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INTEGRATED IN-WHEEL ELECTRIC VEHICLE POWERTRAIN DESIGN

Joseph Herring

INTEGRATED IN-WHEEL ELECTRIC

VEHICLE POWERTRAIN DESIGN

by

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The University of Texas at Arlington in Partial Fulfillment

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HONORS BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

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May 2019

ACKNOWLEDGMENTS

I dedicate this work to my mother and father, for always pushing me to be the very best I can be. As a first generation college graduate, their sacrifices and selflessness have allowed me to pursue my dreams, and for that, I am forever grateful. To my beautiful wife, Katherine, who has always been by my side through this journey, you are my why. To my teammates and best of friends, Jacob Lamotte-Dawaghreh, Logan Pechal, Matthew Smith, Tim Pugliese, Garrett Tolar, I have thoroughly enjoyed every moment of working on this project with you all. Without the hard work and dedication of each and every one of you, this project would not be possible. I also deeply thank Dr. Yawen Wang, our faculty advisor and mentor for this senior project, for his guidance and support over the last year. My team and I extend great appreciation for the generous help of Don Carkin, Jake Floyd, Alex Rhodes, and Darren Jackson of Lockheed Martin Missiles and Fire Control.

May 03, 2019

ABSTRACT

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VEHICLE POWERTRAIN DESIGN

Joseph Herring, B.S. Mechanical Engineering

The University of Texas at Arlington, 2019

Faculty Mentor: Yawen Wang

Over 50% of the carbon monoxide, and 25% of the harmful hydrocarbons released into the air in 2013 were attributed to transportation. It is clear that the automobile industry is entering a new paradigm as resources are increasingly spent towards developing electric vehicles as a sustainable transportation source. NextGen Drive is a team of senior Mechanical Engineering students working with Dr. Yawen Wang in the Mechanical Engineering Department at The University of Texas at Arlington to research and develop the design of an integrated in-wheel electric motor powertrain solution for an electric vehicle that offers increased energy efficiency, delivers superior vehicle performance, and improved safety, over traditional automobiles. Some of the primary deliverables for this project include a technical data package of complete engineering drawings, as well as topology optimization for strength-to-weight ratio, and the 3D printing of a prototype.

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CHAPTER 1

INTRODUCTION

1.1 Thesis Scope

The scope of this thesis is to communicate the primary contributions and individual deliverables constituted in an Engineering Senior Design Project over the course of the 2018-2019 academic year. This thesis is comprised of a synopsis of the design project at large, followed by a detailing of the authors unique body of work within it.

1.2 Senior Project Introduction

The objective of this project was to deliver a technical data package of a conceptual design of an in-wheel drive system for an electric vehicle. This system includes a commercial off the shelf (COTS) electric motor, regenerative braking system, hub bearings, rim and tire, mounting brackets and a generalized suspension mount with steering interface.

1.2.1 Environmental Stewardship

Over 50% of the carbon monoxide, and 25% of the harmful hydrocarbons released into the air in 2013 were attributed to transportation. It is clear that the automobile industry is entering a new paradigm as resources are increasingly spent towards developing electric vehicles as a sustainable transportation source.

1.2.2 Superior Vehicle Characteristics

In-wheel drive electric powertrain systems provide better performance, increased efficiency, lower emissions, and more useable cabin space over traditional centrally powered internal combustion and hybrid electric vehicles. The primary factor yielding the benefits that an in-wheel drivetrain offers is torque vectoring. Torque vectoring is the ability to distribute power to each wheel of the vehicle independently. This characteristic allows for tighter and more precise vehicle handling, safer vehicle response to dangerous road conditions, and more energy efficient power transfer.

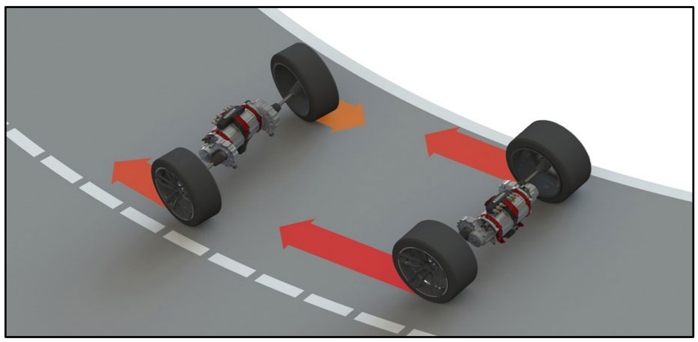


Figure 1.1: Torque Vectoring [1]

1.2.3 Eliminating Mechanical Inefficiencies

An additional advantage of an in-wheel drivetrain is the elimination of many mechanical assemblies used in the traditional vehicle drivetrain. Components such as the transmission, differential, and driveshaft are not necessary with an electric in-wheel drive. Along with the financial benefit behind this, there are associated inefficiencies with mechanical components that will be effectively eliminated.

1.3 Contribution Overview

As Team Captain of the NextGen Drive design team, the author held responsibility for the successful completion of this design effort. This included generating and tracking project scheduling, writing mid-semester reports, organizing team work structure and assignments, and meeting coordination and correspondence with the class faculty, faculty mentor, and project monitor.

CHAPTER 2

LITERATURE REVIEW

2.1 State of the Market

There are currently no commercial in-wheel drive electric vehicles readily available to consumers. A research analysis of the academic literature on in-wheel electric vehicle powertrains yields that most of the research being conducted on this technology is by NTN, who are publishing test data on experimental in-wheel motor systems. These NTN Technical Reviews were submitted to the senior design team by the faculty advisor as the foundation of field research to consider. A review of the professional market yields a few startup companies, such as ProteanDrive and Elaphe. Overall, in-wheel electric drive is an up and coming technology that is yet to be implemented into the commercial market.

2.2 Design Research

A comprehensive review of current state of the art electric vehicles determined that the average electric commuter vehicle has a curb weight of 3,500 lb, 180 HP, and costs $35,000. The comprehensive research of electric vehicles lead to the conclusion that the average weight for vehicle operational analysis will have close to 400 HP, and is anticipated to have a cost of $40,000.

2.2.1 Commuter Electric Vehicle Market Review

The market for electric cars was surveyed in order to determine an average of curb weight, power, and cost, as seen in Table 2.1. The two main vehicle weights reported by manufacturers are Gross Vehicle Weight Rating (GVWR) and curb weight. GVWR includes the base curb weight of the vehicle plus the weight of any optional accessories, cargo and passengers. Curb weight is the weight of the vehicle itself with all fluids, but no passengers, cargo or accessories. The weights below are Curb Weight. The average vehicle weight was used to make assumptions about the design's vehicle during the vehicle operational requirement analysis. Additionally, power and cost specifications would allow the design's competitiveness in the market place.

Table 2.1: Commuter Electric Vehicle Market Review

Vehicle	Power (HP)	Curb Weight (lb)	Base Cost ($)
Nissan Leaf	147	3,471	29,990
Chevy Bolt	200	3,563	36,620
Chevy Volt	149	3,794	33,520
Toyota Prius	121	3,045	27,300
Tesla Model 3	271	3,955	46,000

CHAPTER 3

PRELIMINARY DESIGN PHASE

NextGen Drive has developed a preliminary conceptual design of the mechanical components of an electric drive system that fits inside the wheel hub of a commuter electric vehicle. The design process began with developing a house of quality to define clear customer requirements and derive the corresponding engineering characteristics our system requires in order to best meet our client's needs. The next step was to define system performance requirements. This large engineering task required a substantial amount of research regarding current electric vehicle standard performance and analysis. This vehicle performance analysis served as the foundation and created milestones for later design stages of component trade studies and selection. In summary, this initially led to the selection of a smaller electric motor with a 4:1 planetary gear reduction. However, the primary design goal of maintaining geometric design space of the inner wheel hub ultimately lead to the decision of eliminating the need for a gear reduction step altogether, and instead opting for a larger electric motor that can exceed all vehicle operational requirements while directly driving the wheel.

3.1 House of Quality Analysis

A House of Quality analysis determined that the Gear Reduction System (if applicable) is the most heavily weighted engineering characteristic. Although the design initially included a planetary gear system with a gear reduction of 4, ultimately, due to space constraints within the wheel, the larger EMRAX 348 motor was selected to power the vehicle without any gearing or gear reduction. Disk brakes were selected over drum brakes for better performance, and the ability to incorporate regenerative braking. A custom motor mount manufactured from sheet metal is attached to a larger cast bracket that mounts the wheel hub bearing as well as providing an interface for suspension and steering.

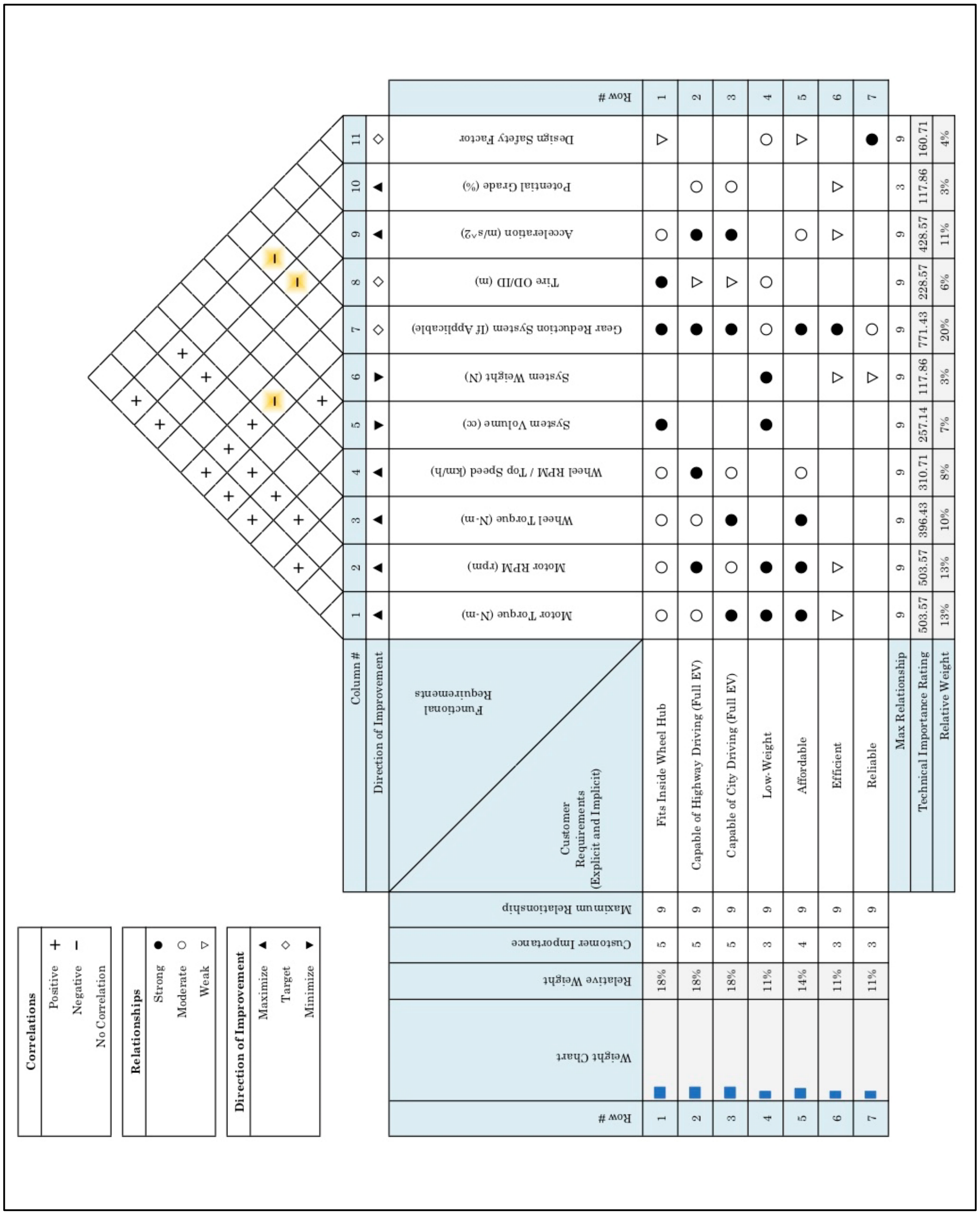


Figure 3.1: House of Quality Project Requirements Analysis

3.2 Design Performance Determination

To determine our performance requirements, we first had to make a few assumptions concerning the geometry of our car, such as the frontal area, drag coefficient, and weight, as well as the environment in which it would drive, such as the atmospheric pressure and temperature. A complete list of these assumptions is given in Table 3.1.

Table 3.1. Vehicle Dynamics Analysis Assumptions

Parameter Assumption	Value
Drag Coefficient ($\boldsymbol{C_D}$)	0.3 [2]
Loaded Vehicle Weight (W)	4500 lb
Frontal Area ($\boldsymbol{A_f}$)	24 ft^2
Tire Outer Diameter	30 in
Minimum Environmental Temperature	-30 °C
Maximum Environmental Pressure	102 kPa
Grade	0 to 33%

Based on these assumptions, three resistive forces including the drag force, the force due to rolling resistance, and the weight of the car were considered in the analysis (see equations 1-3) [2]. These were calculated under eight different driving conditions, shown in Table 3.2.

$$F_D = \frac{1}{2}\rho_{air} v_{air}^2 C_D A_f \tag{1}$$

$$F_{RR} = W \cdot f_R \cdot \cos(\theta) \tag{2}$$

$$F_W = \sin(\theta) \tag{3}$$

Table 3.2: Driving Conditions for Vehicle Dynamics Analysis

Scenario	Constant Velocity	Accelerating
1	100 mph, 0° grade	4 m/s^2 at 5 mph
2	85 mph, 5° grade	3 m/s^2 at 30 mph
3	60 mph, 8° grade	1.75 m/s^2 at 50 mph
4	30 mph, 18.4° grade	1 m/s^2 at 60 mph

Next, torque at the wheel, rpm, and power were calculated for each of these scenarios. The results are shown in Table 3.3 and 3.4 below.

Table 3.3: Vehicle Torque Requirements

Scenario	Constant Velocity	Accelerating
1	380 ft-lb	3234 ft-lb
2	741 ft-lb	2807 ft-lb
3	973 ft-lb	2476 ft-lb
4	1884 ft-lb	2333 ft-lb

Table 3.4: Vehicle Power Requirements

Scenario	Constant Velocity	Accelerating
1	81.1 hp	25.4 hp
2	118.5 hp	132.5 hp
3	124.5 hp	194.8 hp
4	120.6 hp	220.2 hp

Thus, the peak required torque is approximately 3240 ft-lb, and the peak power is approximately 220 horsepower. Note that these values were calculated for the entire vehicle, and each wheel only requires a quarter of the stated value.

CHAPTER 4

CONCEPTUAL DESIGN PHASE

4.1 Initial Design Concept

4.1.1 Motor Selection

Based on the performance requirements that were calculated and established, the most reasonable choice of motor was initially found to be the EMRAX 228, which carried more power in a smaller geometric confinement than any other electric motor in the market. This motor was part of a larger line of motors designed to be integrable into in-wheel drive systems and other hub motor applications. Analysis of the torque capabilities of the EMRAX 228 showed the necessary gear reduction would be 4:1 to achieve the required performance.

4.1.2 Gearbox Selection

Planetary and cycloidal gearing was considered in the possible gearbox design trade study and analysis. The two gearing types are shown in Figure 4.1, with a planetary gear on the left, and cycloidal gear on the right.

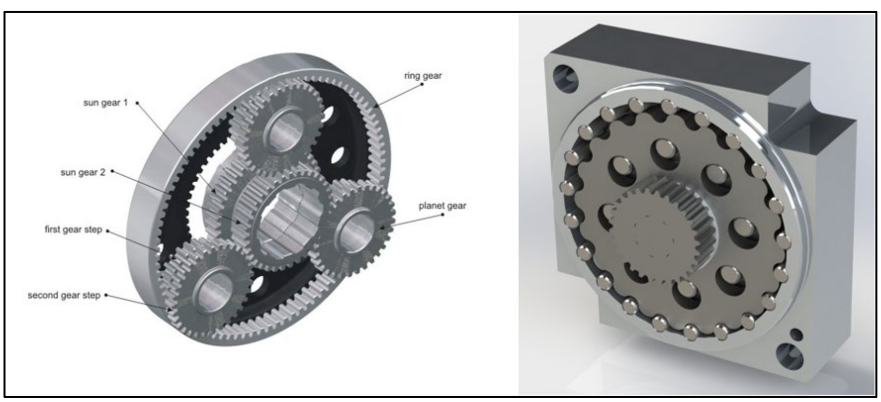


Figure 4.1: Planetary and Cycloidal Gears [3, 4]

Planetary gears are known for their smooth operation, high efficiency, and ability to handle high torque densities. Additionally, planetary gearboxes generally experience backlash between one to five arc-minutes and are ideal for achieving lower gear ratios. On the other hand, cycloidal gears are ideal for achieving slightly higher gear ratios, sometimes as high as or higher than 200:1. These gearboxes also offer highly compact, dense structures that generate zero to one arc-min backlash. This level of precision, however, generally causes cycloidal gearboxes to be more expensive than their planetary counterparts. [5]

Taking the pros and cons of each gearbox type into consideration, the decision was made to move forward with a planetary gearbox in the initial design phase because it was capable of handling high torque densities. After our motor data analysis was complete, and a motor chosen, a gear reduction of 4:1 was needed. It so happens that one of the planetary weaknesses played to our team's gain, since we needed a very low gear reduction.

Our team contacted several companies to see what planetary gearboxes they had to offer, and after conducting this research we decided to choose Apex Dynamics as our supplier. Apex Dynamics manufactures and assembles several high-quality gearboxes that fit our team's needs. We chose to use the AD140-004 planetary gearbox with a gear reduction of four to one.

One of the only COTS gearboxes that met the torque requirements was the Apex Dynamics AD140-004 gearbox. When inserted into the wheel, it caused the motor and structure behind it to protrude out of the wheel several inches. Besides falling outside of the space constraints of the system, it also created a problem for mounting the suspension.

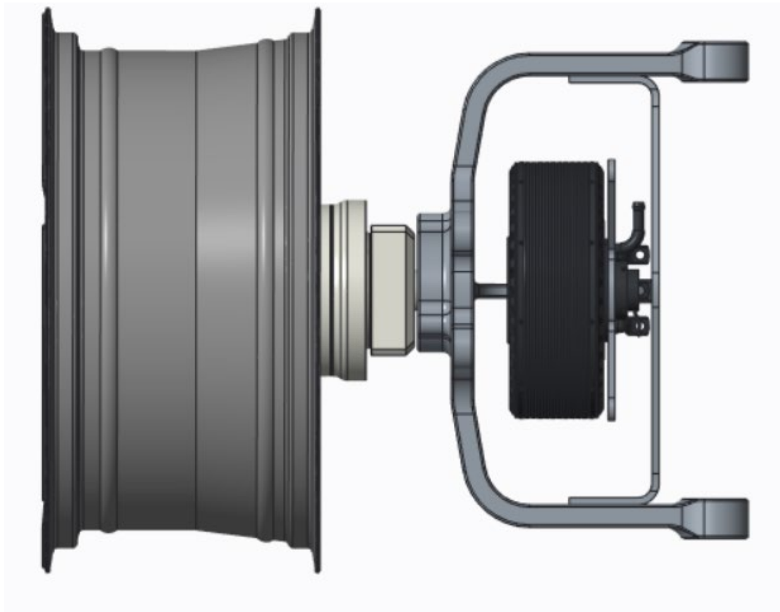


Figure 4.2: Preliminary Design with Gearing Reduction

4.2 Conceptual Redesign

4.2.1 Motor Selection

At this point, the decision was made to use a larger motor in the same line, the EMRAX 348. Although this model cost about $7,000 more than the EMRAX 228 motor (before quantity discounts), its torque capabilities allowed for a direct drive system. Additionally, it was also expected that since the expected vehicle production rate would greatly increase EMRAX's production rate that a reasonable expectation is to get a minimum of 70% discount at quantity [6]. After comparing the size and cost of the EMRAX 228 and Apex Gearbox pair with those of the direct drive EMRAX 348 motor, the decision was made that a direct drive system drivetrain will allow for a more proper in-wheel design fit. This also allows for room to be gained for the mounting of a suspension system design.

4.2.2 Structural Components

Once the motor was selected, the team designed the structural components to accommodate it. One of the driving features of the design was the need to connect the rotating parts to the stationary parts without directing the weight of the car through the motor. Most cars currently on the road use a hub and bearing assembly that fits inside the brake rotor to serve this purpose. These are readily available off the shelf, leading the team to choose this method as well. The outer side of this piece relative to the main body of the car houses the lug nuts which bolt onto the brake rotor and wheel rim. This outer side also connects to the small drive axle from the motor. The inner side of the hub and bearing assembly bolts to the custom designed upright that serves as the suspension, steering, break, and motor mount interface. An exploded view of the entire design is presented below in Figure 4.3.

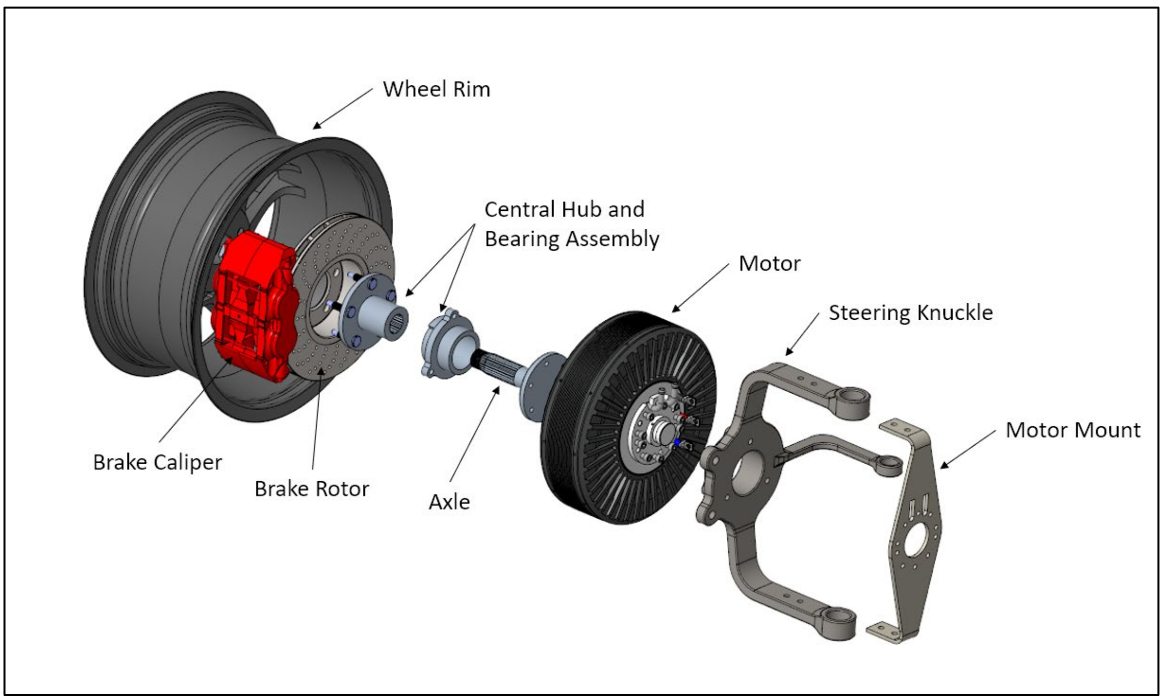


Figure 4.3: Conceptual Design Exploded Assembly View

CHAPTER 5

ANALYSIS AND REDESIGN

5.1 Design Performance Determination

By the recommendation of mechanical engineering faculty member Dr. Woods, the steering axis of the assembly was required to be brought closer to the midline of the wheel itself, otherwise, vehicle handling performance will greatly decrease. The team's client has also expressed that steering performance of the design is a priority. Thus, research was conducted on vehicle suspension kinematic parameters and average commuter vehicle parameters were determined with which we will assume our design will operate.

5.1.1 Steering Knuckle Articulation Redesign

To accomplish the goal of a midline steering axis, the steering knuckle was redesigned to accommodate the articulation of two structural support elbows that connect with thrust bearings that are to be bolted to the steering knuckle cylinder. This new assembly allows the elbows, which are connected to the vehicle suspension system, to pivot independently of the rest of the wheel and motor assembly, thus moving the steering axis directly in line with the wheel, and allowing for optimal steering. Structural arms were also implemented to interface with the motor mount bracket. These design changes may be seen in Figures 5.1 and 5.2 below.

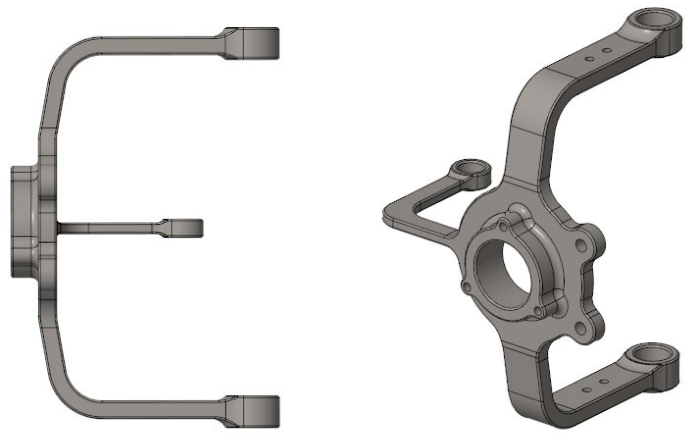


Figure 5.1: Original Steering Knuckle Assembly

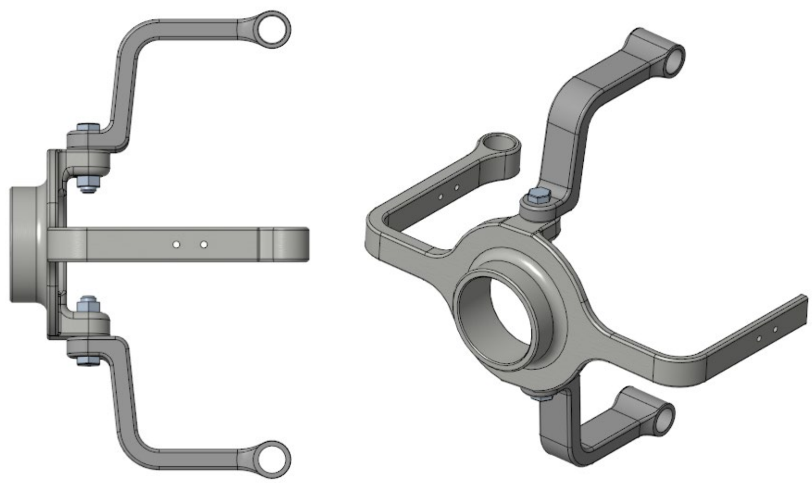


Figure 5.2: Redesigned Steering Knuckle Assembly

5.2 Topology Optimization Analysis

Topology optimization is a branch of Finite Element Analysis (FEA) that utilizes mathematical methods to optimize a particular design parameter for specified constraints and boundary conditions. The primary constraint that drives the optimization process is the specified design space that the final geometry must exist within. The boundary conditions may include other design parameters such as load sets and support structures.

Inspire 2017 is the software utilized to conduct topology optimization to minimize the weight of two structural components critical to the in-wheel design: The steering knuckle, and the motor mount bracket. Minimizing the weight of these components is critical to minimizing unsprung mass for optimum vehicle performance.

In order to conduct a topology optimization with respect to either maximizing stiffness, or as in this case, minimizing mass, the first step is to define the design space that the optimized component geometry may inhabit. The design space was specified for the motor mount bracket. The support and loading locations were also specified, and a static structural safety factor minimum of 2.0. As can be seen in Figure 5.3. below, the topology optimized geometry is generated such that it satisfies all the input requirements.

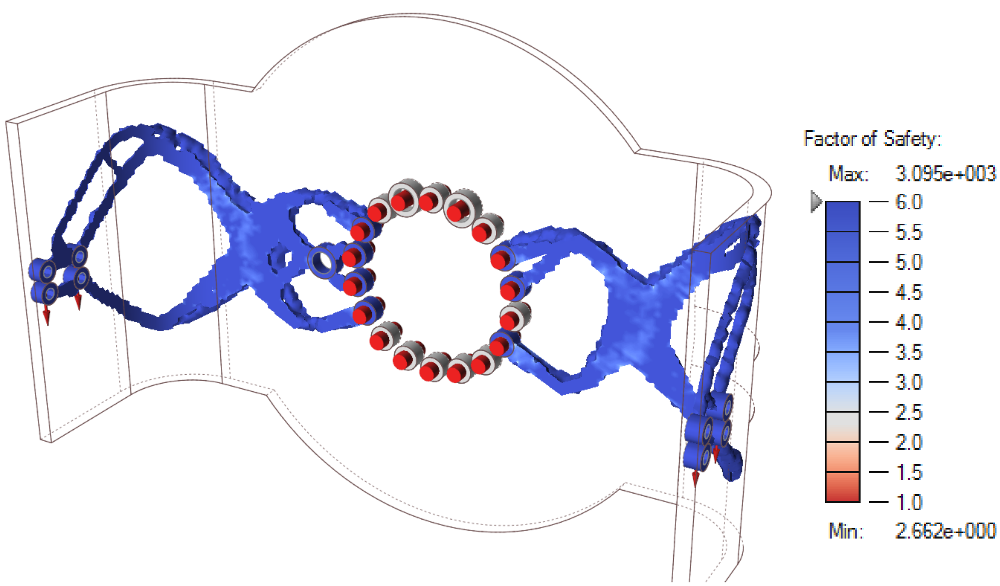


Figure 5.3: Motor Mount Bracket Topology Optimization Design Space

5.2.1 Polynurb Design

Once the optimized design geometry is generated through the topology analysis process, the resulting part is relatively rough on a surface level. The designer must again utilize the Inspire software to generate Polynurbs, which are curved surface geometries that allow for a solid part to be created from the generated topology optimized results. The optimized bracket with incorporated Polynurbs can be seen in Figure 5.5 below.

5.2.2 Motor Mount Comparison

Utilizing topology optimization, the weight of the motor mount bracket was able to be reduced by 40%, while maintaining a comparable static structural safety factor to that of the originally designed component.

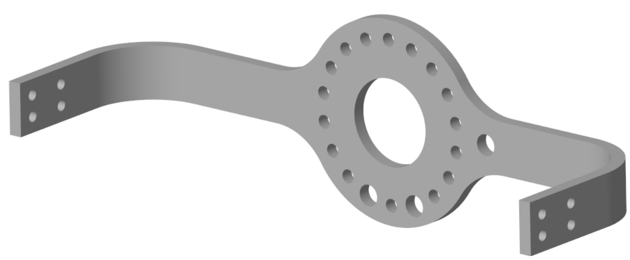


Figure 5.4: Original Motor Mount Bracket Design

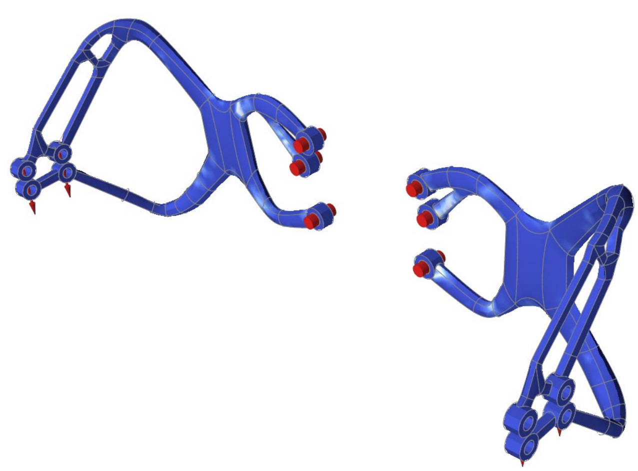


Figure 5.5: Motor Mount Bracket Topology Optimized Design

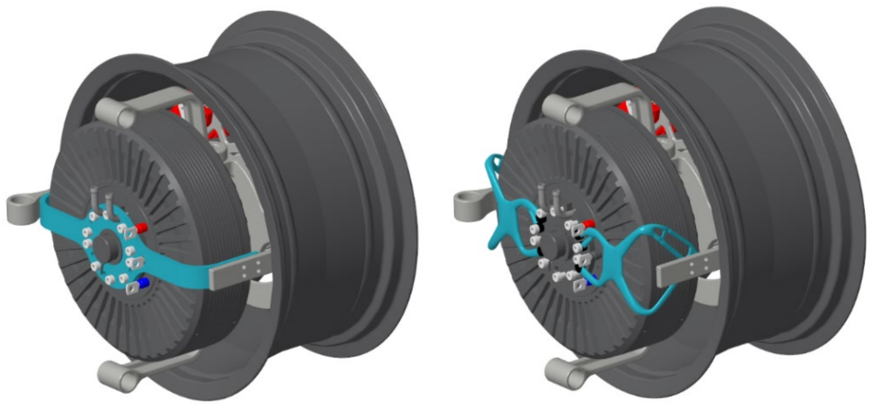


Figure 5.6: Original and Optimized Motor Mount Bracket Comparison

5.2.3 Steering Knuckle Optimization

In the same way, the steering knuckle was optimized to minimize the unsprung mass of the vehicle. In this case, the steering knuckle carries the load of the vehicle and is subject to much greater loads than the motor mount bracket. The weight of the steering knuckle was reduced by 30%, while maintaining a comparable static safety factor.

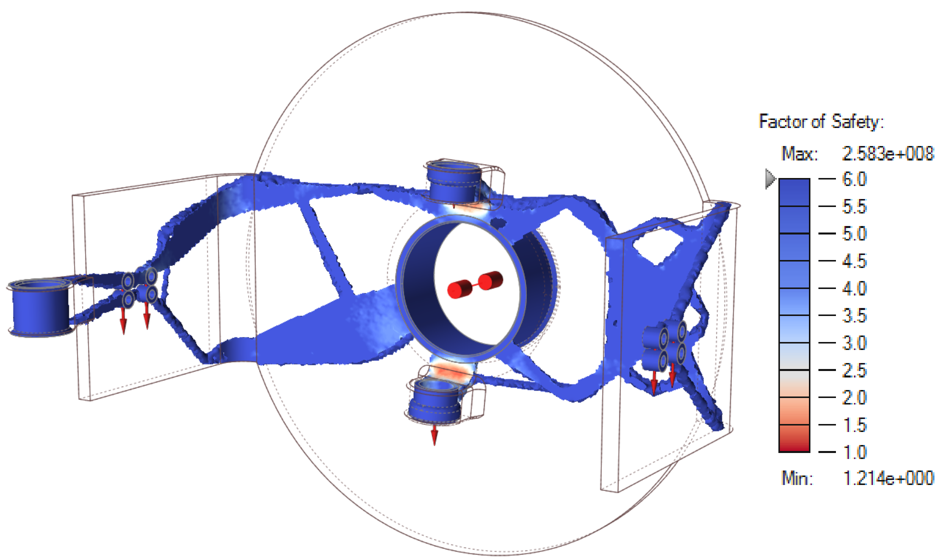


Figure 5.7: Steering Knuckle Topology Optimization Design Space

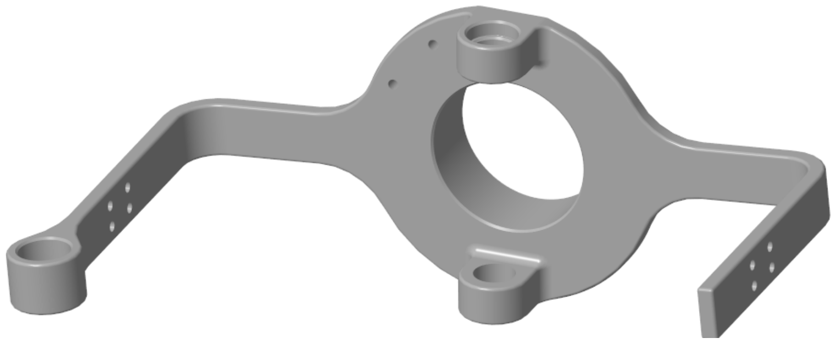


Figure 5.8: Original Steering Knuckle Design

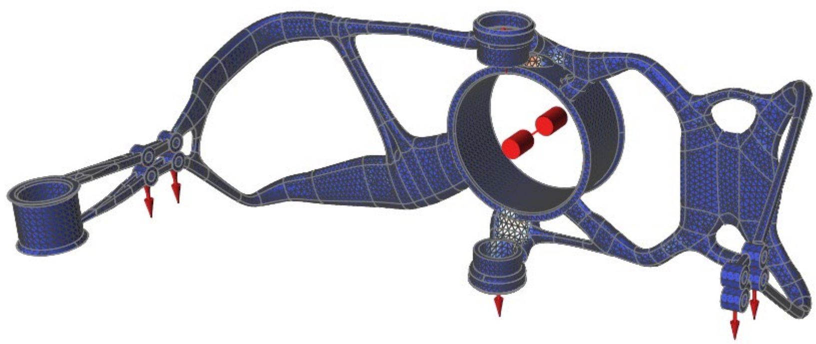


Figure 5.9: Final Topology Optimized Motor Mount Bracket

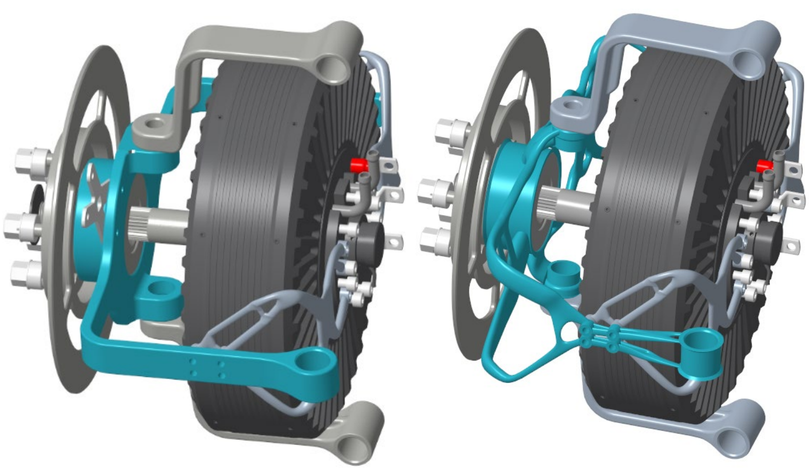


Figure 5.10: Original and Optimized Steering Knuckle Comparison

CHAPTER 6

CONCLUSIONS

The primary objective of the senior design project was to deliver a technical data package constituted of 3D CAD Model of a conceptual design for an integrated in-wheel motor powertrain for electric vehicles. Utilizing the engineering design process, a conceptual design was 3D modeled, analyzed with a series of structural and dynamic analyses, and a geometric representative prototype was produced for a faculty advisor client. Within the context of the statement of work put forth by the client, a commercial off the shelf motor was a necessary assumption for the design team to make. Overall, the body of work that the senior design project represents displays the fact that the successful implementation of an in-wheel drive system will likely require the customization that a uniquely custom-designed motor may offer. In the future, additional aspects of the design project that groups may wish to pursue includes creating a custom motor with an integrated hub bearing to support the vehicle weight, a custom gearbox with a thin depth to area ratio compared to what is typically commercially available, and a tolerance stack-up analysis to determine more accurate tolerances for the engineering drawings of designed components.

APPENDIX A

TECHNICAL DATA PACKAGE 3D CAD MODELS

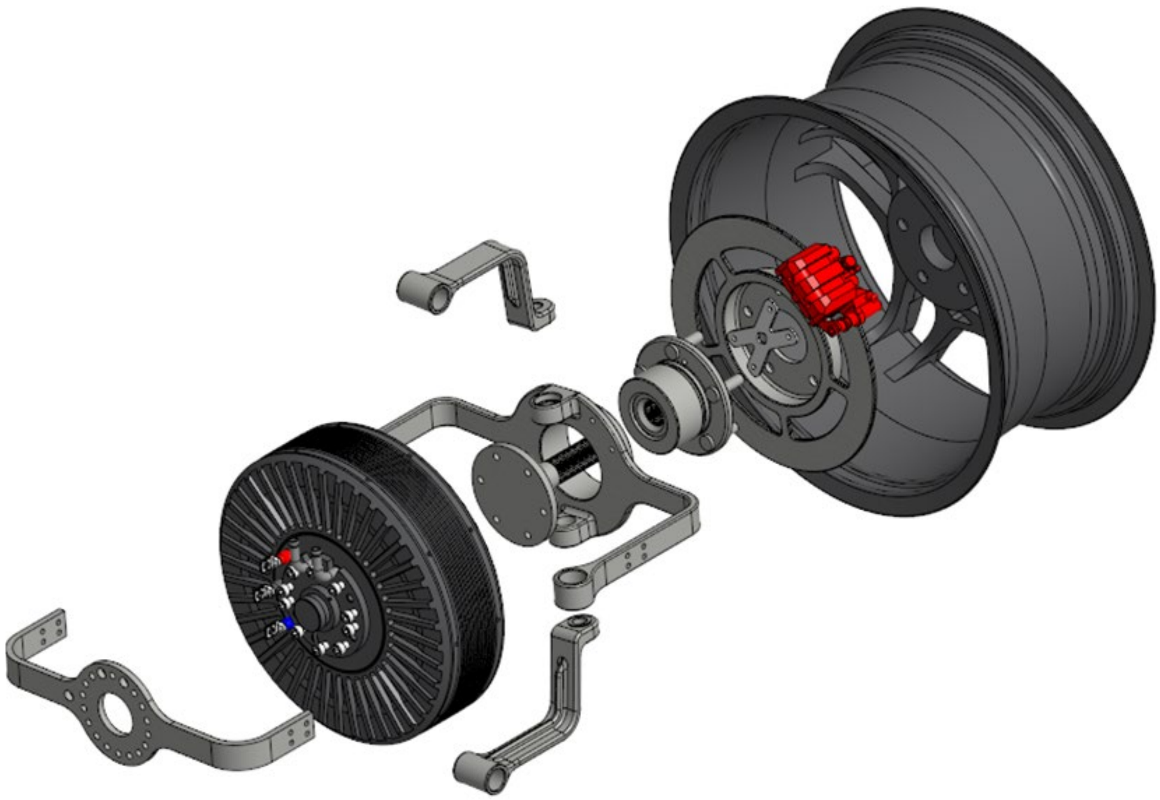


Figure A.1: Exploded Assembly View

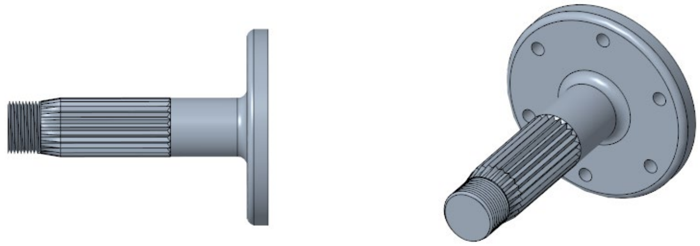


Figure A.2: Splined Torque Shaft

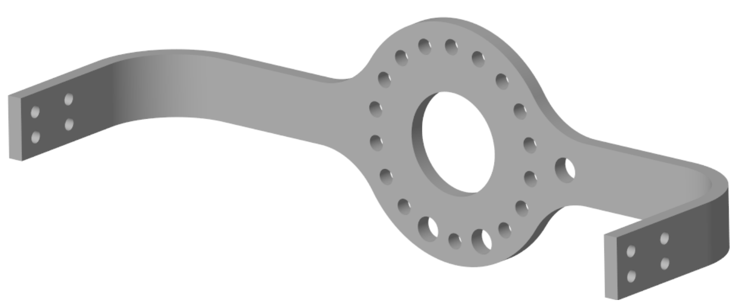


Figure A.3: Motor Mount Bracket

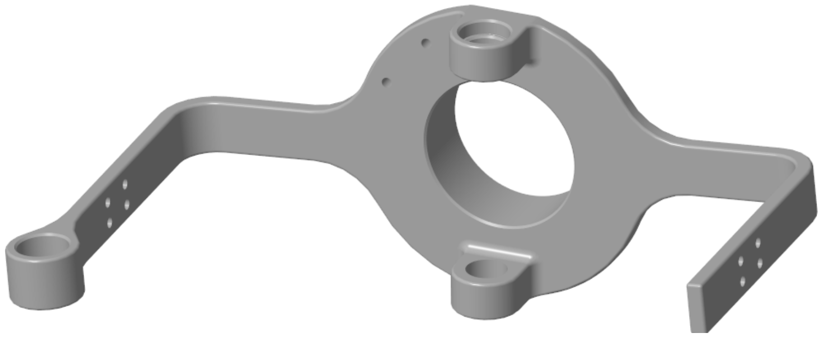


Figure A.4: Steering Knuckle

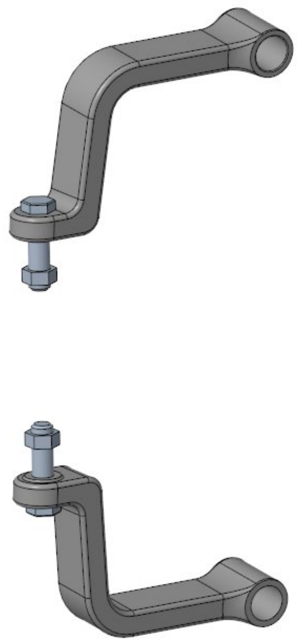
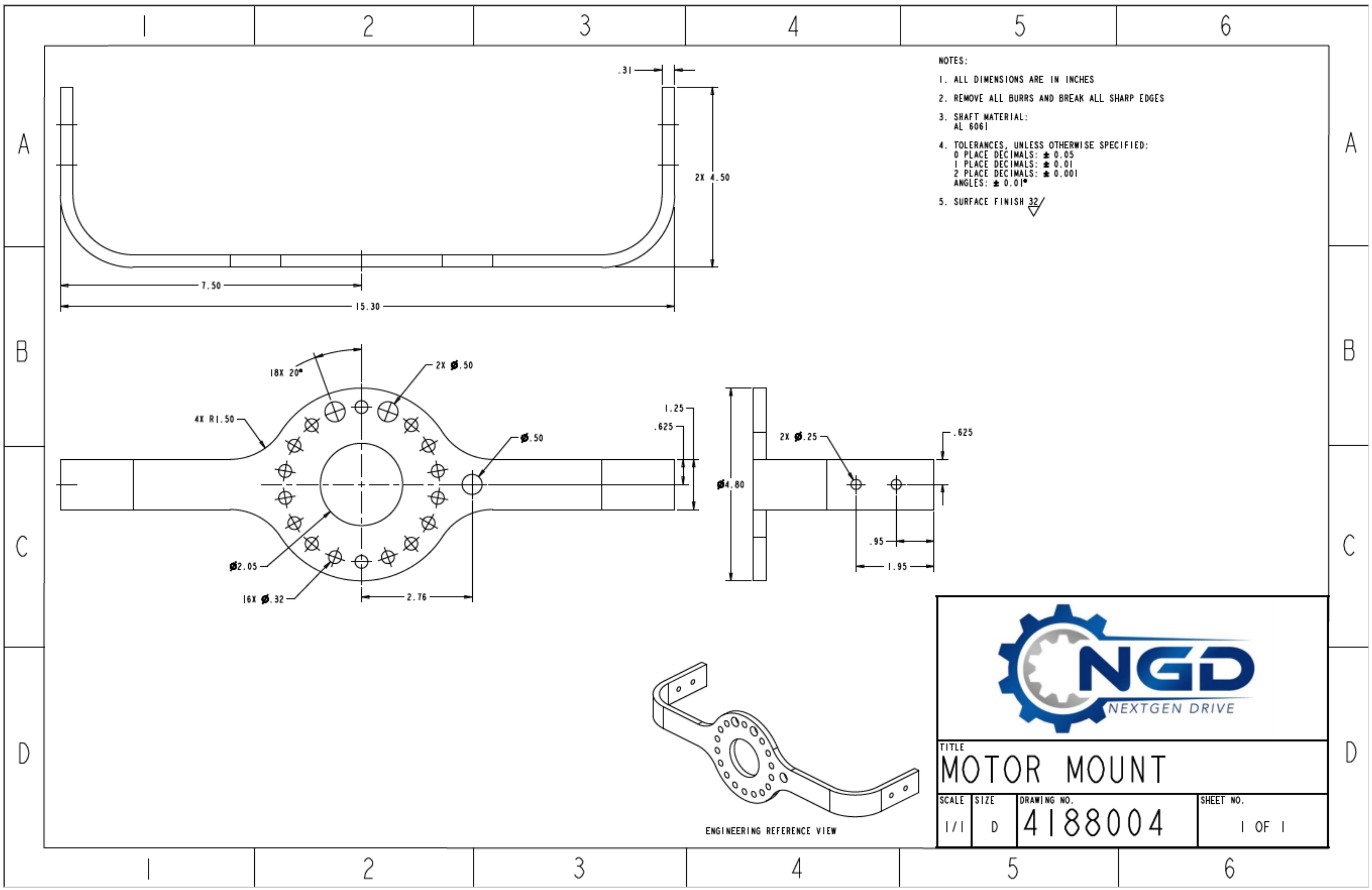
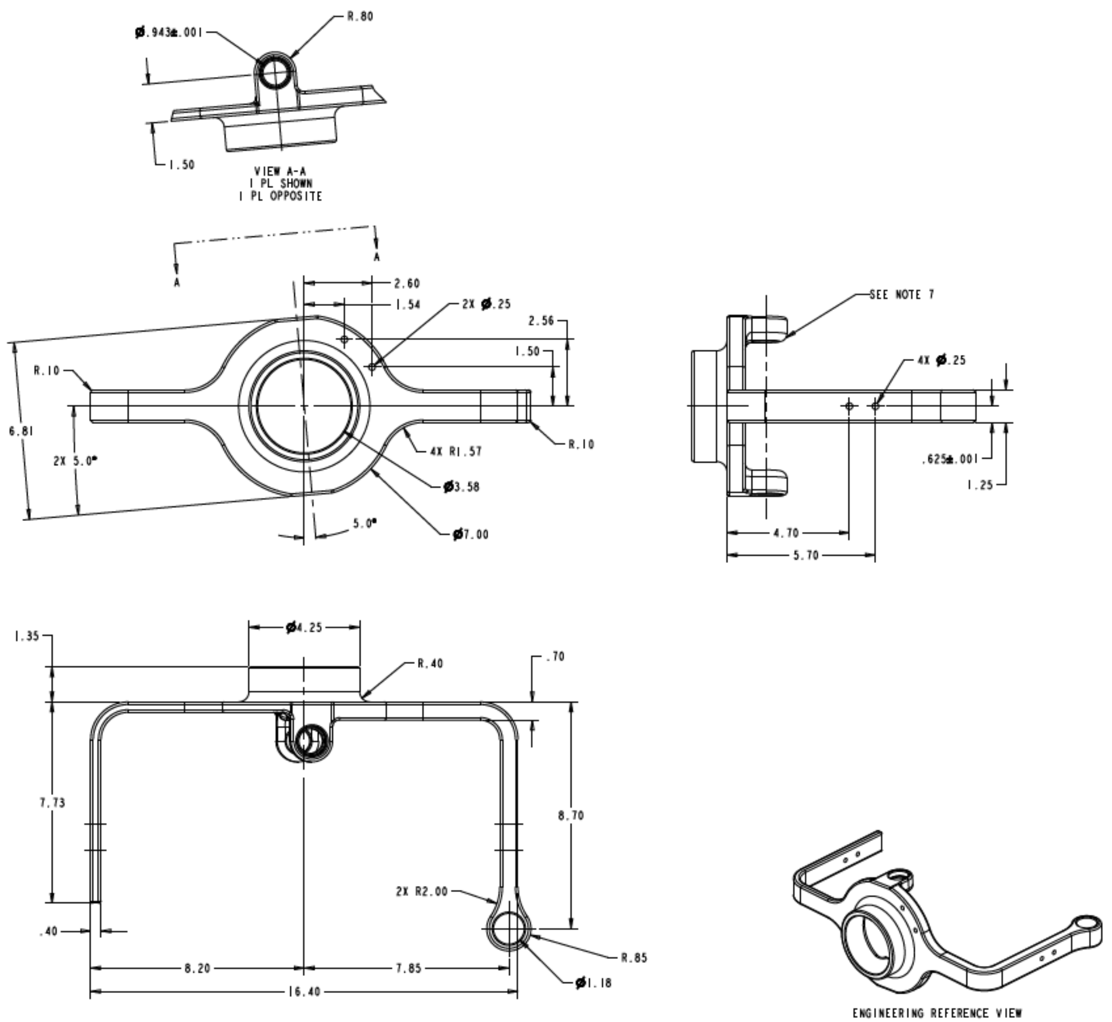


Figure A.5: Structural Support Elbows

APPENDIX B

TECHNICAL DATA PACKAGE ENGINEERING DRAWINGS


NOTES:
1. ALL DIMENSIONS ARE IN INCHES
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. SHAFT MATERIAL:
AL 6061
4. TOLERANCES, UNLESS OTHERWISE SPECIFIED:
0 PLACE DECIMALS: ± 0.05
1 PLACE DECIMALS: ± 0.01
2 PLACE DECIMALS: ± 0.001
ANGLES: ± 0.01°
5. SURFACE FINISH 32
.31
2X 4.50
7.50
15.30
18X 20°
2X ⌀.50
4X R1.50
⌀.50
⌀2.05
16X ⌀.32
2.76
1.25
.625
⌀4.80
2X ⌀.25
.625
.95
1.95
ENGINEERING REFERENCE VIEW
NEXTGEN DRIVE
TITLE
MOTOR MOUNT
SCALE
1/1
SIZE
D
DRAWING NO.
4188004
SHEET NO.
1 OF 1

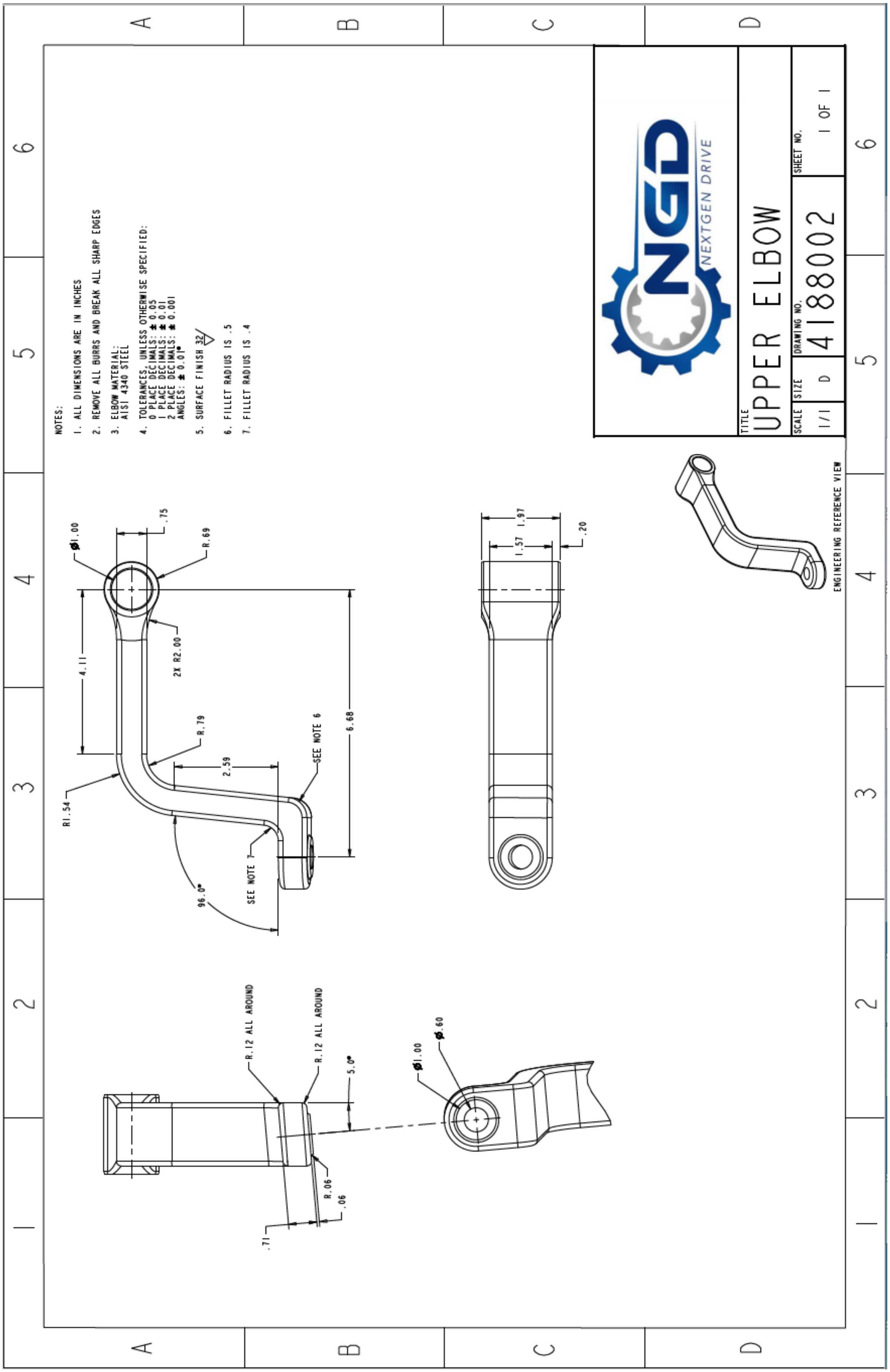


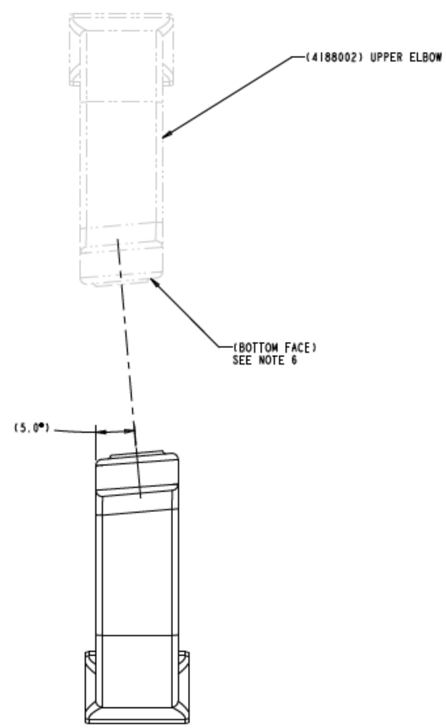
NOTES:

1. ALL DIMENSIONS ARE IN INCHES
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. SHAFT MATERIAL:
   AL 6061
4. TOLERANCES, UNLESS OTHERWISE SPECIFIED:
   0 PLACE DECIMALS: ± 0.05
   1 PLACE DECIMALS: ± 0.01
   2 PLACE DECIMALS: ± 0.001
   ANGLES: ± 0.01°
5. SURFACE FINISH 32
6. ALL FILLET RADII ARE .08, UNLESS OTHERWISE SPECIFIED
7. FILLET RADIUS .28 UNDERNEATH MOUNTING TABS

NGD NEXTGEN DRIVE

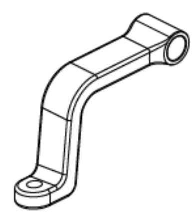
TITLE			
STEERING KNUCKLE			
SCALE	SIZE	DRAWING NO.	SHEET NO.
1/1	D	4188003	1 OF 1


NOTES:
1. ALL DIMENSIONS ARE IN INCHES
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. ELBOW MATERIAL:
AISI 4340 STEEL
4. TOLERANCES, UNLESS OTHERWISE SPECIFIED:
0 PLACE DECIMALS: ± 0.05
1 PLACE DECIMALS: ± 0.01
2 PLACE DECIMALS: ± 0.001
ANGLES: ± 0.01°
5. SURFACE FINISH 32
6. FILLET RADIUS IS .5
7. FILLET RADIUS IS .4
ø1.00
.75
R.69
4.11
2X R2.00
6.68
R.79
2.59
SEE NOTE 6
R1.54
SEE NOTE 7
96.0°
1.57
1.97
.20
R.12 ALL AROUND
R.12 ALL AROUND
5.0°
ø1.00
ø.60
R.06
.06
.71
ENGINEERING REFERENCE VIEW
NEXTGEN DRIVE
TITLE
UPPER ELBOW
SCALE
1/1
SIZE
D
DRAWING NO.
4188002
SHEET NO.
1 OF 1



NOTES:

1. ALL DIMENSIONS ARE IN INCHES
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. ELBOW MATERIAL:
   AISI 4340 STEEL
4. TOLERANCES, UNLESS OTHERWISE SPECIFIED:
   0 PLACE DECIMALS: ± 0.05
   1 PLACE DECIMALS: ± 0.01
   2 PLACE DECIMALS: ± 0.001
   ANGLES: ± 0.01°
5. SURFACE FINISH 32
6. LOWER ELBOW IS MIRROR OF 4188002, UPPER ELBOW. LOWER ELBOW IS MIRRORED ABOUT BOTTOM FACE OF UPPER ELBOW, AS SHOWN.

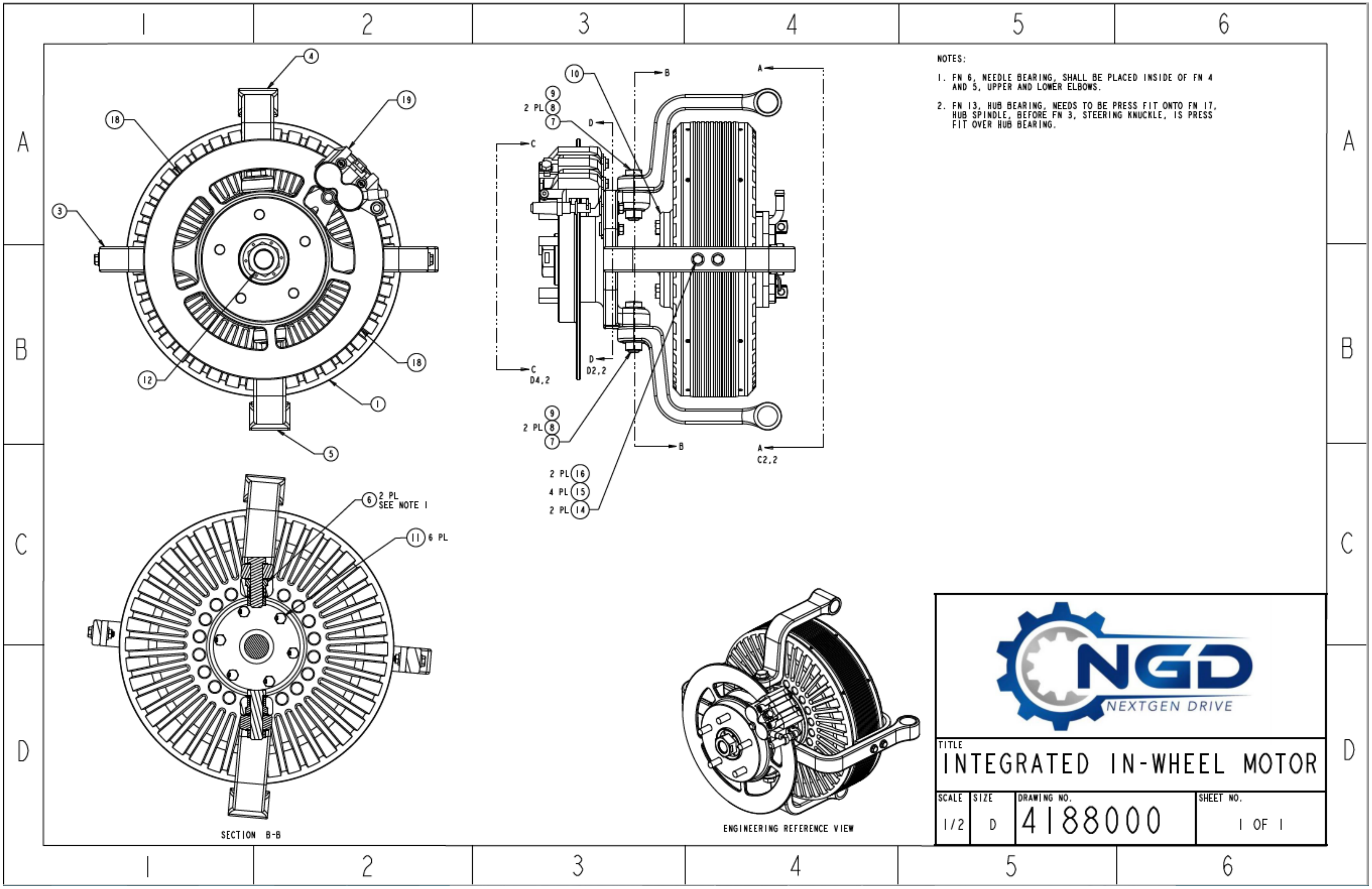


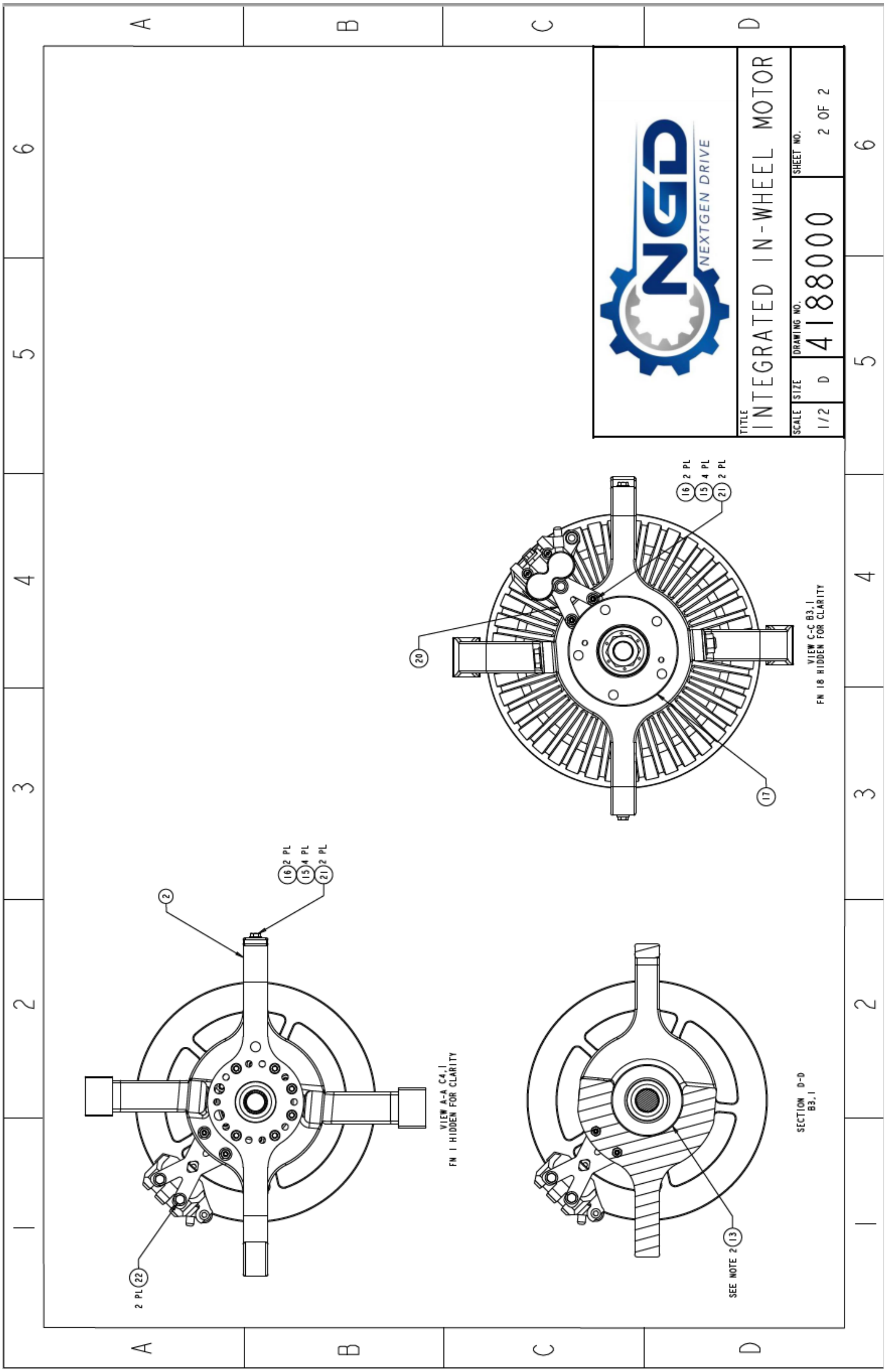
ENGINEERING REFERENCE VIEW

TITLE			
LOWER ELBOW			
SCALE 1/1	SIZE D	DRAWING NO. 4188001	SHEET NO. 1 OF 1

ASSEMBLY BILL OF MATERIALS

PART NAME	FN	QTY
EMRAX 348 MOTOR	1	1
MOTOR MOUNT BRKT	2	1
STEERING KNUCKLE	3	1
UPPER ELBOW SUPPORT	4	1
LOWER ELBOW SUPPORT	5	1
THRUST BEARING	6	2
M15X2X58 BOLT	7	2
M15 WASHER	8	2
M15X2 NUT	9	2
SPLINED SHAFT	10	1
M8X1.25X25 BOLT	11	6
1-14 FLANGE NUT	12	1
WHEEL HUB BEARING	13	1
1/4-20X1.375 BOLT	14	2
1/4 WASHER	15	12
1/4-20 NUT	16	10
WHEEL HUB SPINDLE	17	1
REGEN DISC BRAKE ROTOR	18	1
BRAKE CALIPER	19	1
BRAKE CALIPER BRKT	20	1
1/4-20X1.25 BOLT	21	2
1/4-20X.25 BOLT	22	2


NOTES:
1. FN 6, NEEDLE BEARING, SHALL BE PLACED INSIDE OF FN 4 AND 5, UPPER AND LOWER ELBOWS.
2. FN 13, HUB BEARING, NEEDS TO BE PRESS FIT ONTO FN 17, HUB SPINDLE, BEFORE FN 3, STEERING KNUCKLE, IS PRESS FIT OVER HUB BEARING.
SECTION B-B
ENGINEERING REFERENCE VIEW
NGD
NEXTGEN DRIVE
TITLE
INTEGRATED IN-WHEEL MOTOR
SCALE 1/2
SIZE D
DRAWING NO. 4188000
SHEET NO. 1 OF 1
2 PL
SEE NOTE 1
6 PL
4 PL
C2,2
D2,2
D4,2



TITLE
INTEGRATED IN-WHEEL MOTOR
SCALE
1/2
SIZE
D
DRAWING NO.
4188000
SHEET NO.
2 OF 2
NEXTGEN DRIVE
VIEW A-A C4,1
FN 1 HIDDEN FOR CLARITY
SECTION D-D
B3,1
VIEW C-C B3,1
FN 18 HIDDEN FOR CLARITY
SEE NOTE 2
2 PL
4 PL

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BIOGRAPHICAL INFORMATION

Joseph Herring graduated from the University of Texas at Arlington with an Honors Bachelor of Science in Mechanical Engineering. He plans to continue pursuing his education with a Masters in Mechanical Engineering from the University of Texas at Arlington. His technical interest lies in Engineering Design and Additive Manufacturing. Joseph currently works as a Mechanical Design Engineering Associate at Lockheed Martin Missiles and Fire Control.