rearmost portion of the front face of the driver head restraint support, excluding any padding, set (if adjustable) in its fully rearward position (excluding undertrays).

c. Inboard of two vertical planes parallel to the centerline of the chassis touching the inside of the rear tires at the height of the hub centerline.

T.7.4.2 In side elevation, any part of an Aerodynamic Device must be no higher than 1.2 meters above the ground when measured without a driver in the vehicle

T.7.5 Between Wheels

T.7.5.1 Between the centerlines of the front and rear wheel axles, an Aerodynamic Device may extend outboard in plan view to a line drawn connecting the outer surfaces of the front and rear tires at the height of the wheel centers

T.7.5.2 Except as permitted under **T.7.4.1 above**, any Aerodynamic Devices, or other bodywork, located between the transverse vertical planes positioned at the front and rear axle centerlines must not exceed a height of 500 mm above the ground when measured without a driver in the vehicle.

Bodywork within vertical fore and aft planes set at 400 mm outboard from the centerline on each side of the vehicle is excluded from this requirement.

T.8 FASTENERS

T.8.1 Critical Fasteners

A fastener (bolt, screw, pin, etc) used in a location designated as such in the applicable rule

T.8.2 Critical Fastener Requirements

T.8.2.1 Any Critical Fastener must meet, at minimum, one of the following:

a. SAE Grade 5

b. Metric Grade 8.8

c. AN/MS Specifications

d. Equivalent to or better than above, as approved by a Rules Question or at Technical Inspection

T.8.2.2 All threaded Critical Fasteners must be one of the following:

- Hex head
- Hexagonal recessed drive (Socket Head Cap Screws or Allen screws/bolts)

T.8.2.3 All Critical Fasteners must be secured from unintentional loosening by the use of **Positive Locking Mechanisms** see **T.8.3**

T.8.2.4 A minimum of two full threads must project from any lock nut.

T.8.2.5 Some Critical Fastener applications have additional requirements that are provided in the applicable section.

T.8.3 Positive Locking Mechanisms

T.8.3.1 Positive Locking Mechanisms are defined as those which:

a. Technical Inspectors / team members can see that the device/system is in place (visible).

b. Do not rely on the clamping force to apply the locking or anti vibration feature.

Meaning If the fastener begins to loosen, the locking device still prevents the fastener coming completely loose

T.8.3.2 Acceptable Positive Locking Mechanisms include:

a. Correctly installed safety wiring

b. Cotter pins

c. Nylon lock nuts (where temperature does not exceed 80°C)

d. Prevailing torque lock nuts

Lock washers, bolts with nylon patches and thread locking compounds (Loctite®), DO NOT meet the positive locking requirement.

T.8.4 Requirements for All Fasteners

Adjustable tie rod ends must be constrained with a jam nut to prevent loosening.

T.9 ELECTRICAL EQUIPMENT

T.9.1 Low Voltage Batteries

T.9.1.1 All batteries and onboard power supplies must be attached securely to the frame.

T.9.1.2 All Low Voltage batteries must have overcurrent protection that trips at or below the maximum specified discharge current of the cells.

T.9.1.3 The hot (ungrounded) terminal must be insulated.

T.9.1.4 Any wet cell battery located in the driver compartment must be enclosed in a nonconductive marine type container or equivalent.

T.9.1.5 Battery packs based on Lithium chemistry must:

a. Have a rigid, sturdy and flame resistant casing

b. Be separated from the driver by a Firewall as specified in **T.1.8**

T.9.1.6 All batteries using chemistries other than lead acid must be presented at Technical Inspection with markings identifying it for comparison to a datasheet or other documentation proving the pack and supporting electronics meet all rules requirements

Steel Axle Kit

This is the new and improved steel axle set that is being produced by RCV Performance. Literally every single component has been FEA optimized for reduced weight and maximum strength with a great deal of destructive testing done to follow up the FEA optimization. Compared to the original products, more than 3 pounds per axle has been shaved off with no loss in performance. That being said, the new axles weigh from four to five lbs depending on length. To achieve this, each tripod housing was profile milled around the tripod shape and the stub shafts were bored out. Then the tripod was modified to improve strength and reduce weight. Next, the axle was changed from a gun drilled billet to a formed DOM tube for reduced weight, faster turnaround time, and improved concentricity. Last, the plunging / centering system was discarded based on testing and observations. The result is one of the lightest steel axle assemblies around, all for one of the best prices you can find.

We have developed **more** options for the inboard and outboard housings. We now support the Wavetrack differential, Miata wheel hub, as well as Hyper racing's hub-less wheel center. This allows the wheel center and bearings to mount directly to the tripod housing, therefore effectively removing the need for a wheel hub.

A couple quick technical notes for you. **First, these axles are limited to about 12 degrees of angle.** The tripods will wear out very quickly at higher angles, and if they are plunged all the way into the tripod housing the bar can contact the housing. **Second, you will have to cut your own snap ring grooves and cut the axle to its final length**. Third, we do not supply the spindle nuts for the outboard joints, or the snap rings / bolts for your diff. (Look for recommendations in the FAQ sections!) You should source those yourself as a lot of teams use different parts.

So, here is what's included in a "kit":

2 x Inboard lightened tripod housings. Choose from the available options below.
2 x Outboard lightened tripod housings. Choose from the available options below.
4 x RCV Performance Triangular CV Boots.
4x RCV Billet Tripods
2 x Custom Length Axles, Input lengths for right and left side below.
10 x Snap Rings