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ASME Baja RC Car Drivetrain

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ASME Baja Car contest

Drivetrain

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Contents

Abstract

The ASME Baja car is a student built radio controlled car, which will compete with other schools in a variety races. The drivetrain aspect of the car must smoothly transfer power from the motor to the rear wheels of the car. The drivetrain must be designed in collaboration with the other team member's suspension system to avoid interference. The majority of the parts in the drivetrain are purchased parts intended for various applications. These parts were chosen based on their proximity to the optimal calculated values. Power flows from the electric motor through a single gear reduction, then through a single timing belt drive reduction, next through a differential, then through universal joint axles and finally to the rear wheels. The gear reduction was chosen for gears ability to provide low reduction ratios and high torque capacity. The timing belt drive was chosen for the ability to absorb impact due to their elasticity this helps to maintain smooth power delivery. The differential is used to allow the drive wheels to rotate at different speeds through corners, this further increases smoothness of power delivery as well as improving driver control. Universal joint axles were chosen for their ability to transfer power from a stationary point (differential) to an oscillating point (wheel hub). This drivetrain design allows smooth and controllable power delivery, however, the weight of the rotating components take enough power to reduce the top speed of the car out of the target range.

Introduction

Motivation

Cars of any kind have been an integral part of society for over one hundred years now. The automotive industry provides a rich environment for optimization and creativity in mechanical design. While not all principals of automotive design can be scaled down, there are still many similarities between R/C cars and their larger counterparts. Applying knowledge of car design and principals of mechanical engineering to the R/C Baja car will be a challenging and rewarding project.

Function Statement

A system to deliver power from a motor to the rear wheels is needed on the R/C Baja car.

Requirements

The goal of this design is to produce a strong, efficient and simple drivetrain to propel the R/C Baja car at high speeds. This means that the drivetrain will provide torque to the rear wheels of the car even under the most extreme conditions. Quantifiable requirements include:

- Drivetrain must allow the car to reach a speed of no less than 30mph
- Motor temperature must stay under 160 degrees F when held at half throttle for as long as a fully charged 5200mah LiPo battery lasts.
- Total cost must be under $200 not including any electronic components.

- Must comply with all ASME and ROAR vehicle requirements.
- Drivetrain must be able to resist 24 in-lbs of static torque with the wheels held fixed for 24hours
- Drivetrain must not exceed a height of 2.5in from the top of the chassis.
- Differential must withstand the heat and stress of half throttle operation while holding alternating drive wheels still for 5sec at a time, for 1min.

Engineering Merit

Since this project requires a certain top speed and torque holding, the design of the drivetrain system will need to optimize components for efficient power transmission at specific torque and rpm values. Other calculations include rpm and torque of various components, as well as stresses in these components.

Scope of effort

In this build two people are dividing the design and building duties into suspension and drivetrain. This build is also using many previously designed components. Therefore, this project will be able to be ready for competition in march.

Success Criteria

A successful design will need to finish the race first. This means that the drivetrain must continue to deliver power to the wheels throughout the duration of the race. This car will also be competing against others and will need to place in the top 5 for the sprint, slalom, and baja events.

Success scenario:

The car passes all testing and or places in the top 5 at the competition.

Design and Analyses

Approach: Proposed Solution

The power from the motor will be transmitted from an electric motor to the rear wheels via constant mesh spur gear train, and a synchronous belt drive reduction for an overall reduction of 12 to 1. From the reduction, an open differential will be designed to allow use of pre-existing side and spider gears. This system will function as a single assembly that will be held together by a 3d printed support structure.

Design Description (picture, sketch, rendering)

Starting from the motor, the steel pinion gear will be attached to the motor output shaft via set screw as is standard in R/C design. This pinion will then mesh with an idler gear which will be in mesh with a plastic composite spur gear that is flange mounted to a 1/4in diameter steel shaft. A timing belt pulley will be fitted to the shaft via set screw between two roller bearings. The differential will be mounted allowing a center to center distance of 1.528in to the timing belt pulley. A 15mm wide 3mm GT pitch neoprene fiberglass timing belt will transfer power from the timing belt pulley to the differential housing which has 3mm GT pitch grooves around its perimeter. The differential will be mounted with roller bearings and a bearing cap, that fastens it to the support structure. On the outside of the roller bearings the axle shafts will be attached to universal jointed two-piece drive shafts that allow axial articulation so that they will not interfere with suspension travel.

Benchmark

The benchmarks used will be two store bought high performance R/C cars. The drivetrain components will be tested for the same requirements as the drivetrain being designed. The first car is an HPI RS4 this car will be tested for top speed using the same motor that the Baja car will use. This will isolate the gear ratio and efficiency of the drivetrain. The other car is of unknown origin and will be tested for static torque holding.

Performance Predictions

Calculations made in figure 13 show that the vehicle will reach a top speed of 37.46 MPH. the drivetrain's weakest link is the timing belt pulley which is calculated to hold up to 42in-lbs of torque before tooth jump occurs and is rated for 22.67in-lbs of torque an average of these values gives a predicted 32in-lbs of static torque holding capability.

Description of Analyses

Since the motor being used was already decided, the logical starting point for analysis of a drivetrain is the find the power and torque produced by the motor. figure 9 shows these calculations. The chronology of the next few calculations is not as straight forward. Parameters such as gear pitch, belt pitch, final drive ratio (FDR), and gear and pulley teeth numbers must all be decided simultaneously in order to ensure size, top speed, and static torque holding are all kept within range. Figure 10 shows the decisions for gear/pulley arrangement as well as the number of teeth/grooves on each, except the idler gear. Figure 11 determines the idler gear size needed to maintain clearance between the timing belt pulley and the motor. Figure 12 calculates a design torque for the belt drive system based on previously decided values. Next figure 17 proves that the previously decided values for the belt design will meet the required torque holding. Even though the rated torque value was determined to be just under the required static torque holding value, the tooth jump will not occur until almost double the required static torque. Therefore, it can be assumed that the static torque holding will be well above the requirement. Figure 13 proves that the previous values for FDR will be sufficient to provide the required top speed of 30mph. Figure 14 proves that the drive shaft angle in the horizontal plane of the vehicle will stay within the allowable limit for u joint drive shafts. Figures 15 and 16 show clearance calculations for the bevels gears inside the diff and the spurs gears respectively.

Scope of Testing and Evaluation

In order to test the requirements of static torque holding a slender rod of uniform mass will be attached to the pinion gear of the motor in order to test exact torque holding. The top speed can be tested by installing a GPS on the vehicle. Additionally, the drivetrain must not allow the motor to overheat, a laser thermometer will be used to measure the motor temp at half throttle for the entire length of one battery charge. The assembled drivetrain must also maintain full functionality after being dropped onto concrete from a height of 2 feet. The motor will not be installed during this test. Finally, the differential must maintain full functionality after holding alternating drive wheels still for 5sec at a time for 1min, at half throttle operation speed.

Analyses

Design Issues

1. Making sure that the differential housing can be manufactured and assembled, while still staying small.
2. Attaching the spur gear and timing belt pulley to the shaft while still being able to hold static torque.
3. reducing number of components in the drivetrain while still maintaining top speed and torque holding capabilities.
4. determining how much static torque a belt drive system will hold.

Calculated Parameters

Using the specifications of the motor and the voltage of the battery it was calculated that the maximum speed of the motor is 38160rpm. A tire diameter of 4.1in and an assumed 0.3in of tire ballooning with a gear ratio of 12:1 the max speed was calculated to be 37.46mph. tire ballooning is a term that describes the radial deformation of a soft rubber tire at high speed. This number must be assumed because an accurate number would require the total vehicle weight which is not known at this time.

Static torque holding was the most difficult parameter to determine. First the rated torque was determined from the calculation for the number of grooves in the timing belt pulley in figure 10, and the shaft speed calculated in figure 17. Table 41 in figure 7 was then used to determine that the rated torque for the belt drive system is 22.67in-lbs shown in figure 17. Next the torque required for tooth jump to occur was determined to be 42in-lbs from figure 1. Since both of these torque values assume that the belt drive system is in motion they do not accurately represent the static torque holding, however, it can be seen in Table 41 of figure 7 that as rpm decreases the rated torque increases. All the way to about 60in-lbs at 10rpm. This makes the tooth jump torque the lower value. So it can be determined that the belt drive system will hold up to 42in-lbs of static torque.

Best Practices

The weakest point in the drivetrain in the belt drive system, therefore it was designed with a safety factor of greater than 2 for static torque holding and a safety factor of 3.74 for rated torque.

Device: Parts, Shapes and Conformation

The support material for the drivetrain will be 3D printed and its form will be largely artistic to give the drivetrain some aesthetic value. The material will have a minimum thickness of 0.125in. The design will also need to keep the overall height within the required 2.5in from the top of the chassis. The most innovative part of this design is that the support structure will not bolt to the chassis, instead it will be located in the horizontal plane by the aluminum suspension structures and then clamped to the chassis using a sheet metal clamp that bolts itself to the chassis. This will allow rapid removal of the entire drivetrain assembly for any needed repair work.

Device Assembly, Attachments

The drivetrain assembly must mount to an aluminum chassis and must not interfere with suspension components and their travel.

Tolerances, Kinematics, Ergonomics, etc.

Tolerances in the drivetrain must be tight in order to ensure parallelism between gear faces. This will ensure that the entire face width of the gears is utilized and that no axial thrust is generated. Parallelism tolerance will be plus or minus .005in. all tolerances will ultimately be produced by the hole locations of the 3d printed parts. In order to keep these tolerances tight, all hole will be printed smaller than needed, they will then be bored to final size using a drill press. This will ensure the tightest tolerances possible.

Technical Risk Analysis, Failure Mode Analyses, Safety Factors, Operation limits

The biggest risk in the project is the differential manufacturing, as one small error could ruin the whole part and the process would then need to be repeated. 3d printing will be greatly beneficial in this regard since any errors can be rapidly corrected and the part can be reprinted.

The weakest point in the drivetrain in the belt drive system, therefore it was designed with a safety factor of greater than 2 for static torque holding and a safety factor of 3.74 for rated torque. These safety factors will ensure that the device will not fail under normal operation, however, the nature of a race car is not normal operation, therefore the drop testing and differential testing will simulate extreme operation limits.

Methods and Construction

Construction

Description

The drivetrain of the R/C baja car will use both bulk manufactured parts and parts made here at CWU. All parts made at CWU will be 3D printed out of ABS plastic. Bulk items include shafts gears pulleys and belts.

Drawing Tree, Drawing ID's

Figure 3 in appendix B shows the first half of the differential case that will be 3d printed.

Parts list and labels

Aftermarket parts

1. (2x) roller bearing with inner diameter of 0.25" and face width of under 0.25"
2. (2x) skateboard bearings with inner diameter of 8mm and face width of 7mm
3. (1x) steel shaft .25" diameter 3.0" long
4. (1x) 32 groove 3mm GT timing belt pulley for 15mm belt width
5. (1x) 40 groove 3mm GT timing belt pulley for 15mm belt width
6. (4x) fasteners for holding two halves of diff. together.
7. (1x) 63 groove 3mm GT timing belt at 15mm wide
8. (4x) 4.1" diameter R/C buggy tires
9. (4x) fasteners for bearing cap for diff.
10. (2x) fastener to hold powertrain clamp down
11. (1x) 22 tooth 48 pitch spur gear

Parts made in house

12. (1x) Diff carrier 1
13. (1x) Diff carrier 2
14. (1x) Diff housing
15. (2x) Diff side housing
16. (1x) Motor support structure
17. (1x) Diff support structure
18. (1x) Diff bearing cap
19. (1x) Drivetrain clamp

Manufacturing issues

3D printed parts have the advantage of the ability to fit to any design shape, however their strength is the weak point. Deciding a minimum thickness value of 0.125in for all 3d printed parts (except the differential internals) is important to maintain strength in the design. Another issue is achieving the proper fit of the bevel gear internal of the differential. Clearance was calculated in figure 15 however it is still exceedingly difficult to determine the exact dimensions for the diff internal supports. This was ultimately solved by modifying the design to allow bronze bushings to set the clearance. The length of these bushings was shortened until the gear clearance was within spec.

Discussion of assembly, sub-assemblies, parts, drawings (examples)

Two major assemblies make up the drivetrain the first is the differential which is then held by the support structure and the diff bearing cap.

Testing Method

Introduction

Testing of the R/C Baja cars drivetrain will be done by first testing the engine temp using a laser thermometer to ensure that the motor will not overheat during any other tests. Next the top speed will be tested on a flat smooth surface using a gps installed in the vehicle. Next static torque holding will be tested by attaching a slender rod the pinion gear. The next test will be the diff testing where alternating rear wheels will be held for 5sec at a time for 1min at half throttle operation speed. The final test will be the drop test where the drivetrain assembly will be dropped onto concrete in any orientation from a height of 2 feet.

Method/Approach

All of the testing done requires minimal setup and equipment except the torque test. Equipment needed for the testing includes; a laser thermometer and a gps.

The torque test will need to use an aluminum shaft whose length and diameter are determined in figure 18. This will allow the center of mass of the rod to act at a horizontal distance away from the center of the pinion gear giving a force times a distance of 24in-lbs. this test will then be left for 24 hours while periodically checking for any movement of the aluminum shaft.

Test Procedure

Motor temp test

1. Support vehicle so that the rear wheels are suspended.
2. Install fully charged battery and turn on remote and receiver.
3. Slowly trim throttle up to half power.
4. Use laser thermometer to measure temp of the motor case (make sure to sweep all exposed areas of the case to find any heat concentrations that may cause damaging temp spikes)

5. Stop test immediately if temp rises over 160F at any point on the motor case
6. Allow test to run until battery power begins to fade.

Top speed test

1. Install gps to the vehicle making sure that it is not near the motor or steering components.
2. Use suitable vacant parking lot to take 3 battery charges worth of top speed runs
3. Record the highest speed achieved for each battery (this will test which battery is the strongest)
4. Record the top speed overall as the test result.

Static torque holding

1. Attach vehicle to table with both rear wheels held from rotating.
2. Attach aluminum shaft to pinion gear of motor use level to make sure that the rod is perfectly horizontal.
3. Let sit for 24 hours
4. If rod is in the same angular position after 24 hours the test is passed.

Diff test

1. Support vehicle so that the rear wheels are suspended.
2. Install fully charged battery and turn on remote and receiver.
3. Slowly trim throttle up to half power.
4. While wearing gloves, steadily apply pressure to the side of one of the rear wheels so that it stops rotating within 2sec.
5. Hold wheel for 5sec after it has stopped rotating then release the wheel.
6. Repeat steps 4 and 5 while alternating between the two rear wheels for 60 secs.
7. Verify that the diff still allows full steering lock turns on pavement at minimum speed without wheel slip.

Deliverables

Top speed and Max torque holding numbers will be compared to predicted numbers. Redesigns will be made if any test is failed until a design that satisfies all 5 tests is achieved.

Budget/Schedule/Project Management

Proposed Budget

Discuss part suppliers, substantive costs and sequence or buying issues

All parts of the drivetrain will have at least 1 duplicate so that the car can be rebuilt after testing to failure. All purchased components will be bought from SDP-SI.com except for fasteners. Total cost from SDP-Si.com will be about $100

Determine labor or outsourcing rates & estimate costs

The 3D printer costs $6 per cubic inch of material used. No more than $50 will be spent on 3D printing.

Labor

Since there are no outsourced parts the total labor will only be the time taken for manufacturing of the drivetrain.

Estimate total project cost

Total cost should be $150

Funding source(s)

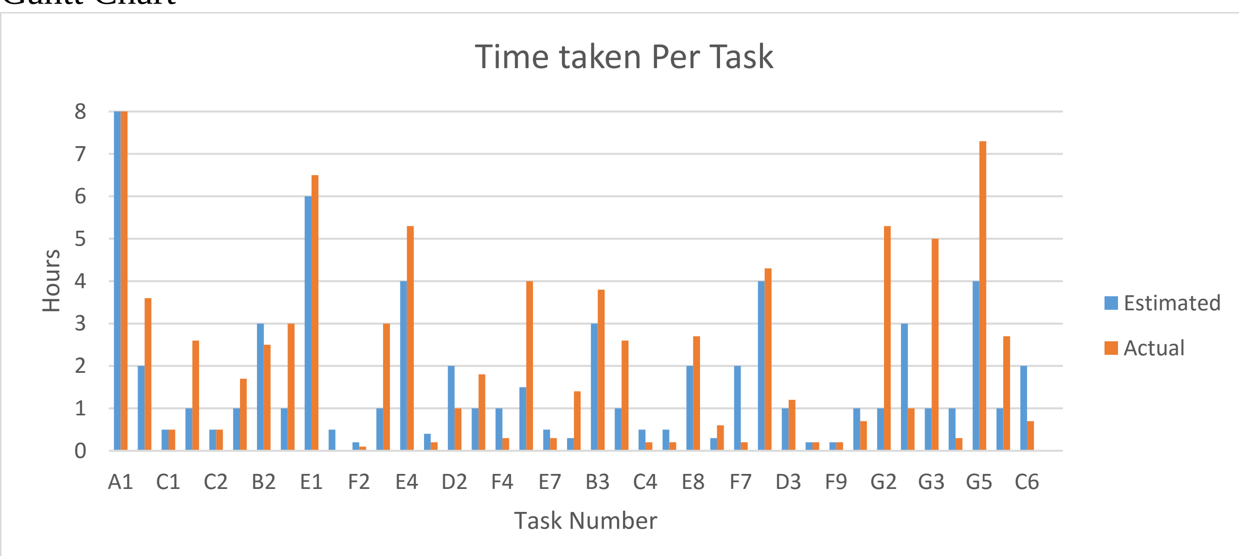
No funding has been sourced

Proposed schedule

Estimate total project time

If 10hr per week is spent on the project and the project takes 25 weeks total time should be 250hr.

Gantt Chart



A1	proposal
B1	verify calculations and 3D printed part dimensions
C1	diff carriers 1 and 2 printed
D1	parts ordered
C2	support structure 3d printed
F1	flaws in diff carriers measured
B2	modifications made to diff carriers
C3	reprint diff carriers
E1	hog out center of diff housing 1
E2	turn down OD of diff carrier 1 to prep for press fit
F2	press diff carrier 1 onto diff housing 1
E3	turn diff side gear bushings
E4	turn diff housing 2 (two parts nedded)
F3	mock assembly of diff (layout for bolt holes)
D2	purchase hardware needed fo diff, bearing cap, and gear hub
E5	drill bolt holes through diff carrier 1 and housing 2
F4	assemble diff and check function
E6	turn gear hub
E7	drill holes in gear hub
F5	mount spur gear to gear hub and shaft
B3	flaws in support structure analysed and redrawn
B4	diff bearing cap model updated to match support structure model
C4	support structure reprinted
C5	diff bearing cap 3d printed
E8	drill holes in diff bearing cap and support structure
F6	assemble drivetrain and check function
F7	fine tune drivtrain assembly
E9	construct drivetrain clamp
D3	pruchase hardware for drivetrain clamp
F8	install drivetrain on chassis
F9	verify fitment with suspension components
G1	aquire lazer thermometer and test motor temp
G2	preform top speed test
E10	build alluminum rod for static torque holding test
G3	preform static torque holding test
G4	preform diff test
G5	final redraw of 3D printed parts
B5	make duplicate of all 3D printed parts
C6	final rebuild of assembly
F10	drivetrain ready for competition

Project Management

Human Resources

- Roger Beardsley
- Charles Pringle
- Craig Johnson
- Aidan Pringle
- Tedman Bramble
- Matt Burvee

Physical Resources: Machines, Processes, etc.

- 3D printer in Houge hall
- Drill press in Hougue hall
- Sheet metal tools in Houge hall

Soft Resources: Software, Web support, etc.

- SDP-SI.com
- file.lasersaur.com/docs-thirdparty/The_World_of**_Timing_Belts**.pdf
- Machine elements in mechanical design by Robert l. Mott

Financial Resources: Sponsors, Grants, Donations

Since this is a low cost project if will be self-funded.

Discussion

Design Evolution / Performance Creep

From the original concept the design was a single gear reduction paired with a belt drive reduction. This design would have been simpler, however, it was not calculated to be able to withstand the amount of torque produced by the motor. Another issue was that the shaft speed was too high for the timing belt. After searching for a way to calculate belt torque capacity, a resource was found that proved that MXL pitch belts are far inferior to 3mm GT for high torque and speed applications. From equations and charts found in this new resource it became necessary to reduce the shaft speed and increase the size of the timing belt pulley. This was accomplished by reducing the pinion gear size down to 15 teeth and increasing the gear size to 96 teeth this dropped the shaft speed down to 2658 RPM and allowed for a much bigger timing belt pulley while still achieving a similar final drive reduction.

Project Risk analysis

The biggest cost on this project is the tires at $70, however, all other expenses are very low. This makes the overall financial risk very low. The whole project is also very portable, making build and repair work faster and cheaper. The only substantial risk involved in the project is it durability. At the end of the day this is a race car and will most definitely be subjected to extreme forces. Strong components and the ability for rapid replacement of broken components are the only defenses against these forces inherent to a racing environment.

Successful

To finish first, one must first finish. (Roger Beardsley)
The Scarab Took First place at the 2017 ASME conference.

Conclusion

Restate your design title and its complete design readiness.

The proposed drivetrain for the ASME Baja car has been optimized to meet and exceed all requirements determined by the nature of the challenge and those of the chosen motor. The system will prove to be strong, fast, compact, and cheap to produce. Most importantly however is its ability to function as a fully separate module that can be removed from the vehicle by simply loosening two screws and lifting it off the chassis.

Restate your important analyses and how this contributes to success.

Analysis was focused on top speed and static torque holding. It would have been simple to design the drivetrain to do one or the other, but combining the two has proved difficult. These two parameters must be designed simultaneously as well in order to make sure that the vehicle will function without fail.

Success was largely achieved due to the speed of the vehicle. Speed of the vehicle was dependent on the drivetrain holding together. Therefore this analysis was essential to the vehicle's success.

Restate your design predicted performance vs actual performance, with respect to your requirements. Use bullets if appropriate.

- Top speed 37.67mph
- Max static torque holding 23in-lbs

Acknowledgements: For gifts, advisors and other contributors

- Roger Beardsley
- Charles Pringle
- Craig Johnson
- Aidan Pringle
- Tedman Bramble
- Matt Burvee

References: You should reference your texts, web sites, technical papers and any other information supporting your proposal.

Reference 1:
http://file.lasersaur.com/docs-thirdparty/The_World_of_Timing_Belts.pdf

Reference 2:
http://www.sdp-si.com/

Reference 3:
http://www.york-ind.com/contact.htm

Reference 4:
https://traxxas.com/

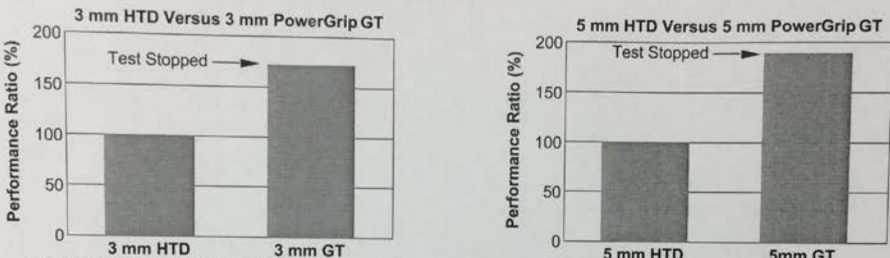
SECTION 4 DRIVE COMPARATIVE STUDIES

The development of the PowerGrip GT belt has produced an impressive range of enhanced properties and subsequent design opportunities for engineers.

Comparative studies, shown in **Figures 7** through **10**, allow designers to make quantitative assessments and to highlight the most significant improvements and design opportunities. Particularly significant points from the comparative studies follow:

4.1 Durability

The greatly increased durability of the PowerGrip GT design has resulted in power capacities far above those quoted for similar size belts of previous designs. The resulting small drive packages will increase design flexibility, space utilization and cost effectiveness.



TEST CONDITIONS: Speed = 6000 rpm
Power = 1.00 lb·in/mm width
Pulleys: Driver = 20 grooves
Driven = 20 grooves

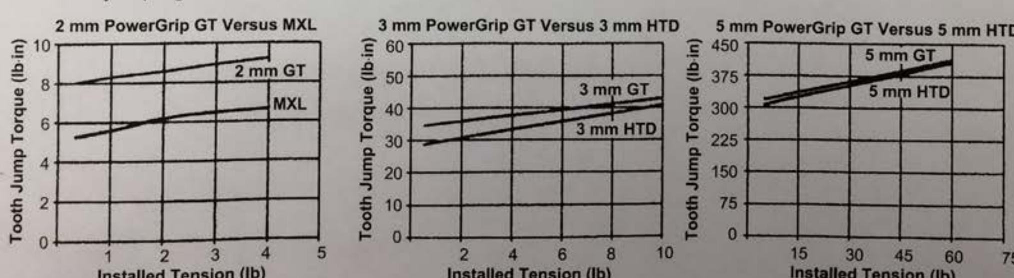
TEST CONDITIONS: Speed = 2300 rpm
Power = 3.35 lb·in/mm width
Pulleys: Driver = 20 grooves
Driven = 20 grooves

Fig. 7 Comparison of Performance Ratios for Various Belts

4.2 Tooth Jump Resistance

The very significant improvement in tooth jump resistance of PowerGrip GT when compared to similar belts has several important advantages.

1. Ratcheting resistance during high start-up torques.
2. Reduced bearing loads, particularly in fixed-center drives. Lower average tensions can be used without encountering tooth jump at the low tension end of the tolerance ranges.
3. Reduced system losses result from lower pre-tensioning, with less potential for tooth jumping.



TEST CONDITIONS:
Speed = 1130 rpm
Belt Width = 4.8 mm
Pulleys: Driver = 20 grooves
Driven = 20 grooves

TEST CONDITIONS:
Speed = 750 rpm
Belt Width = 6 mm
Pulleys: Driver = 30 grooves
Driven = 30 grooves

TEST CONDITIONS:
Speed = 2300 rpm
Belt Width = 15 mm
Pulleys: Driver = 20 grooves
Driven = 20 grooves

Fig. 8 Comparison of Tooth Jump Torques for Various Belts

Figure 1: Reference 1 T-8

The tension members are embedded in neoprene or polyurethane. The neoprene teeth are protected by a nylon fabric facing which makes them wear resistant.

The contributions of the construction members of these belts are as follows:

1. **Tensile Member** – Provides high strength, excellent flex life and high resistance to elongation.
2. **Neoprene Backing** – Strong neoprene bonded to the tensile member for protection against grime, oil and moisture. It also protects from frictional wear if idlers are used on the back of the belt.
3. **Neoprene Teeth** – Shear-resistant neoprene compound is molded integrally with the neoprene backing. They are precisely formed and accurately spaced to assure smooth meshing with the pulley grooves.
4. **Nylon Facing** – Tough nylon fabric with a low coefficient of friction covers the wearing surfaces of the belt. It protects the tooth surfaces and provides a durable wearing surface for long service.

6.1 Characteristics Of Reinforcing Fibers

Polyester

Tensile Strength	160,000 lbs/in^2
Elongation at break	14.0%
Modulus (approx.)	2,000,000 lbs/in^2

One of the main advantages of polyester cord over higher tensile cords is the lower modulus of polyester, enabling the belt to rotate smoothly over small diameter pulleys. Also, the elastic properties of the material enable it to absorb shock and dampen vibration.

In more and more equipment, stepping motors are being used. Polyester belts have proven far superior to fiberglass or Kevlar reinforced belts in these applications.

High-speed applications with small pulleys are best served by polyester belts under low load.

Kevlar

Tensile Strength	400,000 lbs/in^2
Elongation at break	2.5%
Modulus	18,000,000 lbs/in^2

High tensile strength and low elongation make this material very suitable for timing belt applications. Kevlar has excellent shock resistance and high load capacity.

Fiberglass

Tensile Strength	350,000 lbs/in^2
Elongation at break	2.5 – 3.5%
Modulus	10,000,000 lbs/in^2

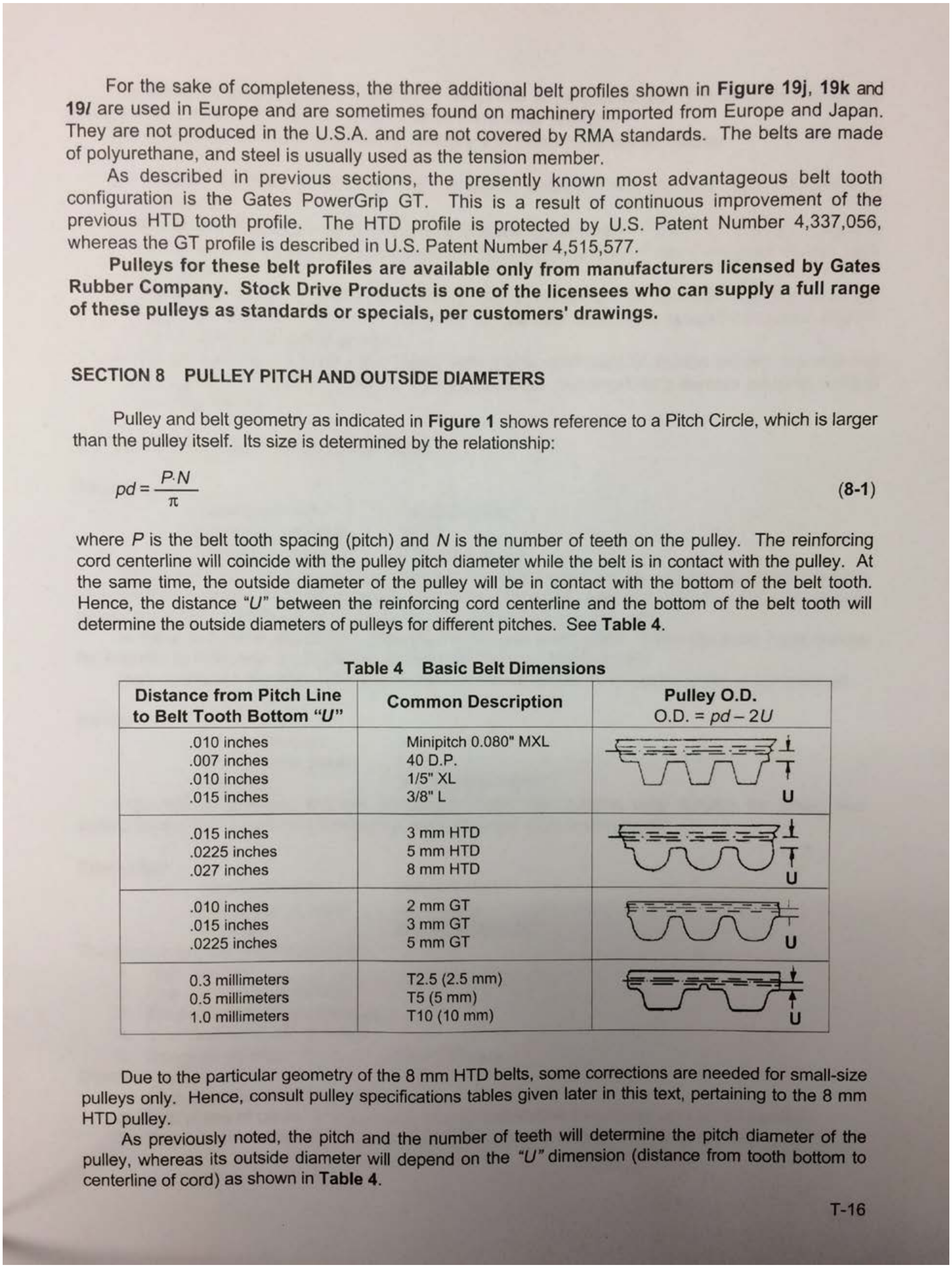
The most important advantages are:

1. High strength.
2. Low elongation or stretch.
3. Excellent dimensional stability.
4. Excellent chemical resistance.
5. Absence of creep, 100% elongation recovery.

Disadvantages:

1. High modulus (difficult to bend).
2. Brittleness of glass. Improper handling or installation can cause permanent damage.
3. Poor shock resistance. No shock absorbing quality when used in timing belts.

Figure 2: Reference 1 T-11



For the sake of completeness, the three additional belt profiles shown in **Figure 19j**, **19k** and **19*l*** are used in Europe and are sometimes found on machinery imported from Europe and Japan. They are not produced in the U.S.A. and are not covered by RMA standards. The belts are made of polyurethane, and steel is usually used as the tension member.

As described in previous sections, the presently known most advantageous belt tooth configuration is the Gates PowerGrip GT. This is a result of continuous improvement of the previous HTD tooth profile. The HTD profile is protected by U.S. Patent Number 4,337,056, whereas the GT profile is described in U.S. Patent Number 4,515,577.

Pulleys for these belt profiles are available only from manufacturers licensed by Gates Rubber Company. Stock Drive Products is one of the licensees who can supply a full range of these pulleys as standards or specials, per customers' drawings.

SECTION 8 PULLEY PITCH AND OUTSIDE DIAMETERS

Pulley and belt geometry as indicated in **Figure 1** shows reference to a Pitch Circle, which is larger than the pulley itself. Its size is determined by the relationship:

$$pd = \frac{P \cdot N}{\pi} \qquad \textbf{(8-1)}$$

where *P* is the belt tooth spacing (pitch) and *N* is the number of teeth on the pulley. The reinforcing cord centerline will coincide with the pulley pitch diameter while the belt is in contact with the pulley. At the same time, the outside diameter of the pulley will be in contact with the bottom of the belt tooth. Hence, the distance "*U*" between the reinforcing cord centerline and the bottom of the belt tooth will determine the outside diameters of pulleys for different pitches. See **Table 4**.

Table 4 Basic Belt Dimensions

Distance from Pitch Line to Belt Tooth Bottom "*U*"	Common Description	Pulley O.D. O.D. = $pd - 2U$
.010 inches .007 inches .010 inches .015 inches	Minipitch 0.080" MXL 40 D.P. 1/5" XL 3/8" L	U
.015 inches .0225 inches .027 inches	3 mm HTD 5 mm HTD 8 mm HTD	U
.010 inches .015 inches .0225 inches	2 mm GT 3 mm GT 5 mm GT	U
0.3 millimeters 0.5 millimeters 1.0 millimeters	T2.5 (2.5 mm) T5 (5 mm) T10 (10 mm)	U

Due to the particular geometry of the 8 mm HTD belts, some corrections are needed for small-size pulleys only. Hence, consult pulley specifications tables given later in this text, pertaining to the 8 mm HTD pulley.

As previously noted, the pitch and the number of teeth will determine the pitch diameter of the pulley, whereas its outside diameter will depend on the "*U*" dimension (distance from tooth bottom to centerline of cord) as shown in **Table 4**.

T-16

Figure 3: Reference 1 T-16

- Design with large pulleys with more teeth in mesh.
- Keep belts tight, and control tension closely.
- Design frame/shafting to be rigid under load.
- Use high quality machined pulleys to minimize radial runout and lateral wobble.

SECTION 10 BELT TENSIONING

10.1 What Is Proper Installation Tension

One of the benefits of small synchronous belt drives is lower belt pre-tensioning in comparison to comparable V-belt drives, but proper installation tension is still important in achieving the best possible drive performance. In general terms, belt pre-tensioning is needed for proper belt/pulley meshing to prevent belt ratcheting under peak loading, to compensate for initial belt tension decay, and to prestress the drive framework. The amount of installation tension that is actually needed is influenced by the type of application as well as the system design. Some general examples of this are as follows:

Motion Transfer Drives: Motion transfer drives, by definition, are required to carry extremely light torque loads. In these applications, belt installation tension is needed only to cause the belt to conform to and mesh properly with the pulleys. The amount of tension necessary for this is referred to as the minimum tension (T_{st}). Minimum tensions on a per span basis are included in **Table 16**, on page T-48. Some motion transfer drives carry very little torque, but have right registration requirements. These systems may require additional static (or installation) tension in order to minimize registration error.

Normal Power Transmission Drives: Normal power transmission drives should be designed in accordance with published torque ratings and a reasonable service factor (between 1.5 and 2.0). In these applications, belt installation tension is needed to allow the belt to maintain proper fit with the pulleys while under load, and to prevent belt ratcheting under peak loads. For these drives, proper installation tension can be determined using two different approaches. If torque loads are known and well defined, and an accurate tension value is desired, **Equation** (**10-1**) or **Equation** (**10-2**) should be used. If the torque loads are not as well defined, and a quick value is desired for use as a starting point, values from **Table 17** can be used. All static tension values are on a per span basis.

$$T_{st} = \frac{0.812\,DQ}{d} + mS^2 \quad \text{(lb)} \qquad \textbf{(10-1)}$$

(For drives with a Service Factor of 1.3 or greater)

$$T_{st} = \frac{1.05\,DQ}{d} + mS^2 \quad \text{(lb)} \qquad \textbf{(10-2)}$$

(For drives with a Service Factor less than 1.3)

where: T_{st} = Static tension per span (lbs)
DQ = Driver design torque (lb·in)
d = Driver pitch diameter (in)
S = Belt speed/1000 (ft/min)
where Belt speed = (Driver pitch diameter x Driver rpm)/3.82
m = Mass factor from **Table 16**

T-47

Figure 4: Reference 1 T-47

Table 16 Belt Tensioning Force

Belt	Belt Width	*m*	*Y*	Minimum T_{st} (lbs) Per Span
2 mm GT	4 mm	0.026	1.37	1.3
	6 mm	0.039	2.05	2.0
	9 mm	0.058	3.08	3.0
	12 mm	0.077	4.10	4.0
3 mm GT	6 mm	0.077	3.22	2.2
	9 mm	0.120	4.83	3.3
	12 mm	0.150	6.45	4.4
	15 mm	0.190	8.06	5.5
5 mm GT	9 mm	0.170	14.9	8.4
	15 mm	0.280	24.9	14.1
	20 mm	0.380	33.2	18.7
	25 mm	0.470	41.5	23.4
3 mm HTD	6 mm	0.068	3.81	2.5
	9 mm	0.102	5.71	4.3
	15 mm	0.170	9.52	7.8
5 mm HTD	9 mm	0.163	14.9	6.3
	15 mm	0.272	24.9	12.0
	25 mm	0.453	41.5	21.3
MXL	1/8"	0.003	1.40	1.0
	3/16"	0.004	2.11	1.7
	1/4"	0.005	2.81	2.3
XL	1/4"	0.010	3.30	3.2
	3/8"	0.015	4.94	5.1

NOTE: Y = constant used in **Equations** (**10-4**) and (**10-5**).

Registration Drives: Registration drives are required to register, or position accurately. Higher belt installation tensions help in increasing belt tensile modulus as well as in increasing meshing interference, both reducing backlash. Tension values for these applications should be determined experimentally to confirm that desired performance characteristics have been achieved. As a beginning point, use values from **Table 17** multiplied by 1.5 to 2.0.

Table 17 Static Belt Tension, T_{st} (lbs) Per Span – General Values

Belt	4 mm	6 mm	9 mm	12 mm	15 mm	20 mm	25 mm
2 mm GT	2	3	4	5	—	—	—
3 mm GT	—	8	11	15	19	25	—
5 mm GT	—	—	18	22	27	35	43
3 mm HTD	—	5	9	12	16	22	—
5 mm HTD	—	—	13	18	24	33	43

Belt	1/8"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"
MXL	2	3	3	4	5	—	—
XL	2	3	4	5	6	8	9

Most synchronous belt applications often exhibit their own individual operating characteristics. The static installation tensions recommended in this section should serve as a general guideline in determining the level of tension required. The drive system should be thoroughly tested to confirm that it performs as intended.

10.2 Making Measurements

Belt installation tension is generally, measured in the following ways:

Force/Deflection: Belt span tension can be measured by deflecting a belt span 1/64" per

T-48

Figure 5: Reference 1 T-48

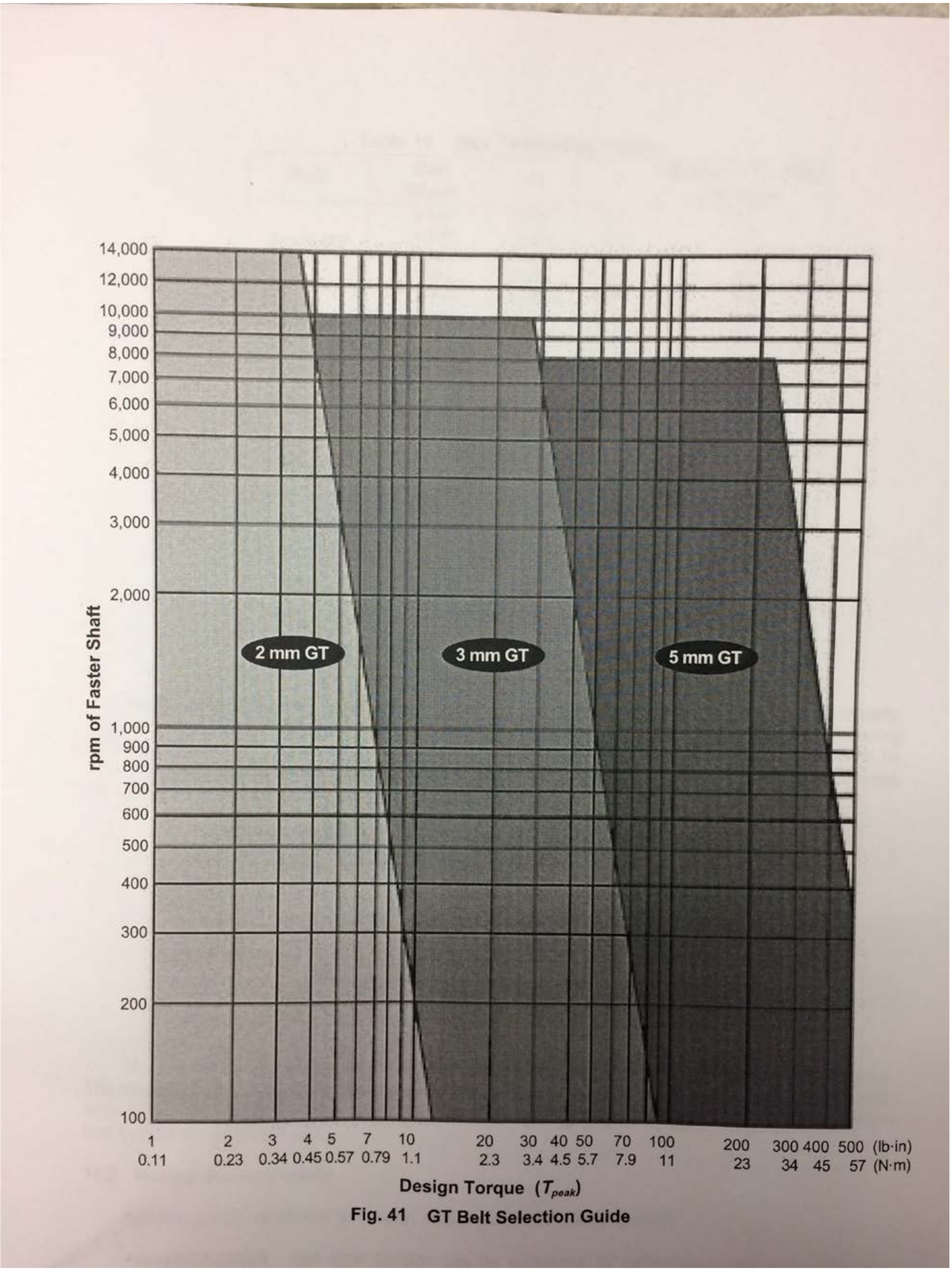


Figure 6: Reference 1 Fig. 41

Table 41 Rated Torque (lb·in) for Small Pulleys — 6 mm Belt Width

3 mm Pitch PowerGrip® GT® Belts

*The following table represents the torque ratings for each belt in its base width at the predetermined number of grooves, pitch diameters and rpm's. These ratings must be multiplied by the appropriate width factor and applicable belt length factor to obtain the corrected torque rating (see **Step 7** of **SECTION 24**, on page T-146).*

Belt Width (mm)	6	9	12	15
Width Multiplier	1.00	1.50	2.00	2.50

Number of Grooves	16	18	20	22	24	26	30	34	38	44	50	56	64	72	80
Pitch Diameter mm	15.28	17.19	19.10	21.01	22.92	24.83	28.65	32.47	36.29	42.02	47.75	53.48	61.12	68.75	76.39
Pitch Diameter inches	0.602	0.677	0.752	0.827	0.902	0.977	1.128	1.278	1.429	1.654	1.880	2.105	2.406	2.707	3.008
rpm of Fastest Shaft: 10	14.02	16.27	18.50	20.70	22.89	25.06	29.38	33.61	37.82	44.00	50.13	56.15	64.10	71.93	79.67
20	12.82	14.92	17.00	19.05	21.09	23.11	27.13	31.06	34.97	40.70	46.38	51.95	59.30	66.53	73.68
40	11.62	13.57	15.50	17.40	19.29	21.16	24.88	28.51	32.12	37.41	42.63	47.75	54.50	61.13	67.68
60	10.91	12.78	14.62	16.44	18.24	20.02	23.56	27.02	30.45	35.48	40.44	45.30	51.69	57.98	64.17
100	10.03	11.78	13.51	15.22	16.91	18.59	21.90	25.14	28.35	33.04	37.67	42.20	48.15	54.00	59.74
200	8.83	10.43	12.01	13.57	15.11	16.64	19.65	22.59	25.50	29.74	33.92	38.00	43.35	48.60	53.74
300	8.12	9.64	11.14	12.61	14.06	15.50	18.34	21.10	23.83	27.81	31.73	35.54	40.54	45.43	50.23
400	7.63	9.08	10.51	11.92	13.32	14.69	17.40	20.04	22.65	26.44	30.17	33.80	38.55	43.19	47.73
500	7.24	8.65	10.03	11.39	12.74	14.06	16.68	19.22	21.73	25.38	28.96	32.45	37.00	41.45	45.79
600	6.92	8.29	9.64	10.96	12.26	13.55	16.09	18.55	20.98	24.51	27.97	31.34	35.73	40.02	44.21
800	6.43	7.73	9.01	10.27	11.52	12.74	15.15	17.49	19.79	23.14	26.41	29.59	33.73	37.76	41.69
1000	6.04	7.30	8.53	9.74	10.94	12.11	14.43	16.67	18.87	22.07	25.20	28.23	32.17	36.00	39.73
1200	5.72	6.94	8.14	9.31	10.46	11.60	13.83	15.99	18.12	21.20	24.20	27.11	30.89	34.56	38.12
1400	5.46	6.64	7.80	8.94	10.06	11.16	13.33	15.42	17.48	20.46	23.36	26.16	29.80	33.32	36.74
1600	5.22	6.38	7.51	8.62	9.71	10.78	12.89	14.93	16.93	19.81	22.62	25.34	28.85	32.25	35.54
1800	5.02	6.15	7.26	8.34	9.40	10.45	12.51	14.49	16.44	19.24	21.97	24.60	28.01	31.29	34.46
2000	4.84	5.94	7.03	8.09	9.13	10.15	12.16	14.10	16.00	18.73	21.39	23.94	27.25	30.43	33.49
2400	4.52	5.59	6.63	7.65	8.65	9.64	11.56	13.42	15.23	17.84	20.37	22.79	25.91	28.90	31.77
2800	4.25	5.28	6.29	7.28	8.25	9.20	11.05	12.84	14.58	17.08	19.49	21.80	24.76	27.58	30.27
3200	4.02	5.02	6.00	6.96	7.90	8.81	10.61	12.33	14.01	16.41	18.72	20.93	23.74	26.40	28.92
3600	3.81	4.79	5.74	6.67	7.58	8.48	10.22	11.88	13.50	15.81	18.03	20.14	22.81	25.32	27.68
4000	3.63	4.58	5.51	6.42	7.30	8.17	9.86	11.48	13.04	15.27	17.40	19.41	21.95	24.32	26.52
5000	3.24	4.14	5.02	5.87	6.71	7.52	9.10	10.60	12.05	14.09	16.02	17.81	20.04	22.05	23.86
6000	2.91	3.77	4.61	5.42	6.21	6.98	8.46	9.86	11.20	13.07	14.81	16.40	18.32	19.98	21.39
8000	2.40	3.19	3.95	4.69	5.40	6.09	7.41	8.63	9.77	11.33	12.70	13.89	15.17	16.09	16.64
10000	1.99	2.72	3.42	4.10	4.74	5.36	6.53	7.59	8.55	9.78	10.79	11.54	12.13	—	—
12000	1.64	2.32	2.97	3.59	4.17	4.73	5.75	6.64	7.41	8.32	8.93	—	—	—	—
14000	1.34	1.98	2.57	3.13	3.66	4.15	5.03	5.75	6.33	6.89	—	—	—	—	—

For Belt Length															
From Length (mm)	120	129	153	180	213	252	294	348	408	480	567	666	786	924	1092
From # of teeth	40	43	51	60	71	84	98	116	136	160	189	222	262	308	364
To Length (mm)	126	150	177	210	249	291	345	405	477	564	663	783	921	1089	1200
To # of teeth	42	50	59	70	83	97	115	135	159	188	221	261	307	363	400
Length Correction Factor	0.70	0.75	0.80	0.85	0.90	0.95	1.00	1.05	1.10	1.15	1.20	1.25	1.30	1.35	1.40

Shaded area indicates drive conditions where reduced service life can be expected.

Continued on the next page

Figure 7: Reference 1 Table 41

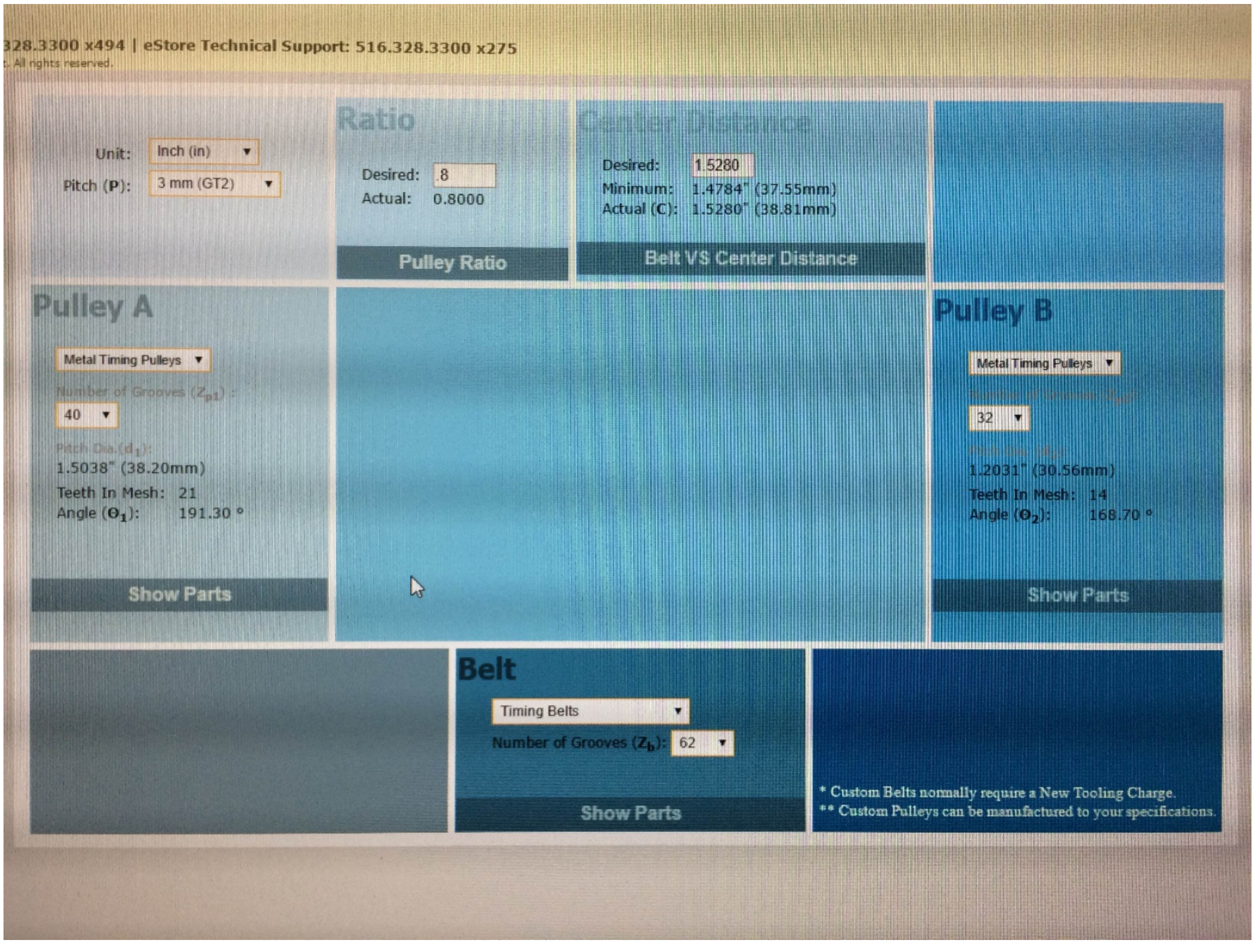
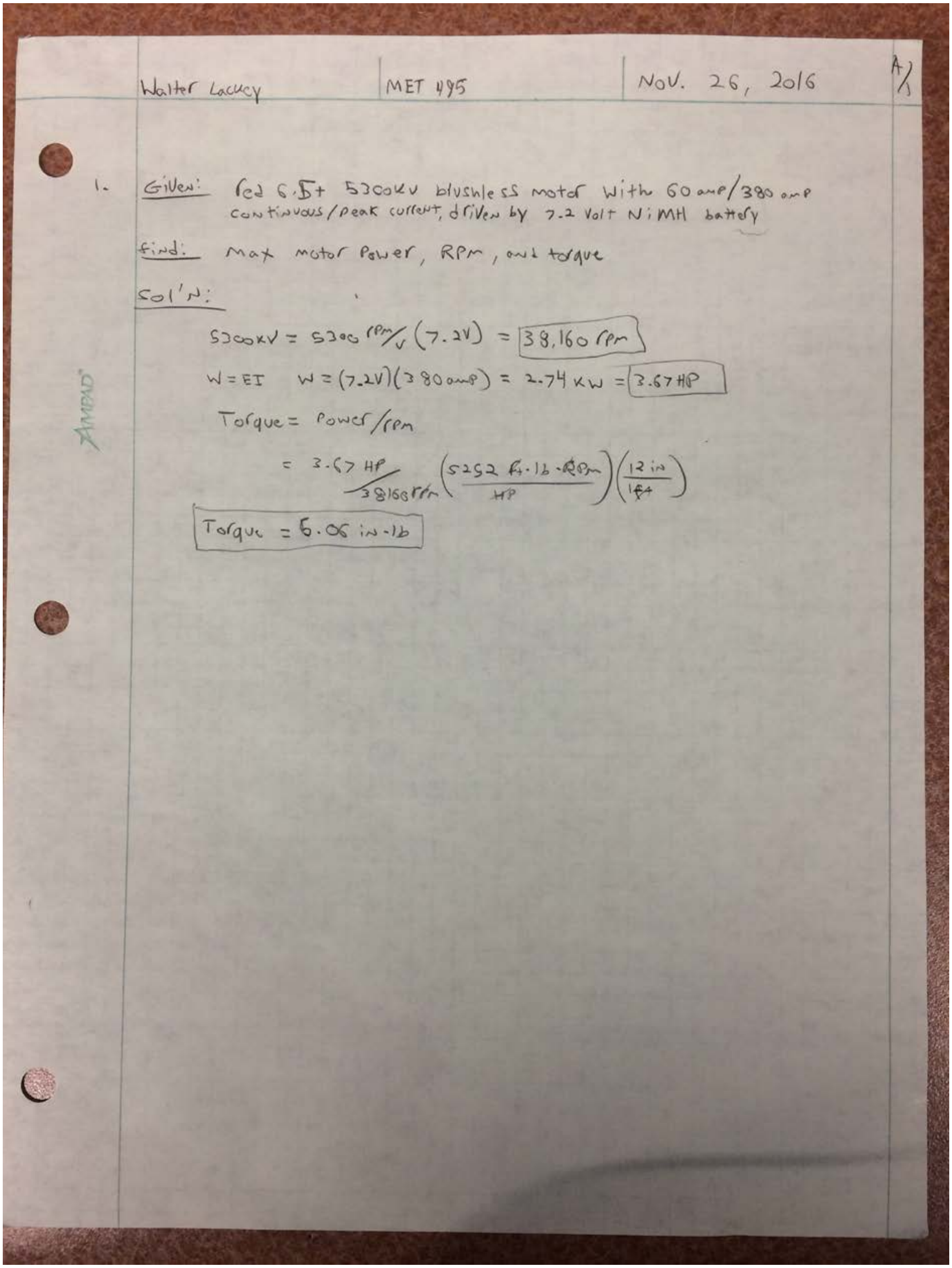


Figure 8: Reference 2 Center distance design

Appendix A Analysis



Walter Lackey | MET 495 | Nov. 26, 2016 | A1

1. Given: red 6.5t 5300kV brushless motor with 60 amp/380 amp continuous/peak current, driven by 7.2 Volt NiMH battery

Find: Max motor Power, RPM, and torque

Sol'n:

$$5300kV = 5300\ \frac{rpm}{V}(7.2V) = \boxed{38{,}160\ rpm}$$

$$W = EI \qquad W = (7.2V)(380\ amp) = 2.74\ kW = \boxed{3.67\ HP}$$

$$Torque = Power/rpm$$

$$= \frac{3.67\ HP}{38160\ rpm}\left(\frac{5252\ ft\cdot lb\cdot RPM}{HP}\right)\left(\frac{12\ in}{1\ ft}\right)$$

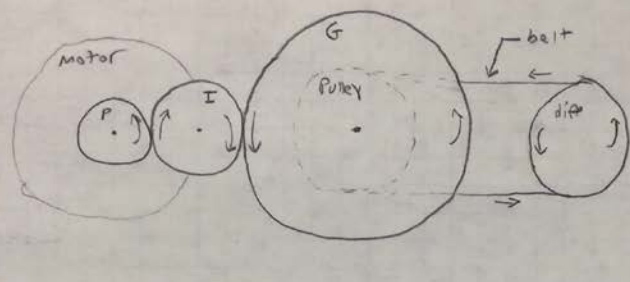
$$\boxed{Torque = 6.06\ in\text{-}lb}$$

Figure 9: A1

Walter Lacuey | MET 495 | Nov. 26, 2016 | A/2

2 **Given:** $D_{diff} \approx 1.45''$ $P_d = 48$ for all gears $fdr \approx 12$

drivetrain configuration.



Find: Choose values $N_P, N_G, N_{pulley}, N_{diff}$ while maintaining high torque holding capacity and low pitch diameters.

Sol'n: Choose $\boxed{3mm\ GT}$ timing belt pitch for high torque holding and low pitch

to maximize torque holding pulley and diff. groove count must be as high as possible. therefore, most of the gear reduction must occur between gears P and G.

Choose $\boxed{P = 10t}$ and $\boxed{G = 96t}$ as these are the smallest and largest possible values. $96/10 = 9.6$

$D_{diff.} = \frac{PN}{\pi}$ $P = .118''$ for 3mm GT $D_{diff} \approx 1.45''$

$N = \frac{D_{diff.}\ \pi}{P}$

$N = \frac{(1.45'')\pi}{.118''}$

$N = 39$ choose 40 for increased torque holding

$D_{diff.} = \frac{(.118'')(40)}{\pi}$

$\boxed{D_{diff} = 1.503'' \text{ @ } N = 40}$

$fdr = 9.6\left(\frac{N_{diff}}{N_{Pulley}}\right)$

$N_{P_{pulley}} = \frac{9.6\ N_{diff}}{fdr}$

$N_{pulley} = \frac{9.6(40)}{12}$

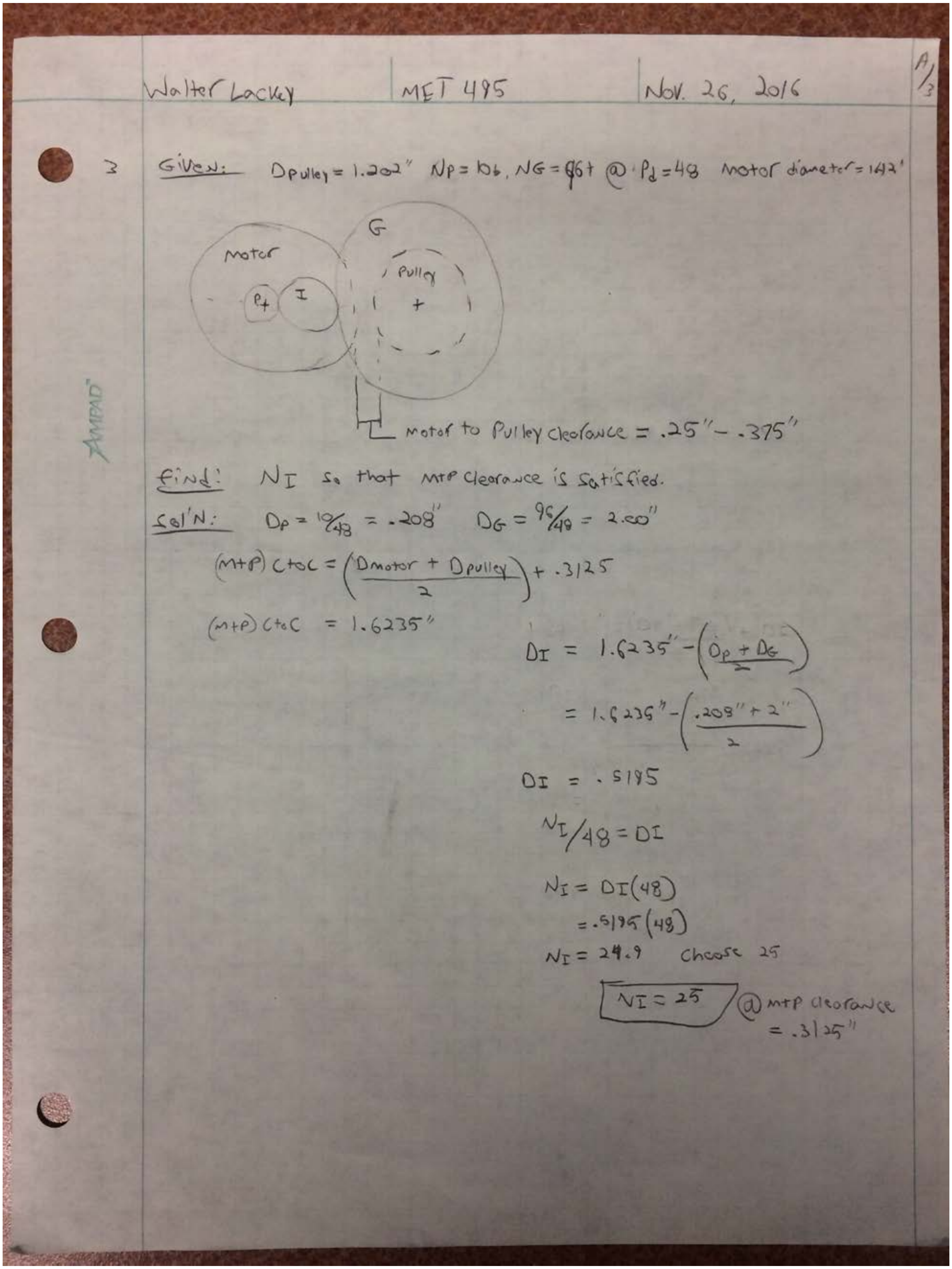
$N_{pulley} = 32$ choose $\boxed{32 = N_{pulley}}$

$D_{pulley} = \frac{PN}{\pi} = \frac{.118''(32)}{\pi}$

$\boxed{D_{pulley} = 1.202''}$

$\boxed{fdr = 12}$

Figure 10: A2



Walter Lackey | MET 495 | Nov. 26, 2016 | A/3

3 Given: $D_{pulley} = 1.202''$ $N_P = 10$, $N_G = 96$ @ $P_d = 48$ motor diameter $= 1.42''$

Find: N_I so that MtP clearance is satisfied.

Sol'N: $D_P = 10/48 = .208''$ $D_G = 96/48 = 2.00''$

$$(M+P)\ CtoC = \left(\frac{D_{motor} + D_{pulley}}{2}\right) + .3125$$

$$(M+P)\ CtoC = 1.6235''$$

$$D_I = 1.6235'' - \left(\frac{D_P + D_G}{2}\right)$$

$$= 1.6235'' - \left(\frac{.208'' + 2''}{2}\right)$$

$$D_I = .5195$$

$$N_I/48 = D_I$$

$$N_I = D_I(48)$$

$$= .5195(48)$$

$N_I = 24.9$ Choose 25

$\boxed{N_I = 25}$ @ MtP clearance $= .3125''$

Figure 11: A3

Walter Lackey | MET 495 | Nov. 26, 2016 | A/4

4

given: static belt tension (T_{st}) = 8 lb D_{diff} = 1.503'' N_G=96 N_P=10
factor (m) for 3mm GT @ 16mm wide = .190 S_m = 38160 RPm
D_{pulley} = 1.202''

find: design torque (DQ)

Sol'N:

$$T_{st} = \frac{1.05(DQ)}{d} + mS^2$$

s = belt speed / 1000 (ft/min)

shaft speed $\left(\pi D_{pulley}\right)\left(\frac{1 ft}{12 in}\right)$ = belt speed

$$\text{shaft speed} = S_m / \left(\frac{N_G}{N_P}\right)$$

$$= 38160\,rpm / \left(\frac{96}{10}\right)$$

shaft speed = 3975 RPm

$$\text{belt speed} = 3975 \frac{rev}{min}\left(\frac{\pi(1.202\ in)}{rev}\right)\left(\frac{1 ft}{12 in}\right)$$

belt speed = 1251 ft/min

$$s = \left(1251\ ft/min\right) / 1000$$

$$s = 1.251\ ft/min$$

$$DQ = \frac{T_{st}\,d}{1.05} - mS^2$$

$$= \frac{(8 lb)(1.503'')}{\left(1.05\ ft^2/min^2\right)} - .19\left(1.251\ ft/min\right)^2$$

$$\boxed{DQ = 11.15\ in \cdot lb}$$

Figure 12: A4

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5 - Given: tire diameter (D_t) = 4.5 in   FDR = 12   motor speed (s_m) = 38160 rpm
tire ballooning = .5 in   Drive train efficiency = 90% (η)

find: top speed of vehicle.

Sol'n:

$$\frac{s_m(\eta)}{fdr} = \text{axle speed}$$

$$\frac{[38{,}160(.90)]}{12} = 2862 \text{ rpm} = \text{axle speed}$$

$$\text{Tire rollout} = \pi\left(D_t + \text{ballooning}\right)\frac{\text{in}}{\text{rev}}\left(\frac{1\text{ ft}}{12\text{ in}}\right)\left(\frac{1\text{ mile}}{5280\text{ ft}}\right)$$

$$= \frac{\pi(5.0\text{ in})}{12 \cdot 5280}\ \frac{\text{mile}}{\text{rev}}$$

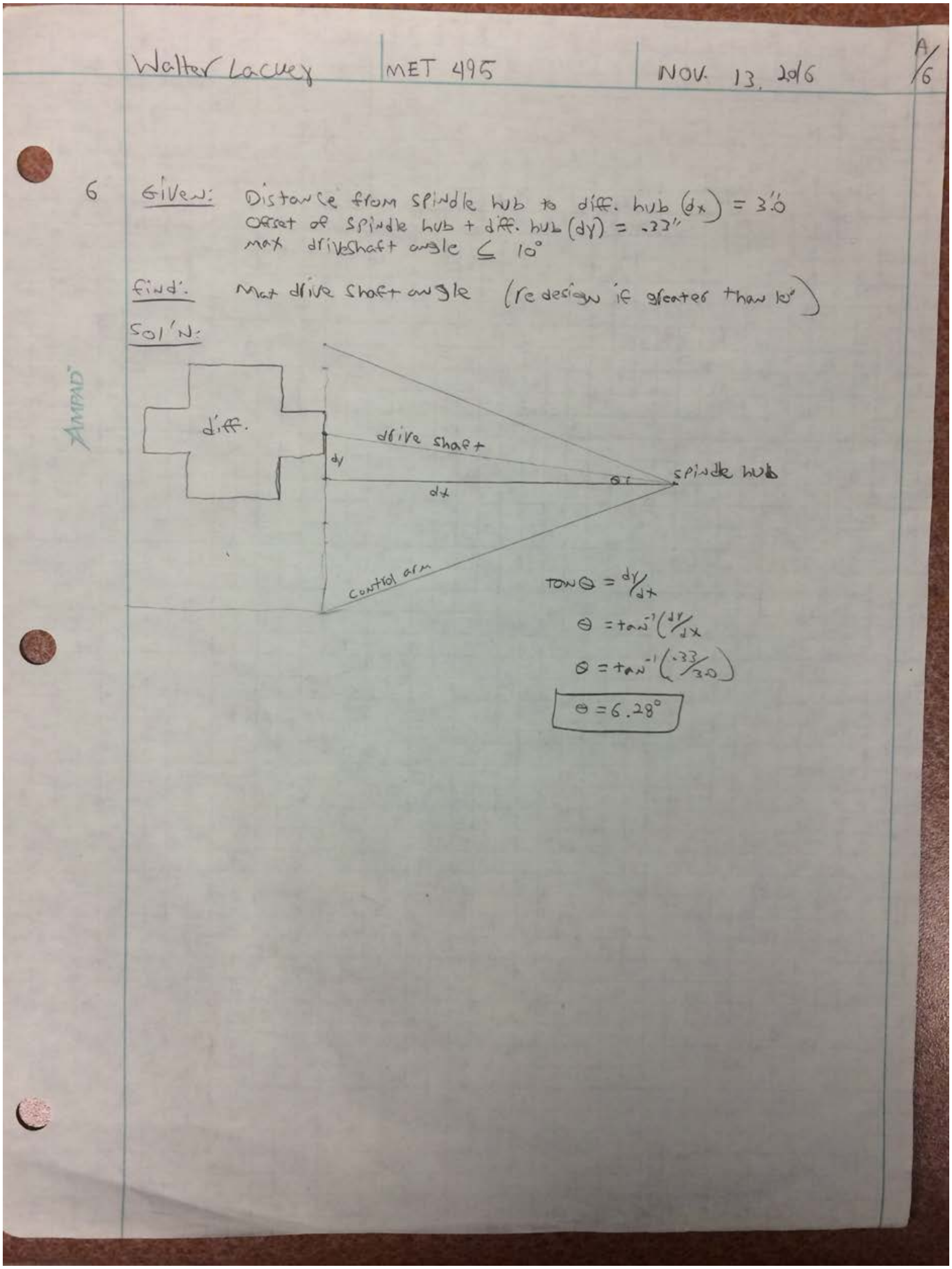
$$\text{Tire rollout} = 247.916\text{E-}6\ \text{mile/rev}$$

$$\text{Top speed} = (\text{axle speed})\ \text{tire rollout}$$

$$= \left(2862\ \frac{\text{rev}}{\text{min}}\right) 247.916\text{E-}6\ \frac{\text{mile}}{\cancel{\text{rev}}}\left(\frac{60\text{ min}}{1\text{ hr}}\right)$$

$$\boxed{\text{Top speed} = 42.6\text{ mph}}$$

Figure 13: A5



Walter Lacuey | MET 495 | NOV. 13, 2016 | A/6

6 Given: Distance from spindle hub to diff. hub $(d_x) = 3.0'$
Offset of spindle hub + diff. hub $(d_y) = .33''$
max driveshaft angle $\leq 10°$

find: Max drive shaft angle (redesign if greater than 10°)

Sol'n:

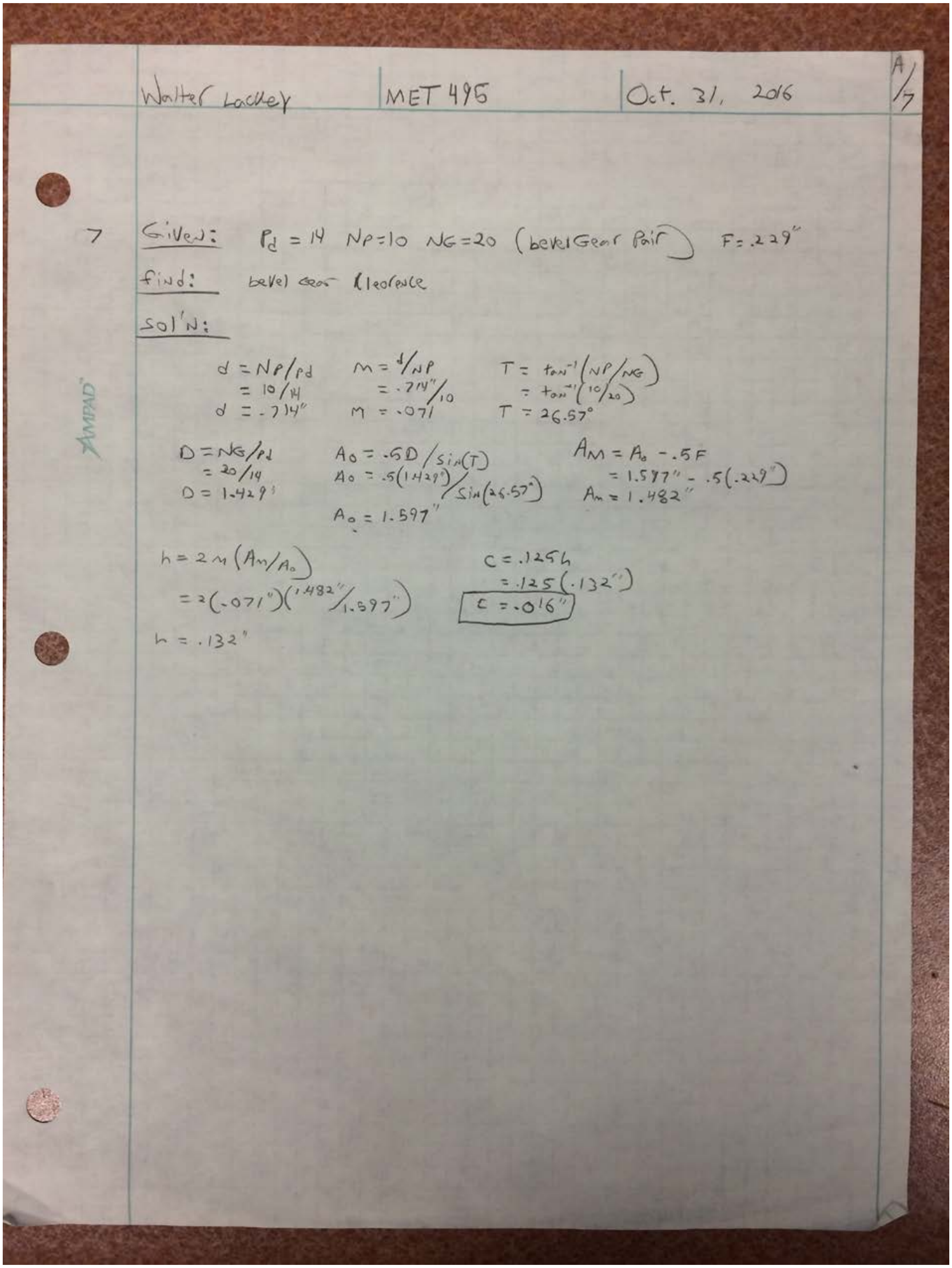
$\tan\theta = \frac{d_y}{d_x}$

$\theta = \tan^{-1}\left(\frac{d_y}{d_x}\right)$

$\theta = \tan^{-1}\left(\frac{.33}{3.0}\right)$

$\boxed{\theta = 6.28°}$

Figure 14: A6



Walter Lackey | MET 495 | Oct. 31, 2016 | A/7

7 Given: $P_d = 14$ $N_P = 10$ $N_G = 20$ (bevel Gear Pair) $F = .229''$

find: bevel gear clearance

Sol'n:

$d = N_P/P_d = 10/14$, $d = .714''$

$m = d/N_P = .714''/10$, $m = .071$

$T = \tan^{-1}(N_P/N_G) = \tan^{-1}(10/20)$, $T = 26.57°$

$D = N_G/P_d = 20/14$, $D = 1.429''$

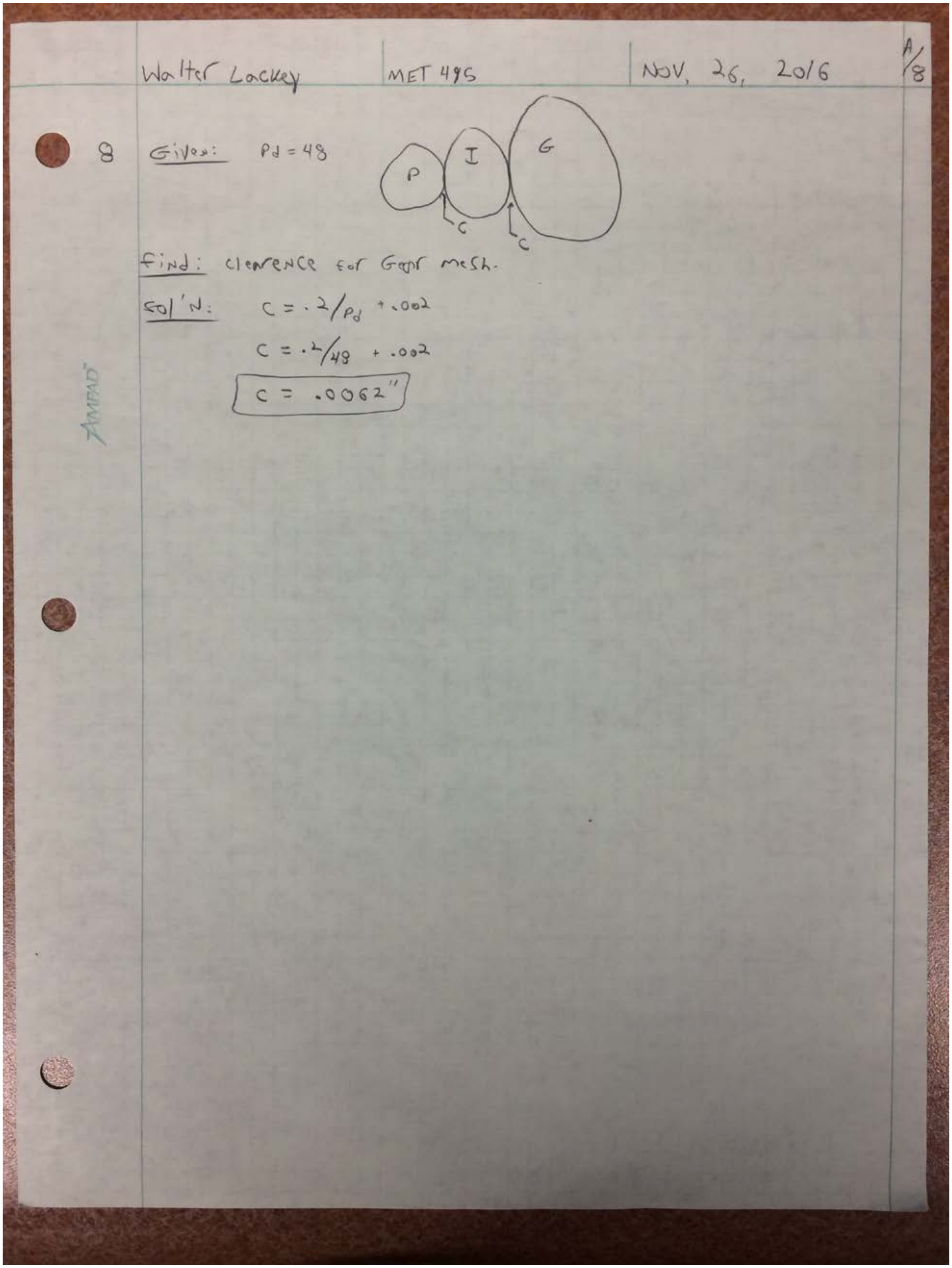
$A_o = .5D/\sin(T)$, $A_o = .5(1.429'')/\sin(26.57°)$, $A_o = 1.597''$

$A_M = A_o - .5F = 1.597'' - .5(.229'')$, $A_m = 1.482''$

$h = 2m(A_m/A_o) = 2(.071'')(1.482''/1.597'')$, $h = .132''$

$c = .125h = .125(.132'')$, $\boxed{c = .016''}$

Figure 15: A7



Walter Lackey | MET 495 | Nov, 26, 2016 | A/8

8 <u>Given:</u> $P_d = 48$

<u>Find:</u> clearence for Gear mesh.

<u>Sol'n:</u>

$$c = .2/P_d + .002$$

$$c = .2/48 + .002$$

$$\boxed{c = .0062''}$$

Figure 16: A8

Walter Lackey | MET 495 | Dec. 6, 2016 | A/9

Given: Speed of motor $(S_m) = 38160$ rpm. Number of teeth on pinion gear $(N_P) = 10$. Number of teeth on gear $(N_G) = 96$. Timing belt pulley groove $(N_c) = 32$. Design torque $(D_Q) = 11.15$ in·lb

Find: Speed of gear shaft (S_s), suitable belt pitch from fig. 41, Rated Torque from Table 41

Sol'n:

$$S_s = S_m / (N_G / N_P)$$

$$= \frac{38{,}160 \text{ rpm}}{96/10}$$

$$\boxed{S_s = 3975 \text{ rpm}}$$

from fig. 41: $\boxed{\text{Choose 3mm GT}}$

from table 41:

9.86 in·lb @ 30 groove
11.48 in·lb @ 34 groove

AVG = 10.67 in·lb @ 32 groove

Width multiplier = 2.50 @ 15mm
Length correction factor = .85 @ 60–70 tooth belt

rated torque = (10.67 in·lb)(2.50)(.85)

$$\boxed{\text{rated torque} = 22.67 \text{ in·lb}}$$

Note: tooth jump torque @ 8 lb tension = 42 in·lb from page T-8

Figure 17 A9

Appendix B – Drawings (each drawing has an ID and complies with ANSI Y14.5)

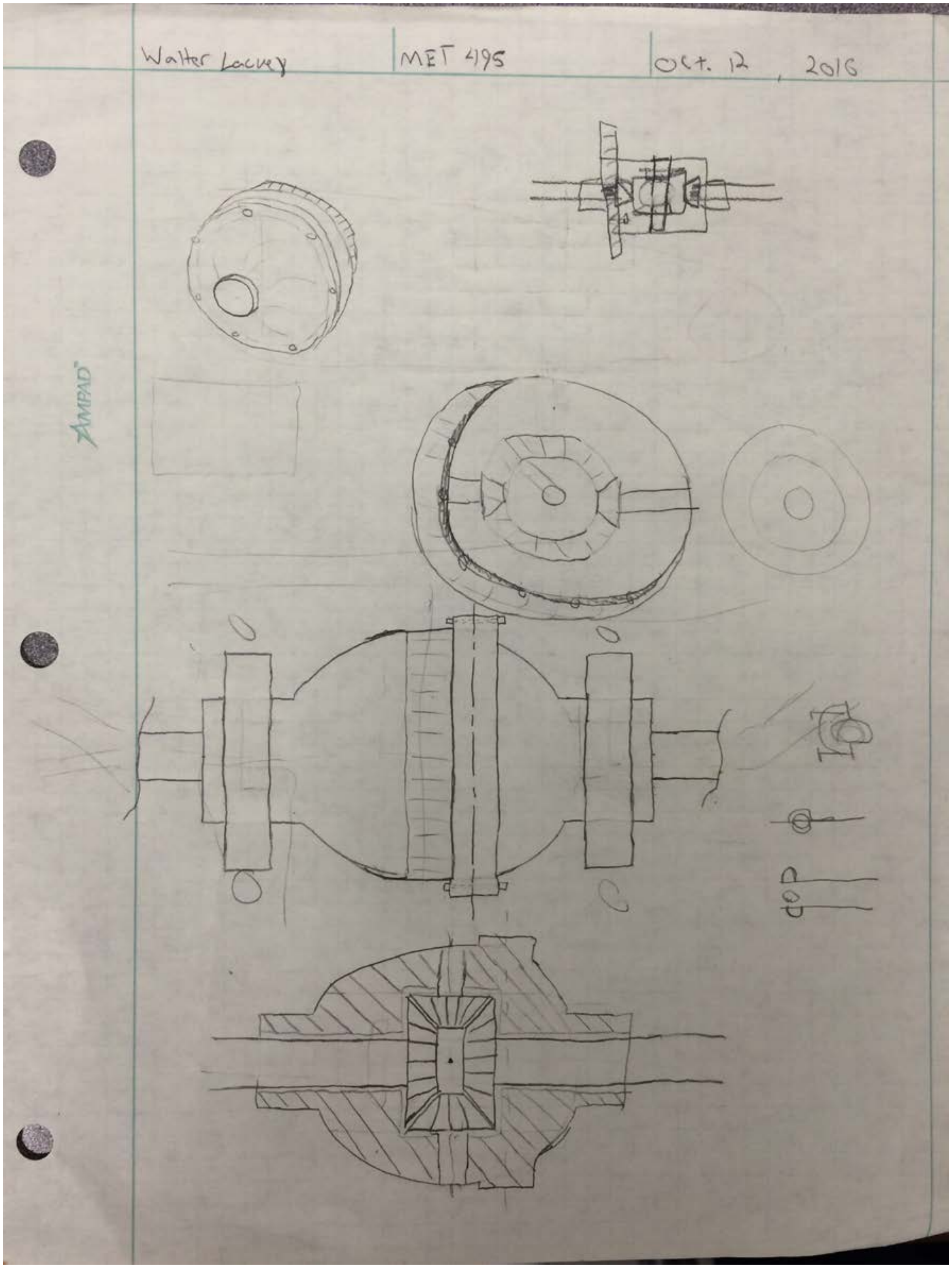


Figure 18

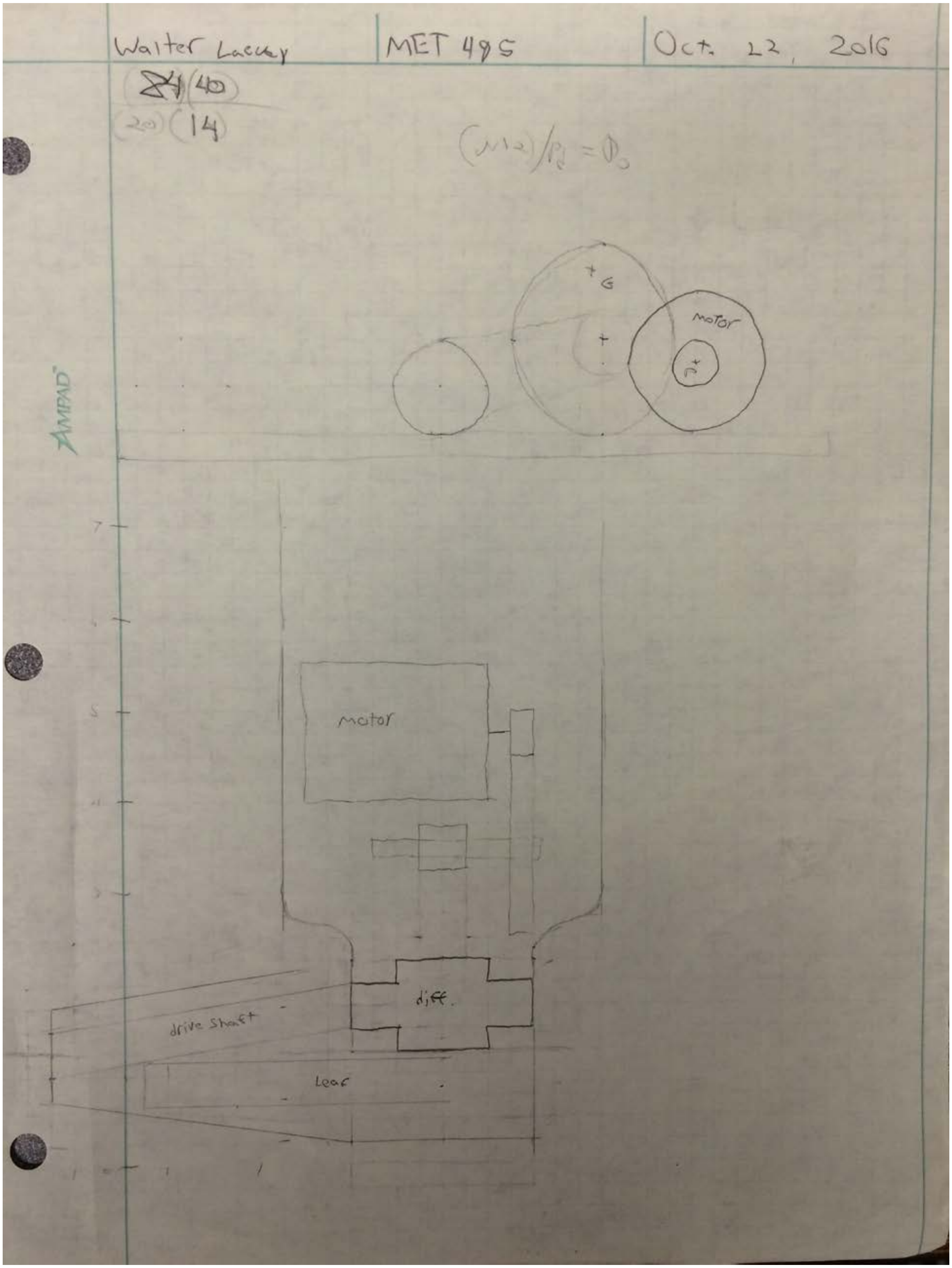

Walter Lackey
MET 495
Oct. 22, 2016
motor
motor
diff.
drive Shaft
Leaf

Figure 19

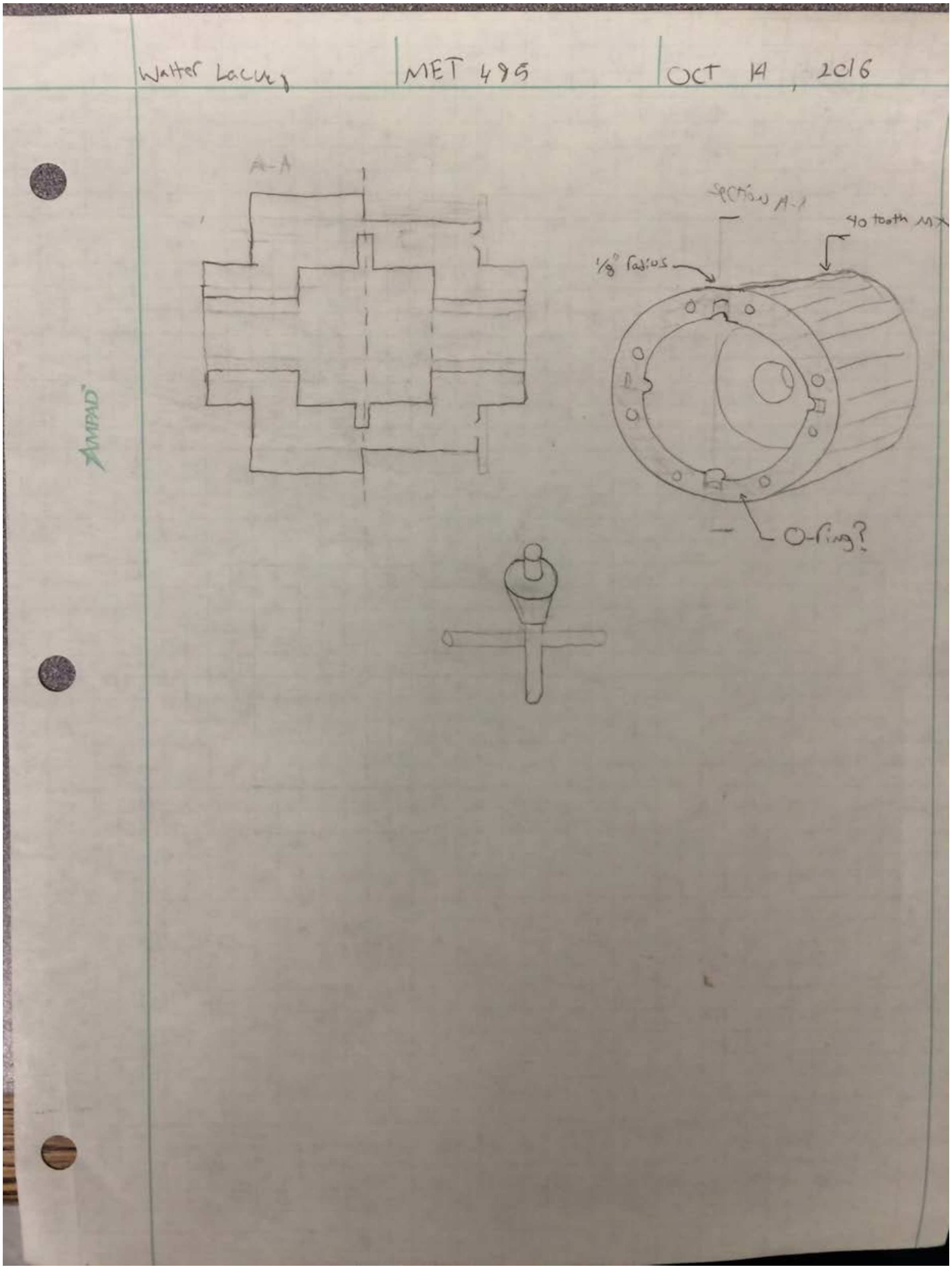

Walter Lacey
MET 495
OCT 14, 2016
A-A
Section A-A
40 tooth MX
1/8" radius
O-ring?

Figure 20

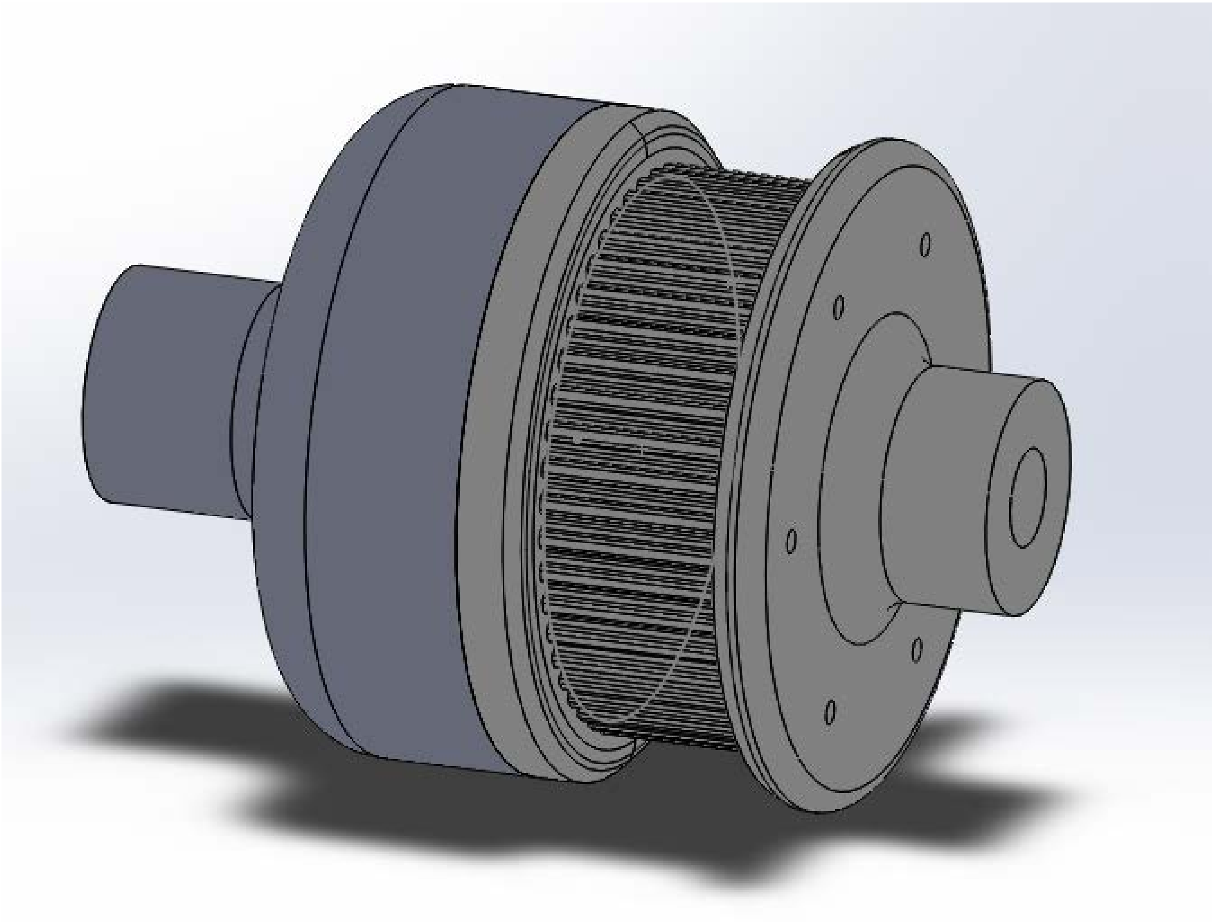


Figure 21

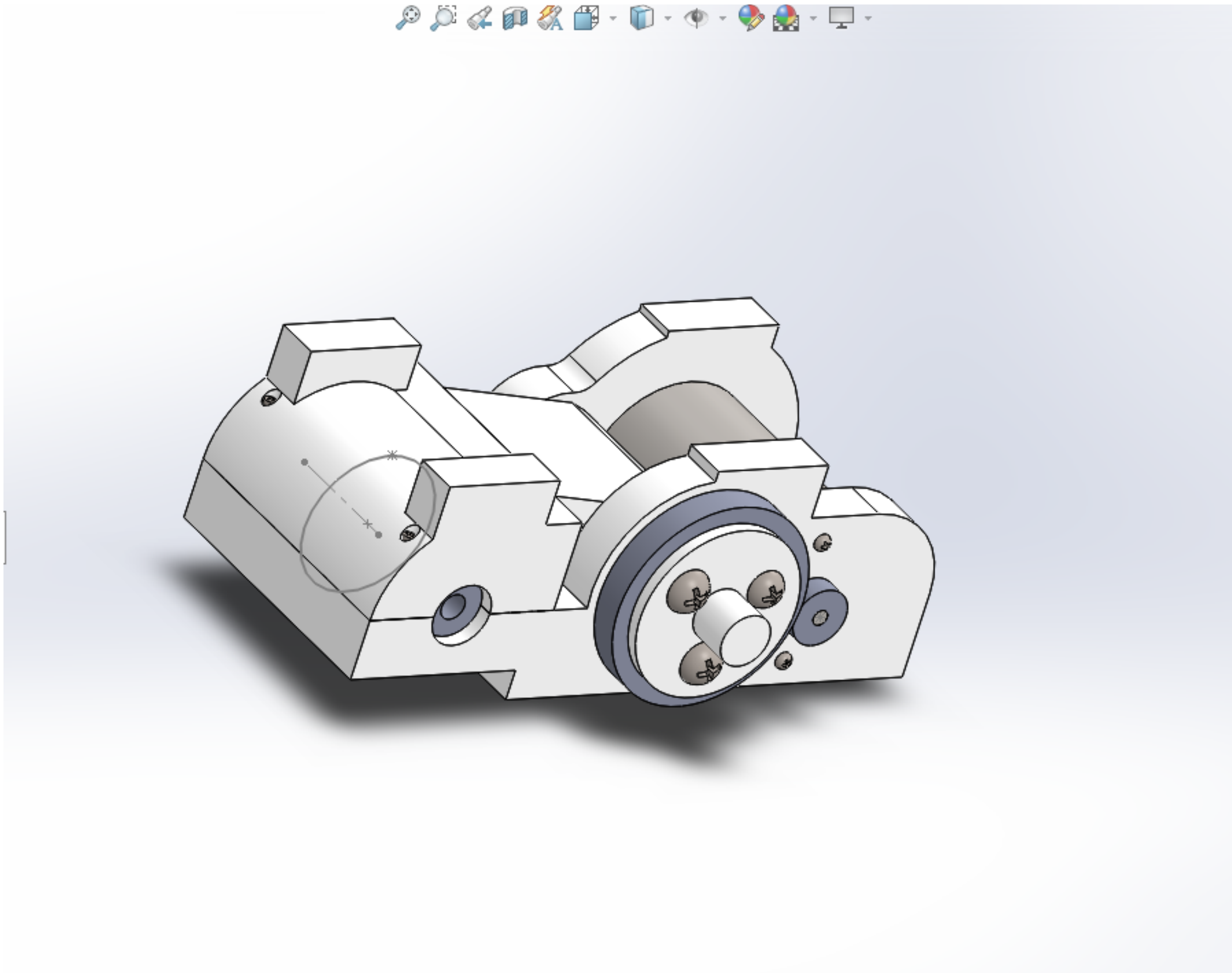


Figure 22

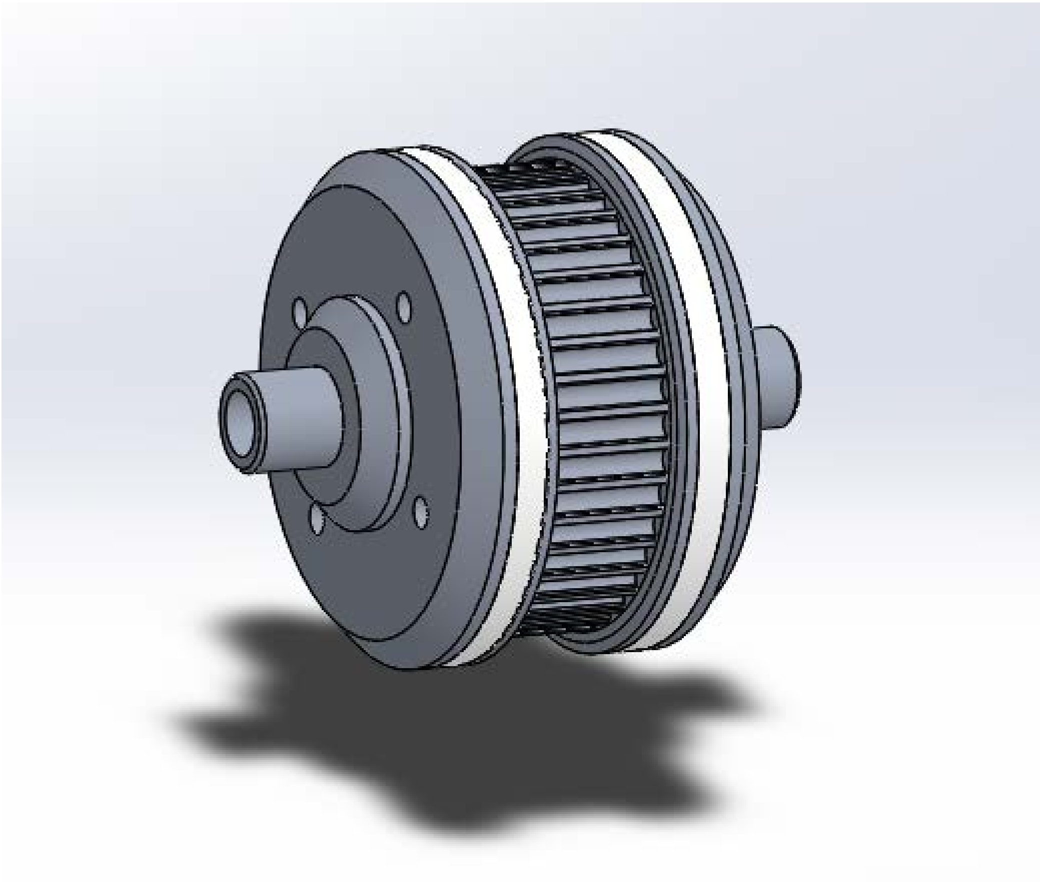


Figure 23

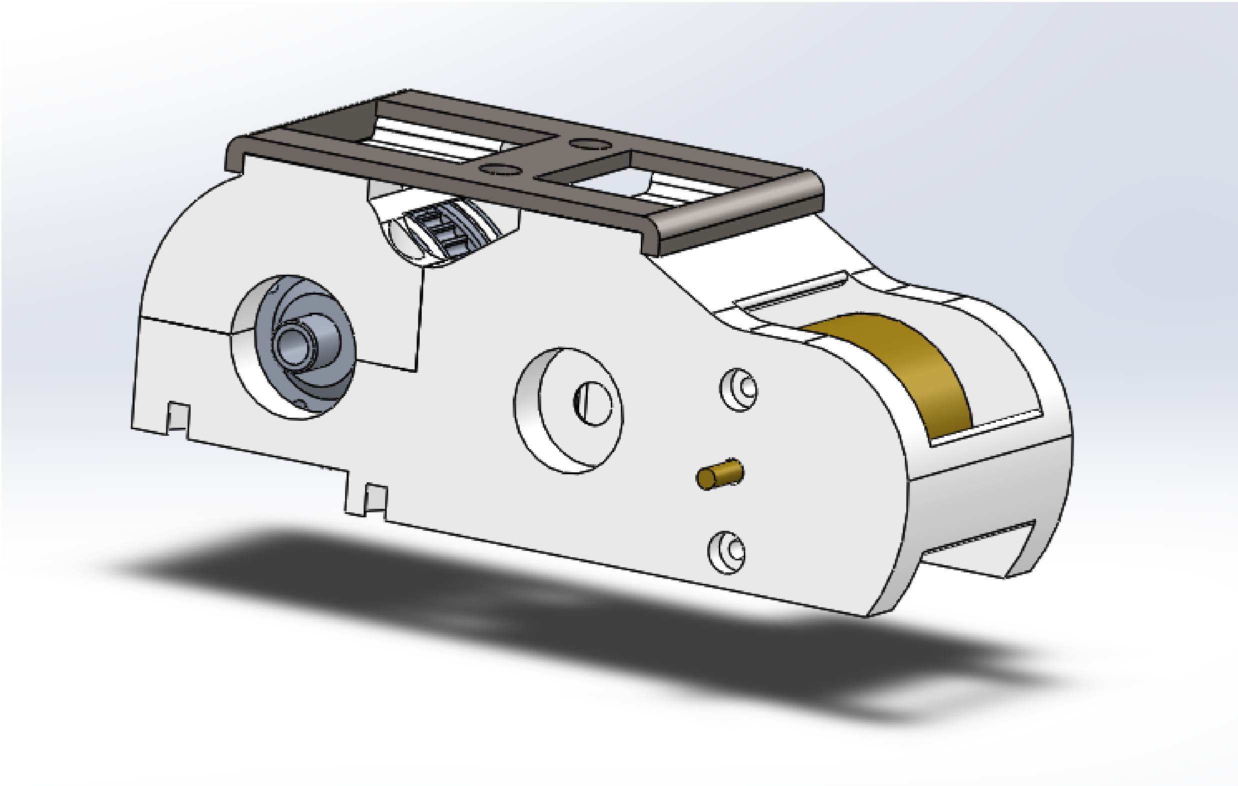


Figure 24

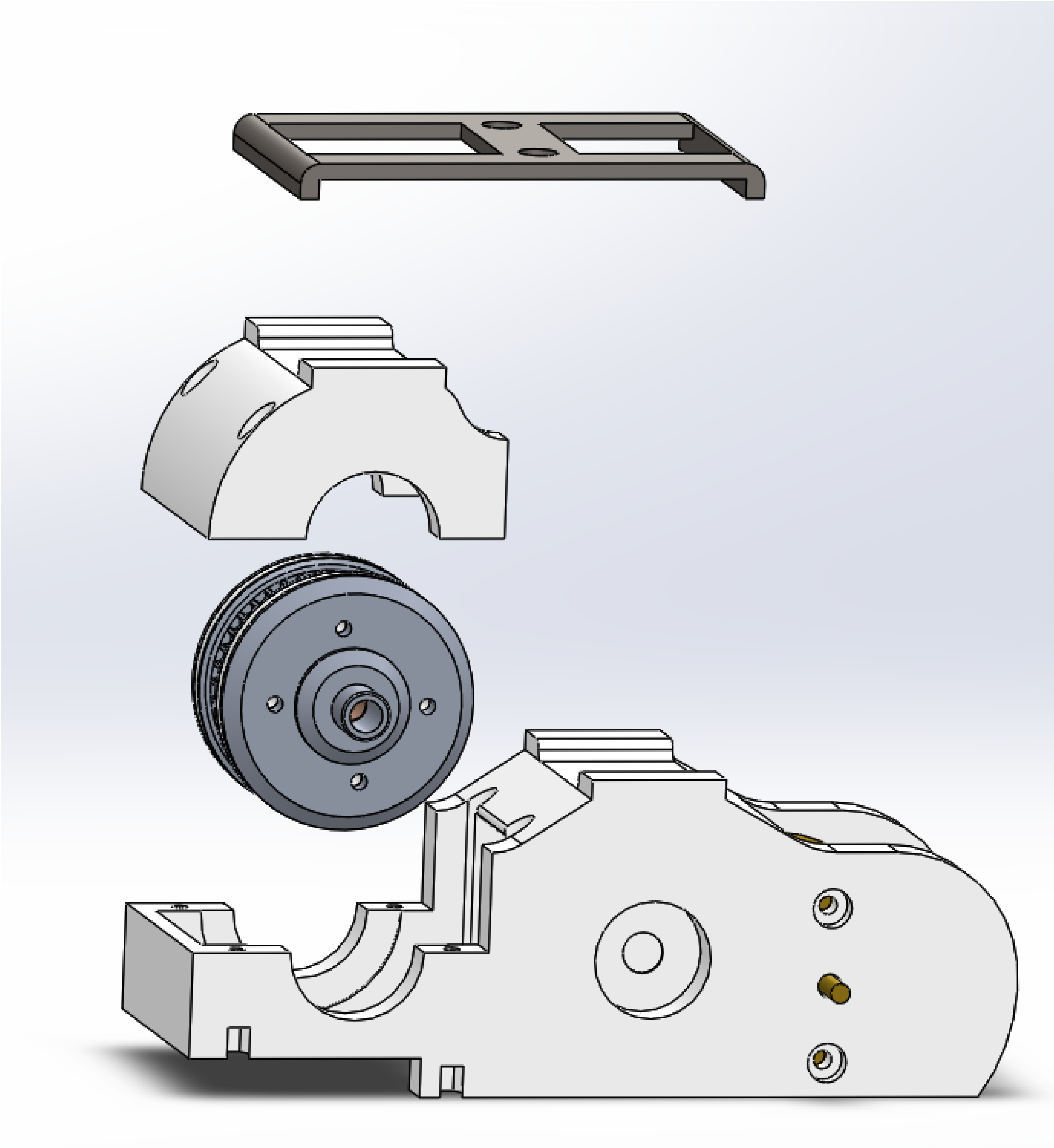
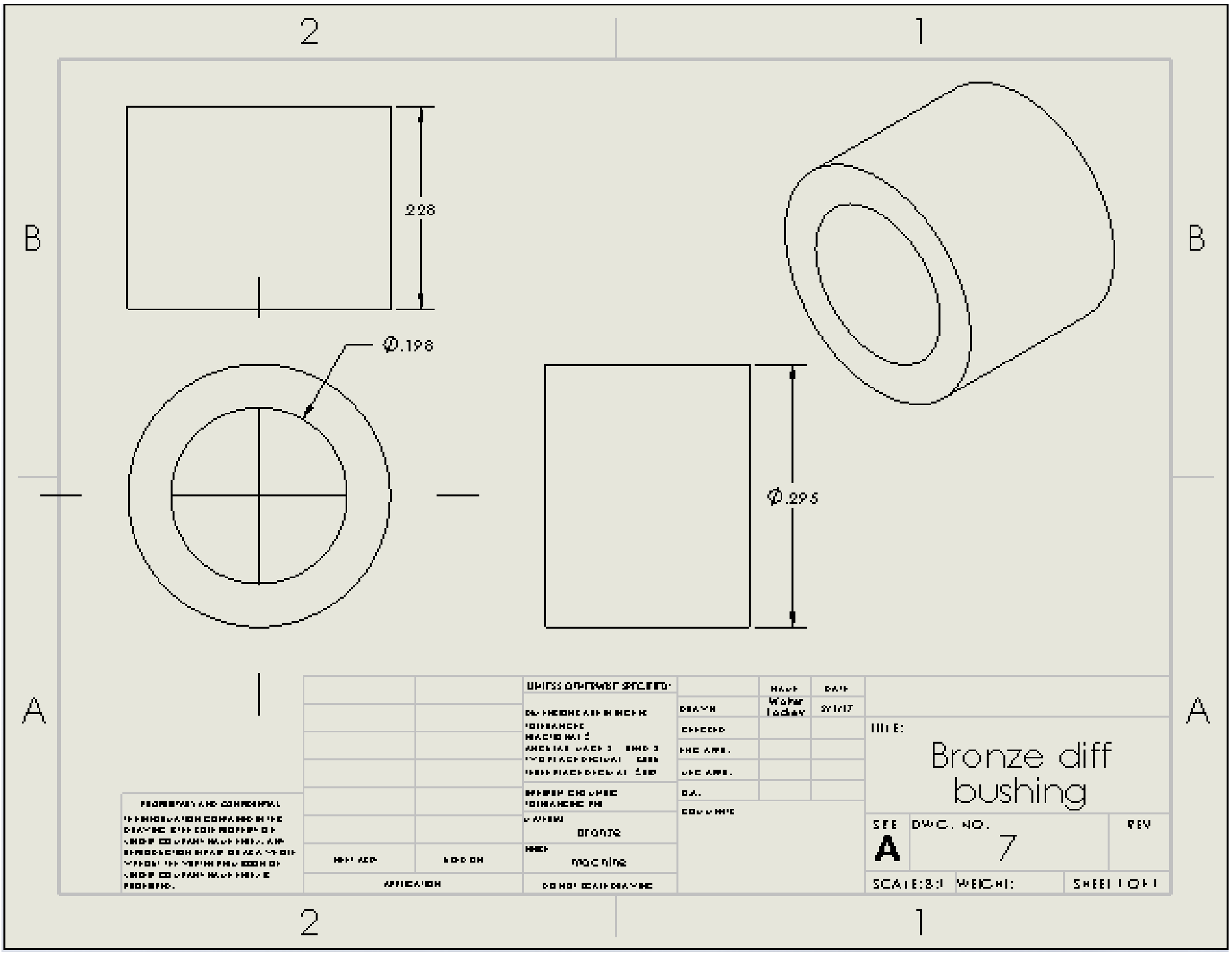


Figure 25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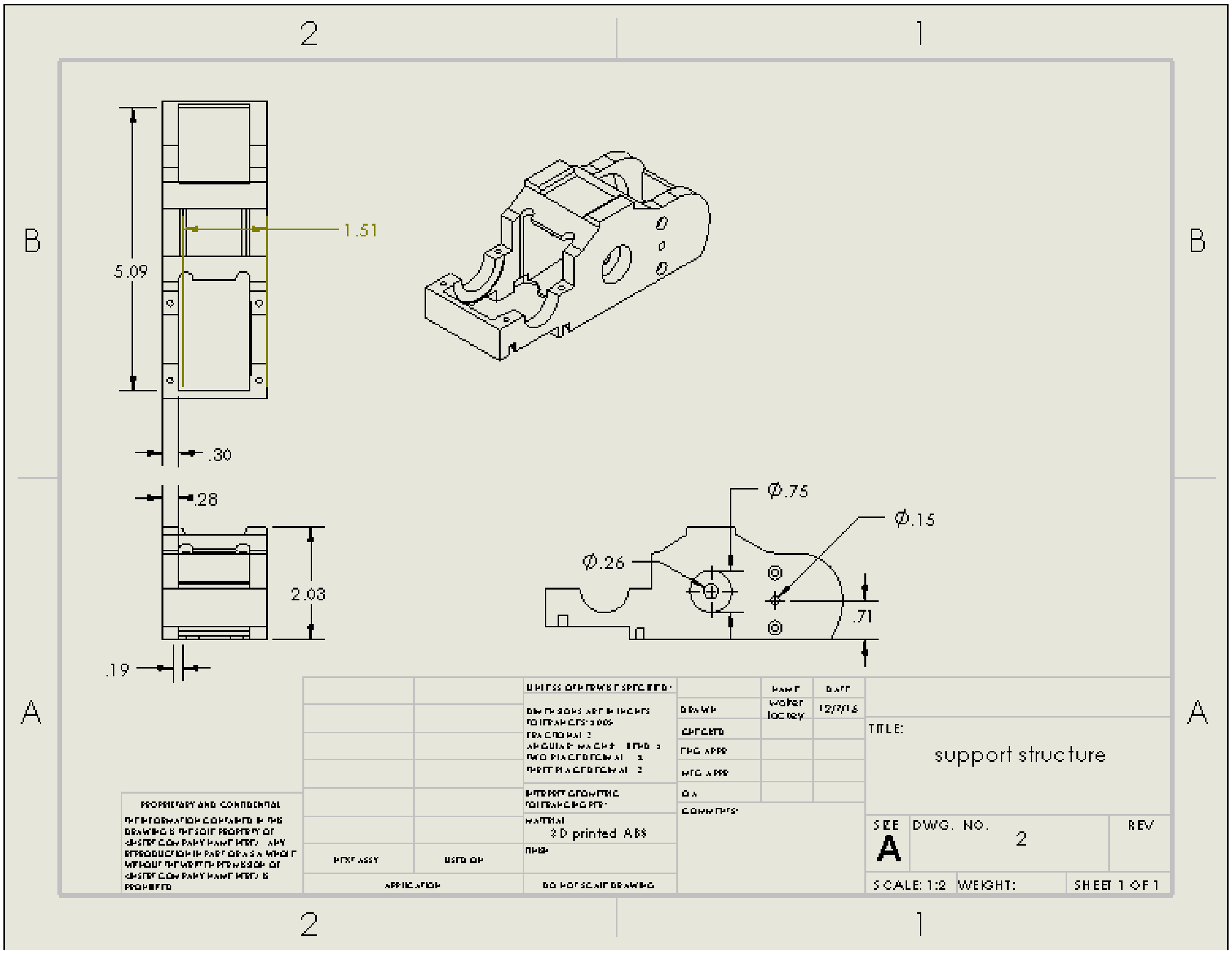
Ø.198
Ø.295
Bronze diff bushing
Bronze
machine
Walker Lackey
DWG. NO.
7
SIZE
A
SCALE: 8:1
WEIGHT:
SHEET 1 OF 1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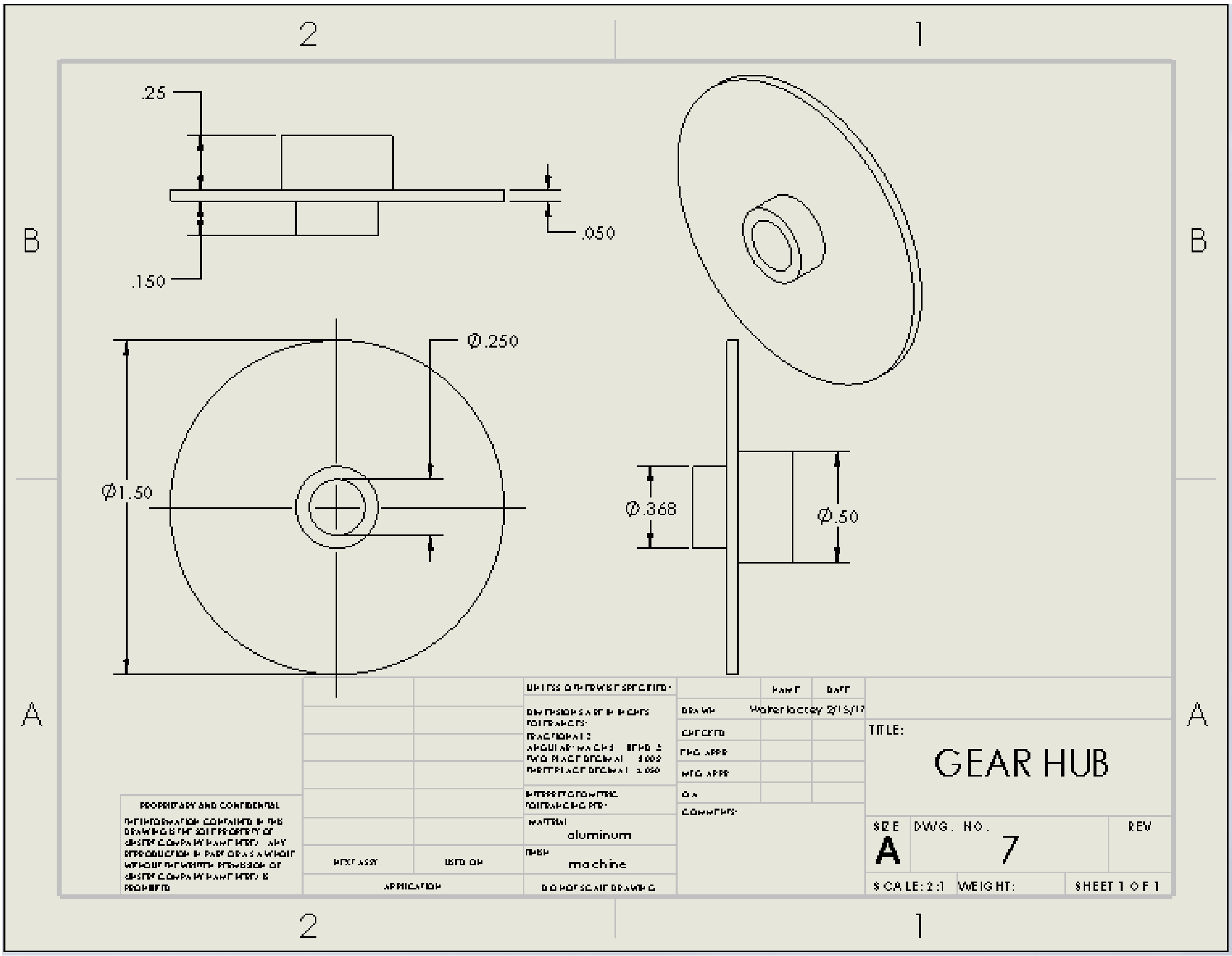
1
B
A
5.09
1.51
.30
.28
2.03
.19
Ø.75
Ø.15
Ø.26
.71
NAME
DATE
DRAWN
walter lackey
12/7/16
TITLE:
support structure
MATERIAL
3D printed ABS
SIZE
A
DWG. NO.
2
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1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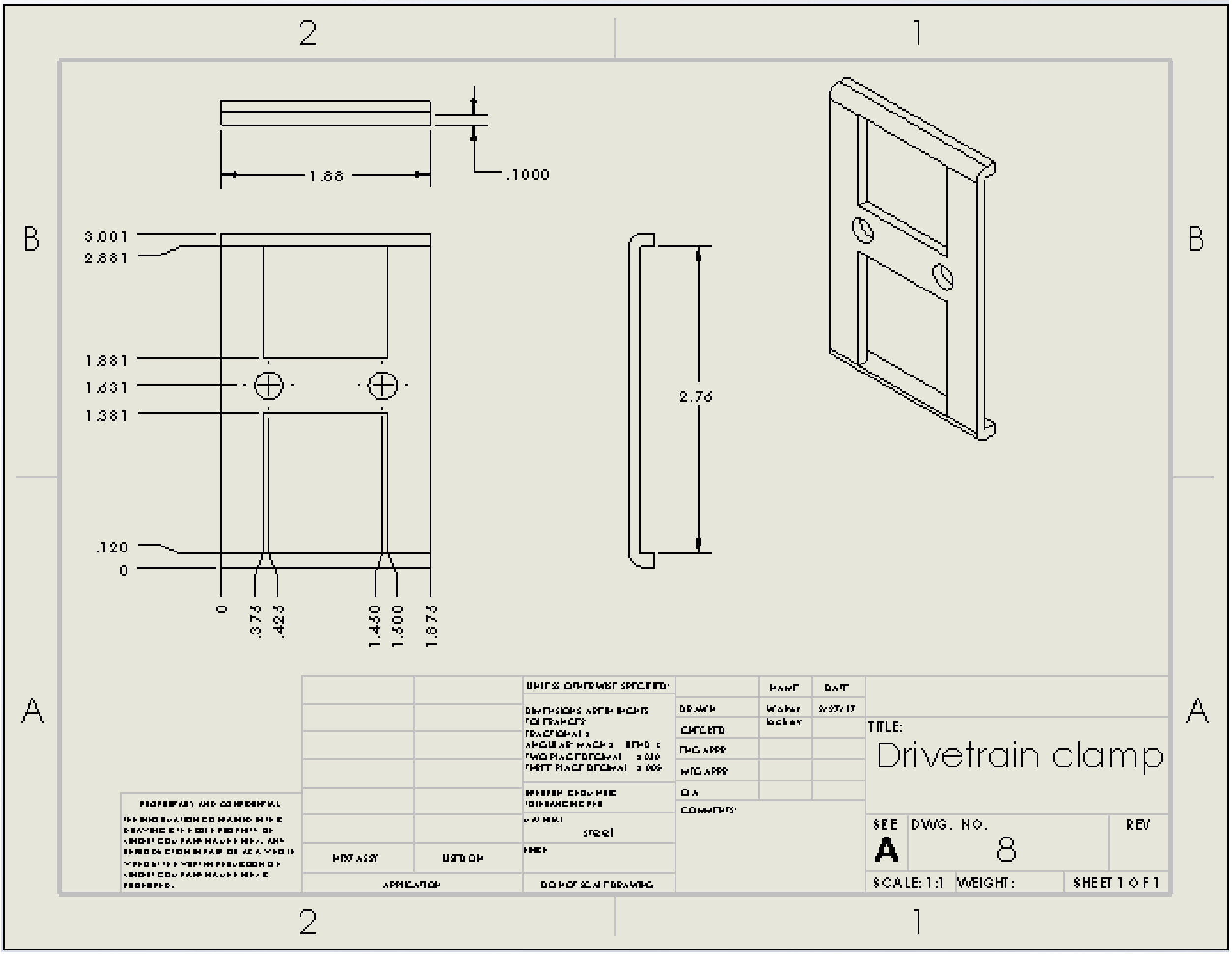
.050
.150
Ø.250
Ø1.50
Ø.368
Ø.50
Walter lacey 2/15/17
TITLE:
GEAR HUB
MATERIAL
aluminum
FINISH
machine
SIZE
A
DWG. NO.
7
REV
SCALE: 2:1
WEIGHT:
SHEET 1 OF 1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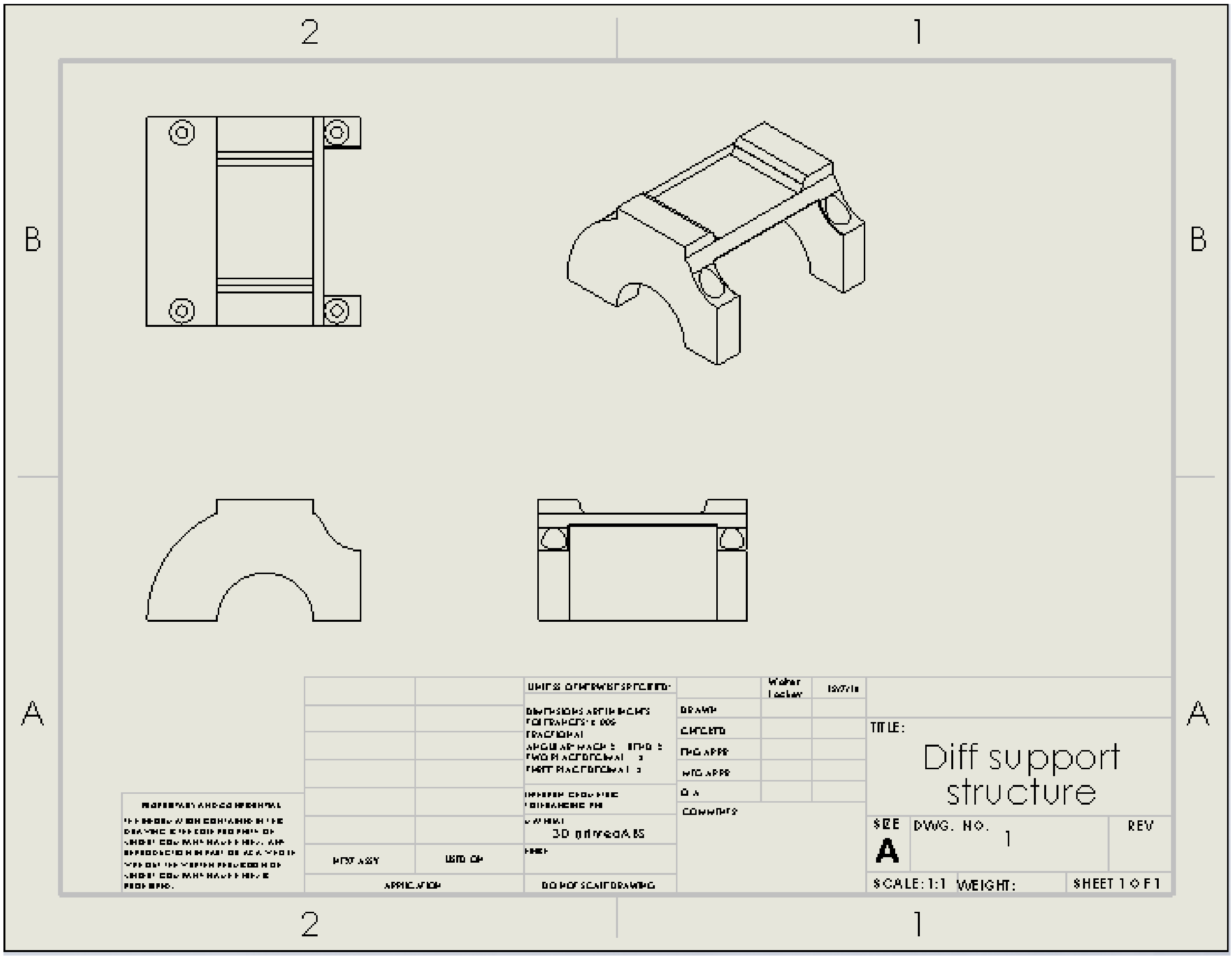
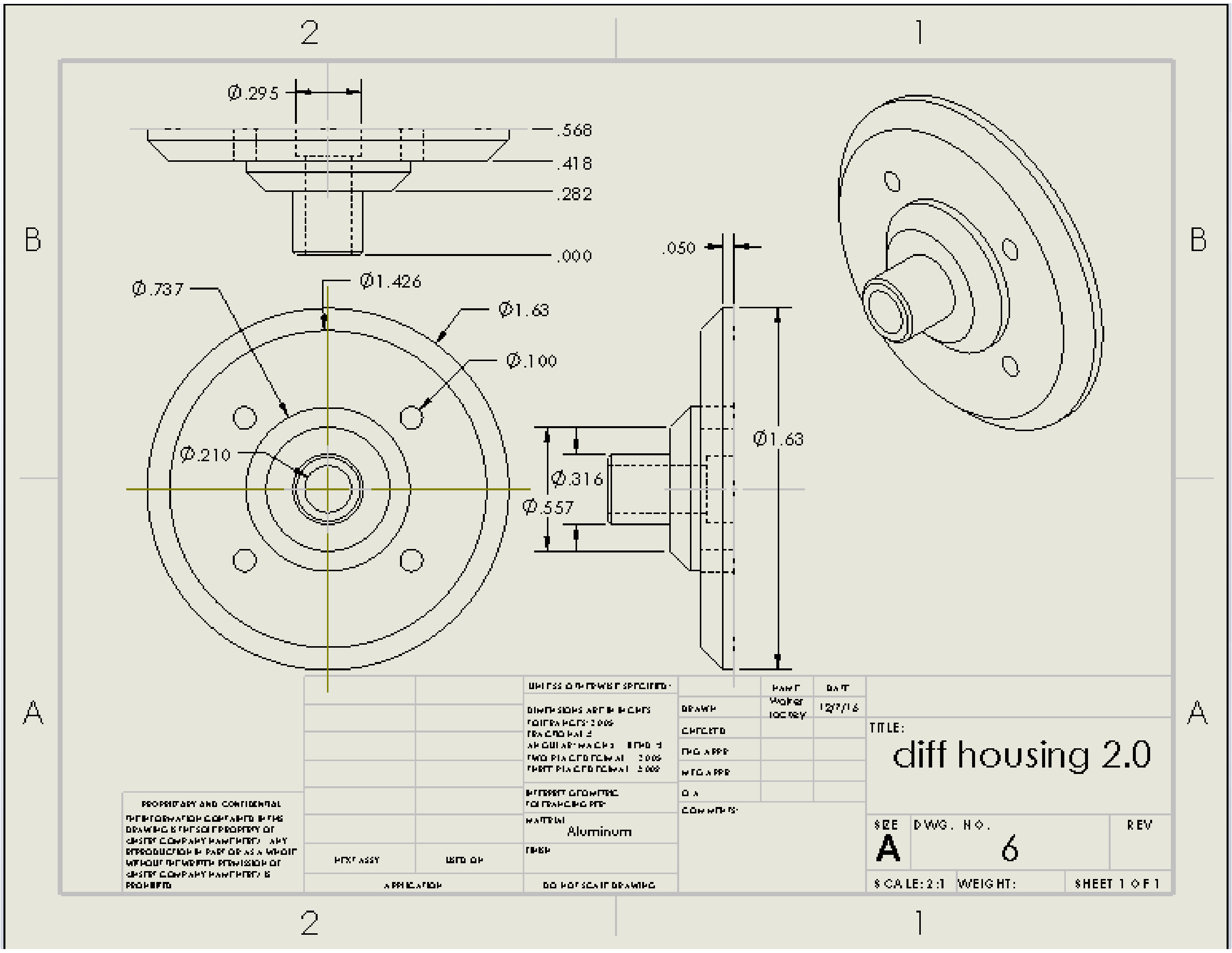
.1000
3.001
2.881
1.881
1.631
1.381
.120
0
0
.375
.425
1.450
1.500
1.875
2.76
DRAWN
Walker locker
TITLE:
Drivetrain clamp
steel
SIZE
A
DWG. NO.
8
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
APPLICATION
DO NOT SCALE DRAWING



2
1
B
A
TITLE:
Diff support structure
SIZE
A
DWG. NO.
1
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
3D printed ABS
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
DRAWN
CHECKED
COMMENTS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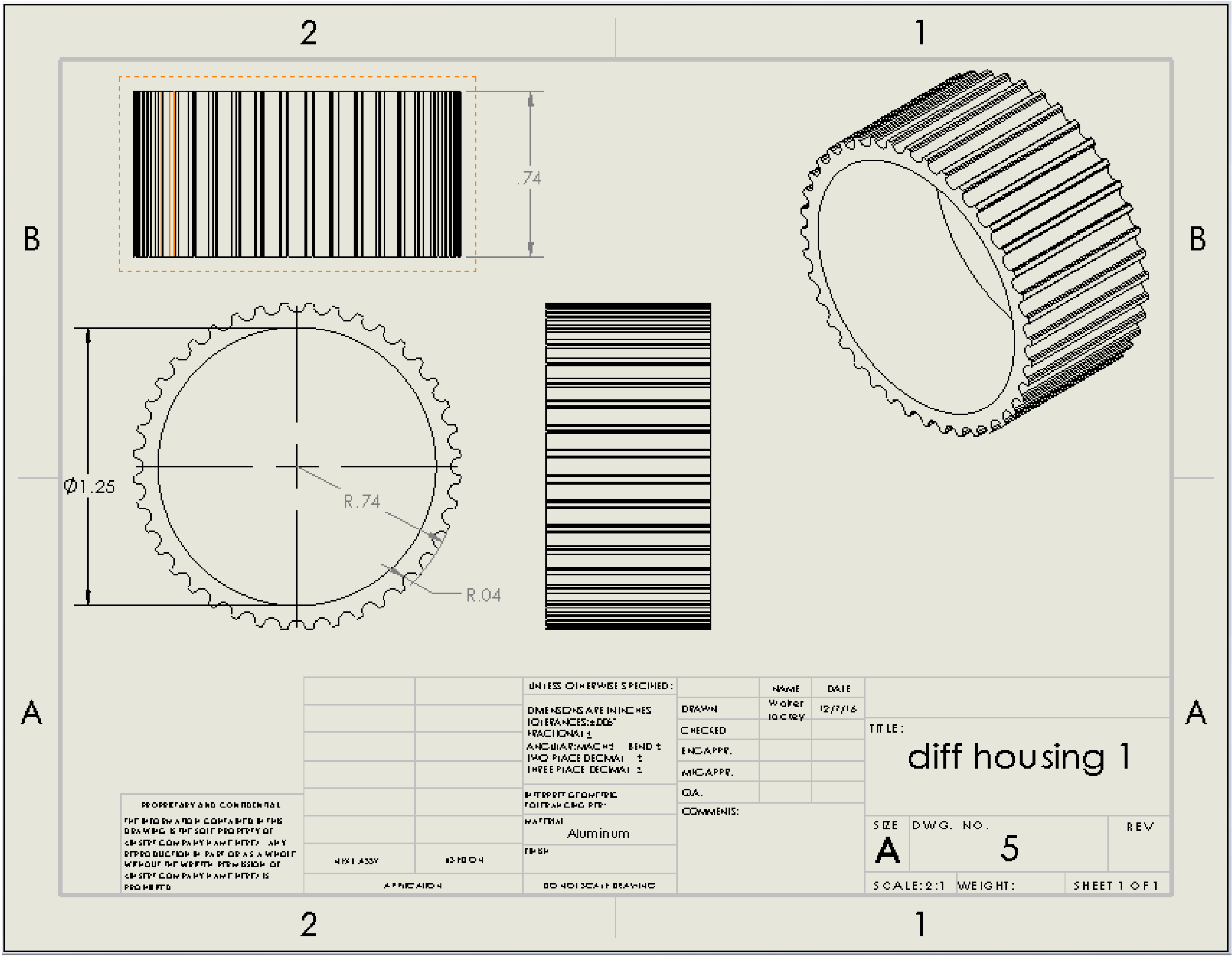
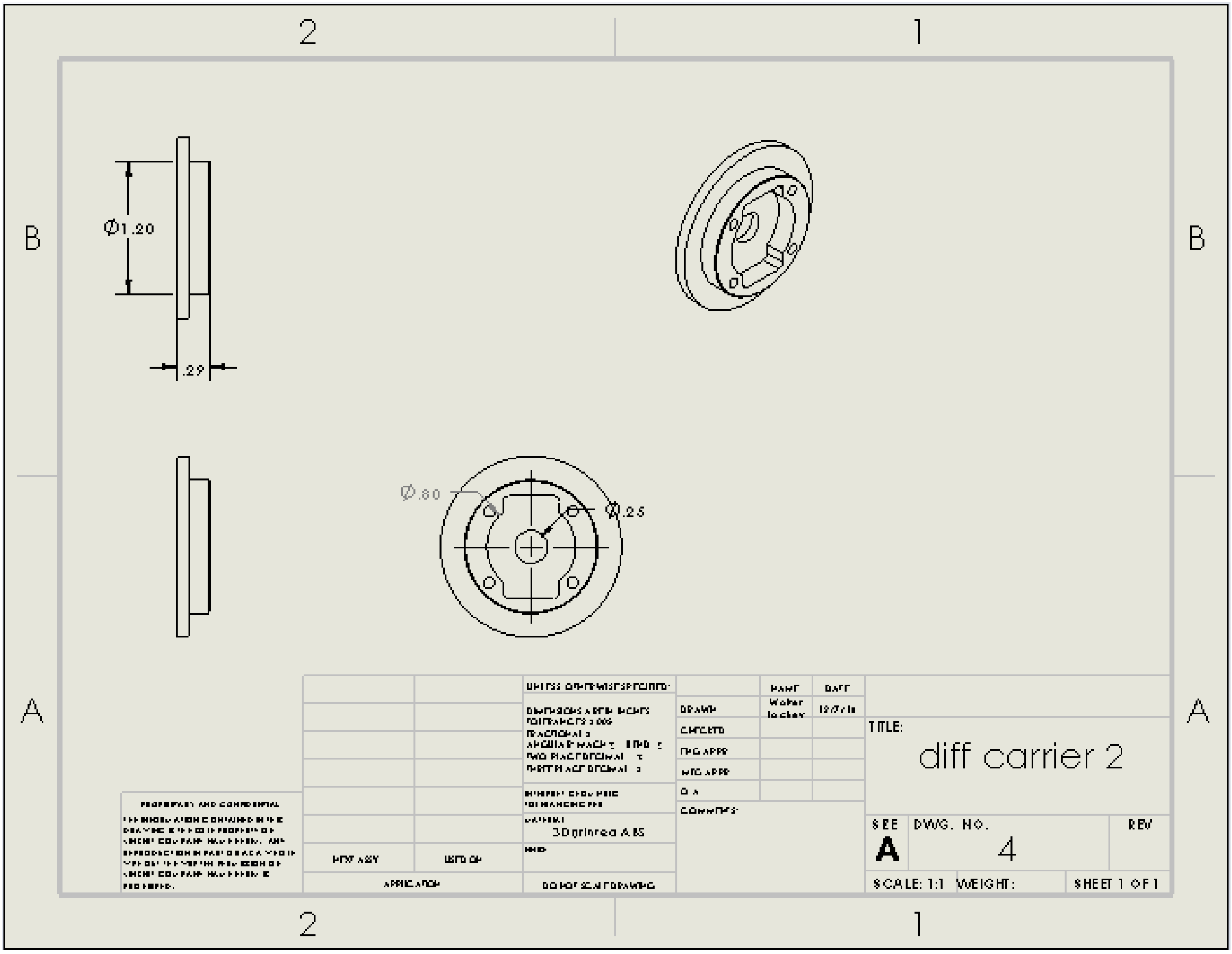
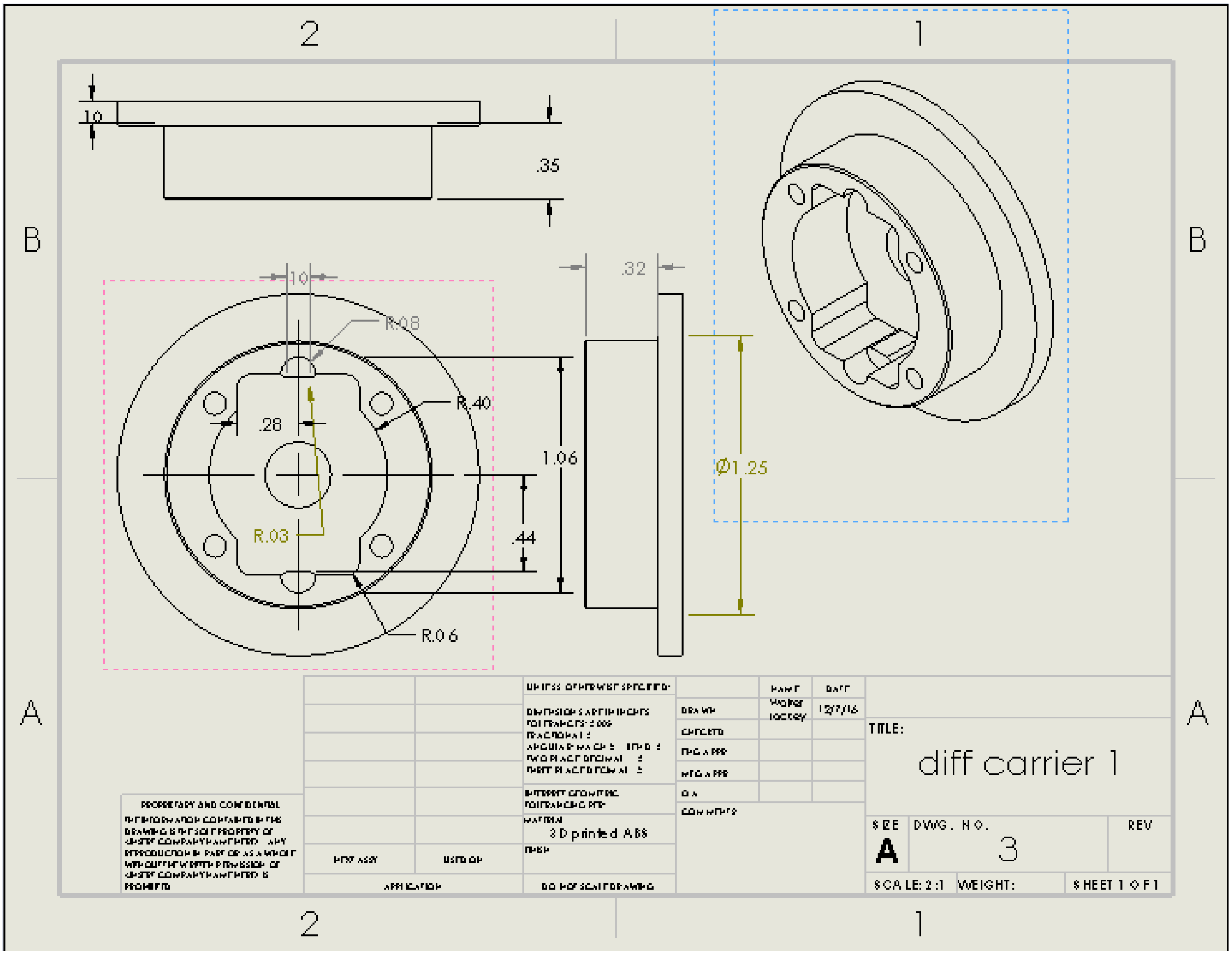
.000
.050
Ø.737
Ø1.426
Ø1.63
Ø.100
Ø.210
Ø1.63
Ø.316
Ø.557
DRAWN
Walker Jockey
12/7/16
MATERIAL
Aluminum
TITLE:
diff housing 2.0
SIZE
A
DWG. NO.
6
REV
SCALE: 2:1
WEIGHT:
SHEET 1 OF 1


2
1
B
A
.74
Ø1.25
R.74
R.04
UNLESS OTHERWISE SPECIFIED:
NAME
DATE
DRAWN
12/7/16
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
MATERIAL
Aluminum
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL
TITLE:
diff housing 1
SIZE
A
DWG. NO.
5
REV
SCALE: 2:1
WEIGHT:
SHEET 1 OF 1


Ø1.20
.29
Ø.80
Ø.25
TITLE:
diff carrier 2
SIZE
A
DWG. NO.
4
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
3D printed ABS
DO NOT SCALE DRAWING
APPLICATION
NEXT ASSY
USED ON
PROPRIETARY AND CONFIDENTIAL
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
NAME
DATE
UNLESS OTHERWISE SPECIFIED:



.10
.35
.10
R.08
.32
R.40
.28
1.06
Ø1.25
R.03
.44
R.06
UNLESS OTHERWISE SPECIFIED:
NAME
DATE
DRAWN
Walker Jackey
12/7/16
CHECKED
TITLE:
diff carrier 1
MATERIAL
3D printed ABS
PROPRIETARY AND CONFIDENTIAL
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
SIZE
A
DWG. NO.
3
REV
SCALE: 2:1
WEIGHT:
SHEET 1 OF 1

Appendix C – Parts List

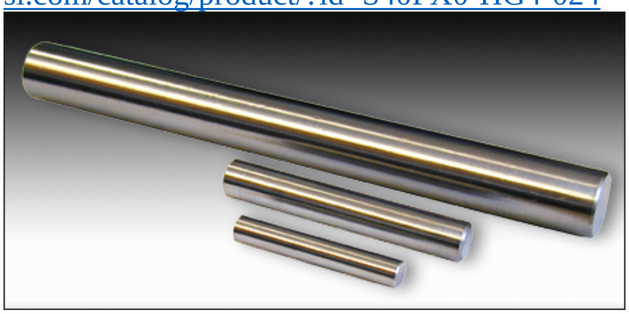
1. (2x) R4A-2RS Bearing 1/4"x3/4"x9/32" inch Sealed Miniature
   http://www.vxb.com/R4A-2RS-1-4-x3-4-x9-32-inch-Sealed-Miniature-p/kit7939.htm

2. (2x) size 608 Bones reds skateboard bearings #BSACBR88

3. (1x) 0.24970" (+.000/-.0002) Dia., 3" Long, 416 Stainless Steel Shaft http://shop.sdp-si.com/catalog/product/?id=S40PX0-HG4-024

4. (1x) 20 groove 3mm GT timing belt pulley for 9mm belt width
5. (1x) 40 groove 3mm GT timing belt pulley for 9mm belt width
6. (4x) fasteners for holding diff. together.
7. (1x) 63 groove 3mm GT timing belt at 9mm wide
8. (4x) PRO-LINE 30 Series Gladiator 2.8 w/F-11 Nitro Rear Wheels Black RC Car #10102-11

9. (4x) 0-80 fasteners for bearing cap for diff.
10. (2x) fastener to hold powertrain clamp down
11. (1x) Redcat Racing 11152 48 pitch 15 tooth spur gear
    https://redcatracing.toys/?product=redcat-racing-11152-brass-pinion-gear-22t-6-module

12. (1x) .08” diameter .5” long shaft
13. (1x) .08” inner diameter ball bearing

Appendix D – Budget

Part	Cost ($)
(1x) Atomik Red 540 6.5T 5300kv brushless motor	0.00
(1x) 3D printed support structure	36.36
(1x) 63 groove 9mm wide 3mm gt2 timing belt	8.21
(1x) HPI racing 48 pitch 96 tooth spur	0.00
(1x) Redcat Racing 48 pitch 15 tooth pinion gear	4.58
(1x) custom 6061 aluminum gear to shaft mounting flange	0.00
(1x) 0.24970" Dia. 3" Long, 416 Stainless Steel Shaft	5.74
(1x) 20 groove 9mm wide 3mm gt2 timing belt pulley	12.57
(2x) R4A-2RS Bearing 1/4"x3/4"x9/32" inch sealed	13.95
(1x) custom steel collar and set screw	0.00
(2x) differential side housing	0.00
(2x) size 608 Bones reds skateboard bearings	0.00
(4x) number 4 nuts for differential	0.00
(2x) threaded drive shaft pin	0.00
(1x) differential cap	15.37
(2x) Traxxas splined U-joint driveshaft	0.00
(1x) 40 groove 3 mm GT2 Pitch timing pulley	10.10
(1x) Differential carrier 1	6.34
(1x) Differential carrier 2	4.78
(2x) Bronze bushing	0.00
(1x) Side and spider gear set	0.00
Shipping and handling charges	25.52
Total	**143.52**

Appendix E – Schedule

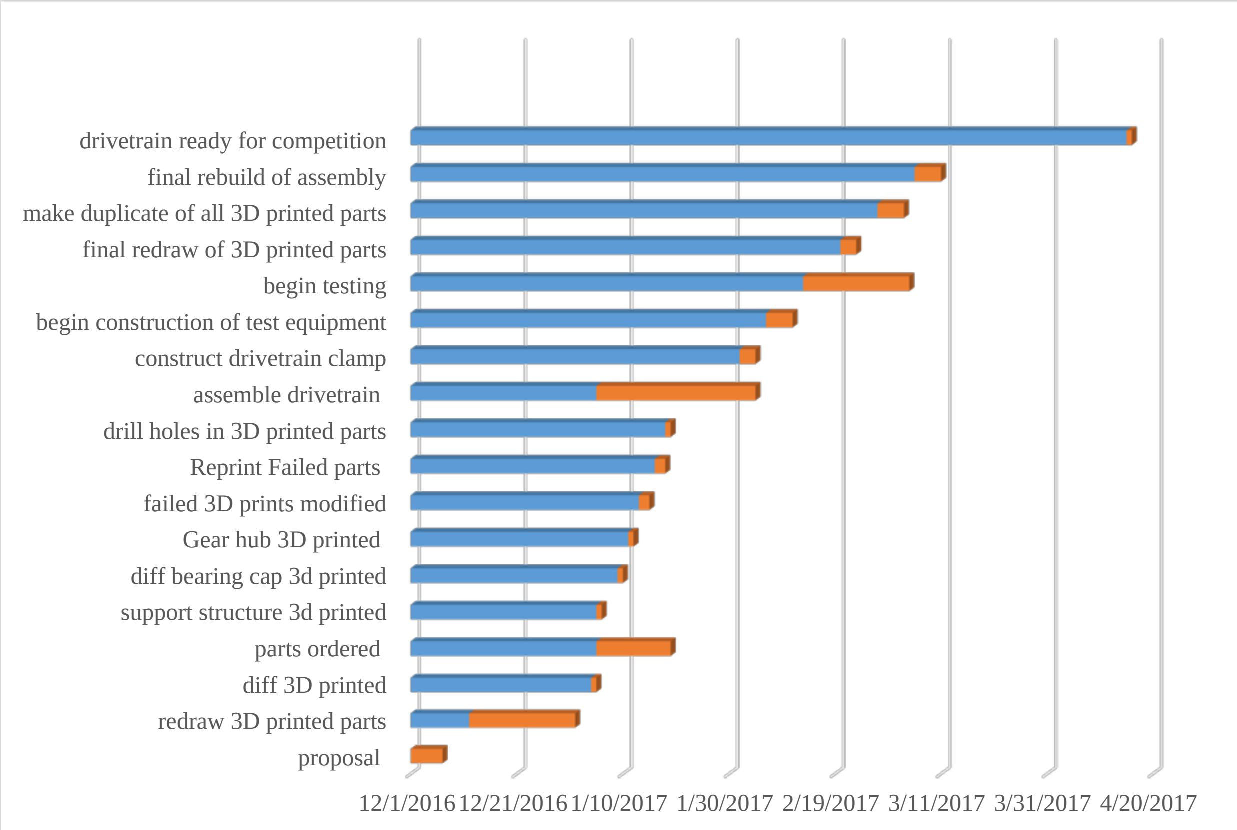


Figure 26 Gantt chart

PROJECT TITLE: ASME Baja Car Drivetrain				
Principal Investigator.: Walter Lackey				
		Duration		
TASK: ID	Description	Est. (hrs)	Actua (hrs)	November / Dec / January / February / March / April / May / June
1	Proposal*			
1a	Outline	0.3	1	
1b	Intro	2	3	
1c	Methods	2	1	
1d	Analysis	2	10	
1e	Discussion	2		
1f	Parts and Budget	2	3	
1g	Drawings	3	4	
1h	Schedule	2	1	
1i	Summary & Appx	4	2	
	subtotal:	19.3	25	
2	Analyses			
2a	engine power and torque	1	0.5	
2b	gear pitch and size	5	10	
2c	belt pitch and size	5	10	
2d	torque holding	1	2	
	subtotal:	12	22.5	
3	Documentation			
3a	diff housing 1	2	4	
3b	diff housing 2	2	4	
3c	support structure	2	10	
3d	diff bearing cap	2	2	
3e	gear hub	1	0.5	
3f	drivetrain clamp	2	1	

3g	diff assembly	1	0.5
3h	drivetrain assembly	1	0.5
3i	screw and gear representation drawings	0.5	0.5
3j	hand drawings	5	3
3k	Kinematic Check	0.5	0.5
3l	ANSIY14.5 Compl	0.5	0.5
3m	Make Object Files	0.5	0.5
	subtotal:	20	27.5
4	Proposal Mods		
4a	Project Schedule	1	1
4b	Project Part Inventory	1	3
4c	Crit Des Review*	1	1
	subtotal:	3	5
7	Part Construction		
7a	Order belt, gears, shaft, bearings	0.3	
7b	print diff housing 1 and 2	0.3	
7c	print gear hub and diff bearing cap	0.3	
7d	print support structure	0.3	
7e	make drivetrain clamp	3	
	subtotal:	4.2	0
9	Device Construct		
9a	Assemble diff.	4	0
9b	assemble shaft	4	0
9c	assemble drivetrain	20	0
9e	Take Device Pictures	0.5	0
9f	Update Website	4	0
	subtotal:	32.5	0

10	Device Evaluation			
10a	List Parameters	1	0	
10b	Design Test&Scope	5	0	
10c	Obtain resources	1	0	
10d	Make test sheets	1	0	
10e	Plan analyses	30	0	
10f	determine parameters	15	0	
10g	Test Plan*	5	0	
10h	Perform Evaluation	2	0	
10i	Take Testing Pics	2	0	
10h	Update Website	2	0	
	subtotal:	64	0	
11	495 Deliverables			
11a	Get Report Guide	0.3	0	
11b	Make Rep Outline	0.5	0	
11c	Write Report	4	0	
11d	Make Slide Outline	4	0	
11e	Create Presentation	4	0	
11f	Make CD Deliv. List	4	0	
11e	Write 495 CD parts	99	0	
11f	Update Website	99	0	
11g	Project CD*	99	0	
	subtotal:	313.8	0	
	Total Est. Hours=	468.8	80	=Total Actual Hrs
Labor$	100	46880		
Note:	Deliverables*			
	Draft Proposal			
	Analyses Mod			
	Document Mods			
	Final Proposal			
	Part Construction			
	Device Construct			
	Device Evaluation			
	495 Deliverables			

Appendix F - Expertise and Resources

Ted Bramble
Matt Burvee

Appendix G –Testing Data

Top speed test (MPH)			
Trial #	First Charge	Second Charge	Third Charge
1	30.8	30.2	27.4
2	27.5	25.1	33.8
3	29.8	33.4	33.1
4	31.7	31.5	32.5
5	32.4	30.9	33.7
	Average Top speed	30.92	
	Top Speed	33.8	

Motor Temp Test (Degrees F)	
Trial #	Temp
1	125
2	130
3	128
Average Temp	127.6666667
Highest temp	130

Static torque holding: PASSED

Diff test: PASSED

Appendix J

Walter Lackey
1002 N. Chamith Ln. Ellensburg WA, 98926
253-221-5994

OBJECTIVE: design engineering position for snowboarding or snowboard related equipment.

ENGINEERING AND DESIGN EXPERIENCE:
Bachelors of science in Mechanical Engineering Technologies.
Proficient in Rhino, AutoCAD, and Solidworks programs.
Graphic designer for Dashboards skimboard company.

ON SNOW RELATED EXPERIENCE:
12 seasons of snowboard experience.
6 seasons of snowboard instructing.
AASI level 2 certified.
Through understanding of all aspects relating to snowboard and snowboard equipment performance.
Vast knowledge of current snowboard equipment design and functionality.

EDUCATION:
Universal Technical Institute (UTI) – Sacramento, CA for automotive technology. 2010
Central Washington University- Ellensburg WA B.S. in Mechanical engineering technology. 2017

EMPLOYMENT:
11/06 – 3/07 **Crystal Mountain Snow School** Instructor's assistant
Crystal Mt, WA

11/07 – 3/08 **Crystal Mountain Snow School** Snowboard Instructor Crystal Mt, WA

11/08 – 3/09 **Crystal Mountain Snow School** Snowboard Instructor
Crystal Mt, WA

11/09 – 4/10 **Soda Springs Mt Resort** Parking Attendant
Truckee, CA

4/11 – 9/12 **Larson Mercedes Benz of Tacoma** Automotive technician Tacoma, WA

11/13-3/14 **Summit at Snoqualmie** Snowboard Instructor
Snoqualmie, WA

6/14 – 8/14 **Sandelin Fisheries** Deck Hand Valdez, AK

11/14-3/15 **Summit at Snoqualmie** Snowboard Instructor
Snoqualmie, WA

5/15 - present **Wild Water River Guides** Raft guide Peshastin, WA

REFERENCES: available upon request