Design of a Drive System for the GE Ice Testing Facility Wind Tunnel

Phase 3 Report – Final Design Report

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Prepared by: Team 21

Blair Osiowy ( ) ___________________

Kalen Friesen ( ) ___________________

Austen Karchut ( )___________________

Jan Freihammer ( )___________________

Prepared for: Paul Labossiere

Kathryn Atamanchuk

Bob Hastings

Rob Baillie

Richard Lawrence

West Canitest R&D Inc.

StandardAero

General Electric Aviation

Executive Summary

West Canitest R&D Inc., StandardAero, and General Electric Aviation are partners in the operation of the General Electric Testing, Research and Development Centre for aircraft gas turbine engine testing. At the test facility, the engine tests are conducted using a 1.2-million-pound wind tunnel. To allow for different types of tests and subject engines, the wind tunnel must be able to be moved a distance of 95 feet along 171-lb crane rail. Currently, the test facility uses forklifts to move the wind tunnel, which is unsafe for both the wind tunnel platform, the forklifts, and operators. Therefore, the test facility required a method to move the wind tunnel that is safer, more reliable, and designed for the sole purpose of moving the wind tunnel.

The design team developed several concepts for a drive system that would move the wind tunnel at a required speed of 2 m/min, overcoming obstacles such as wind load and ice buildup on the rails. Concepts were developed for five initial systems: drive system orientation, drive system input, braking system, power transfer system, and control system. Ultimately, after using quantitative and qualitative selection criteria, the final design was selected for each system and combined with the other top selections to complete the final overall design.

The overall design consists of two separate but identical drive systems. Each drive system attaches to one of the existing mounting plates and the existing wheel shaft. The input to the drive system contains a brushless, 3-phase, AC induction squirrel cage motor to provide the power needed for locomotion. The motor is capable of delivering 30 horsepower, 144 Nm of torque, and has a built-in torque overload protection system. The motor requires a frequency inverter to control the speed.

The braking system uses electromagnetic power-off brakes, so the brakes engage automatically if power is lost when the wind tunnel is in motion. These brakes are simple to maintain and operate, are reliable in extreme weather, and are able to completely immobilize the wind tunnel platform.

Power is transferred from the motor to the wheel through a four-stage gearbox. The first stage of the gearbox is a gear-and-pinion design, while the last three stages are planetary spur arrangements. The speed reduction ratio of all four gear stages combined is 790:1. To transfer the power from stage to stage, and ultimately to the wheel, splined shafts were designed to tolerate the torque applied.

The client needs were evaluated using the pre-defined project specifications. Nearly all of the metrics were achieved, while some of the project specifications can only be measured after the drive system has been implemented.

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1 Introduction

This report provides a complete description of the design and analysis of a wind tunnel drive system for the client companies West Canitest R&D Inc. (WestCaRD), StandardAero, and General Electric Aviation (GE). The wind tunnel is located at the GE Testing, Research and Development Centre (GE TRDC) in the James Armstrong Richardson International Airport grounds in Winnipeg, Manitoba, Canada. The following section of this report will give some background information on the client companies and describe the project, as well as outline the project objectives and client needs and specifications.

1.1 Company Background and Project Description

WestCaRD is the company in charge of the drive system acquisition. They are a non-profit research and development company that focuses on acquiring private and government investments to promote and support the aerospace and defence and aero-engine testing sectors [1]. GE, a partner in the operation of the test facility, is a world leading provider of commercial, military, business, and general aviation jet and turboprop engines and components [2]. StandardAero, the other test facility partner, is one of the largest independent gas turbine engine maintenance, repair, and overhaul (MRO) providers [3]. The purpose of this facility is to conduct various types of gas turbine engine certification and testing, such as ice cloud ingestion, bird ingestion, dust ingestion, hail ingestion, blade-out testing, and endurance testing.

The TRDC facility has a 1.2-million-pound wind tunnel that is required for various engine tests. The wind tunnel rests atop a large steel platform supported by four-wheel carriages. The wheel carriages contain two wheels each, which rest on 171-lb crane rail. Figure 1 provides a view of the new 11-fan wind tunnel.

[redacted]

Figure 1: GE TRDC facility new wind tunnel

The wind tunnel must be moved to provide the correct distance from the subject engine to the exit of the wind tunnel for testing, as well as to install new engines on the testing rig. The method used to move the wind tunnel currently involves pushing the tunnel with a large forklift to start the motion. Stopping the wind tunnel is accomplished with another smaller forklift that slows the momentum of the tunnel and platform. The forklift method is unsafe, unreliable, and will lead to damage to the platform if use is continued.

To eliminate the forklift locomotion method, a separate drive system was previously designed for the platform that used four hydraulic motors, but the design was not implemented due to high complexity and high cost. Therefore, WestCaRD is looking for a more reliable, simpler, and less expensive option to drive the wind tunnel platform. The premise of this project is to provide a new method of moving the wind tunnel within the constraints provided by the surroundings and the clients.

1.2 Project Detail

The following section outlines the project in detail. Provided is a list of the project objectives and a summary of the client needs. An in-depth assessment of the client needs and all other project management aspects is included in Appendix A.

1.2.1 Project Objectives

To achieve success in this project, the design team was required to achieve the project objectives. The overarching objective was to provide an overall system design that WestCaRD can easily implement. The design team could achieve this objective by meeting the deliverables of the project. The deliverables are listed below:

- A full design concept represented in engineering drawings
- CAD models of the designed components
- Finite element analysis (FEA) of critical components
- Procedural documentation
- Approximate cost of the entire drive system kept as low as possible
- List of vendors willing to supply all components

By delivering the items above to WestCaRD before December 6th, provided that the design and cost are acceptable, the project can be deemed a success.

1.2.2 Project Needs and Specifications

The project needs were items that the stakeholders required in the design. The needs were design outputs and results of the project. They also include limiting factors that were adhered to throughout the project. TABLE *I* lists prioritized project needs that were determined at the outset of the project. The “Importance” column is based on a scale from 1 to 5, where a ‘1’ represents a low priority need and a ‘5’ represents a high priority need. The priority levels were determined by the design team and approved by WestCaRD and StandardAero based on information from the project kickoff meeting:

TABLE I: Weighted Needs List

Need ID	Need	Importance	Need ID	Need	Importance
A	Drive system is operable after long periods of inactivity	4	M	Drive and braking systems operate without damage to platform or wind tunnel systems	5
B	Drive system is simple to maintain	3	N	Platform remains stationary unless being moved by drive system	5
C	Drive system propels platform using existing rail system (171# Crane Rail)	5	O	Platform can stop in an emergency	5
D	Drive system is operable in extreme weather	5	P	Drive system can be operated from control room	2
E	Drive system has long working life	3	Q	Drive system requires no more than 120/208 single/3-phase power	4
F	Drive system has control redundancies (multiple ways to stop/start)	3	R	Drive system does not obstruct working area	3
G	Drive system can start/stop at any point over the length of the track	5	S	Drive system is contained on platform	4
H	Drive system can start regardless of obstruction	5	T	Vendors will be contacted and components sourced/priced	3
I	Drive system is safe for operator	5	U	Drive system is simple to operate	4
J	Drive system is safe for bystanders	5	V	Total cost of drive system is kept to a minimum	3
K	Drive system has a form of overload protection if design torque is exceeded	5	W	Drive system is aesthetically pleasing	1
L	Drive system moves at a safe speed	5			

The ability to meet the client needs is judged using the project specifications. Project specifications are the performance metrics developed explicitly for the project that quantitatively evaluate various aspects of the design and indicate whether the design meets client expectations. The project specifications are listed in TABLE *II* along with the acceptable criterion values:

TABLE II: Project specifications list

Metric #	Metric	Units	Marginal Value	Ideal Value
1	Corrosion resistance	mm/year	1 to 3	<1
2	Time to perform annual maintenance of drive system	Minutes	<120	<60
3	Existing rail system is used	Binary	Pass	Pass
4	Drive system can perform two full passes of rail system at -40°C	Binary	Pass	Pass
5	Years of operation before major overhaul/replacement of drive system	Years	10	>20
6	Number of start and stop methods	Number	1	3
7	Minimum distance travelled	cm	30	1
8	Height/mass of obstruction that will prevent wheels from rolling	cm/kg	TBD	TBD
9	Compliance with test site safety policies for moving equipment	Binary	Pass	Pass
10	Percentage of the maximum design torque at which overload protection engages	Percentage	90%	85%
11	Minimum speed test	m/min	1	0.1
12	Maximum speed test	m/min	<5	1
13	Damage occurs during drive system initiation	Binary	Pass	Pass
14	Damage occurs during braking system initiation	Binary	Pass	Pass
15	Static brake system is operational	Binary	Pass	Pass
16	Distance travelled between emergency stop activation and platform coming to a stop	m	<1	0.5
17	Drive system power requirements do not exceed available power	Binary	Pass	Pass
18	Footprint in excess of original platform	m^2	<5	0
19	Percentage of total parts that are sourced with vendor confirmation	Percentage	>50	100
20	Operator training course grade	Percentage	>80	100
21	Time to explain drive system to stakeholders	Minutes	<120	60
22	Quoted costs	$	<original design	TBD
23	Stakeholder opinion	subj	3 out of 5	5 out of 5

Once the overall design is discussed, the metrics from TABLE *II* can be evaluated.

2 Design Overview

The final design is presented in Figure *2*: Overview of the Drive system with the major components labeled.illustrating the complete drive system with the major components labeled.

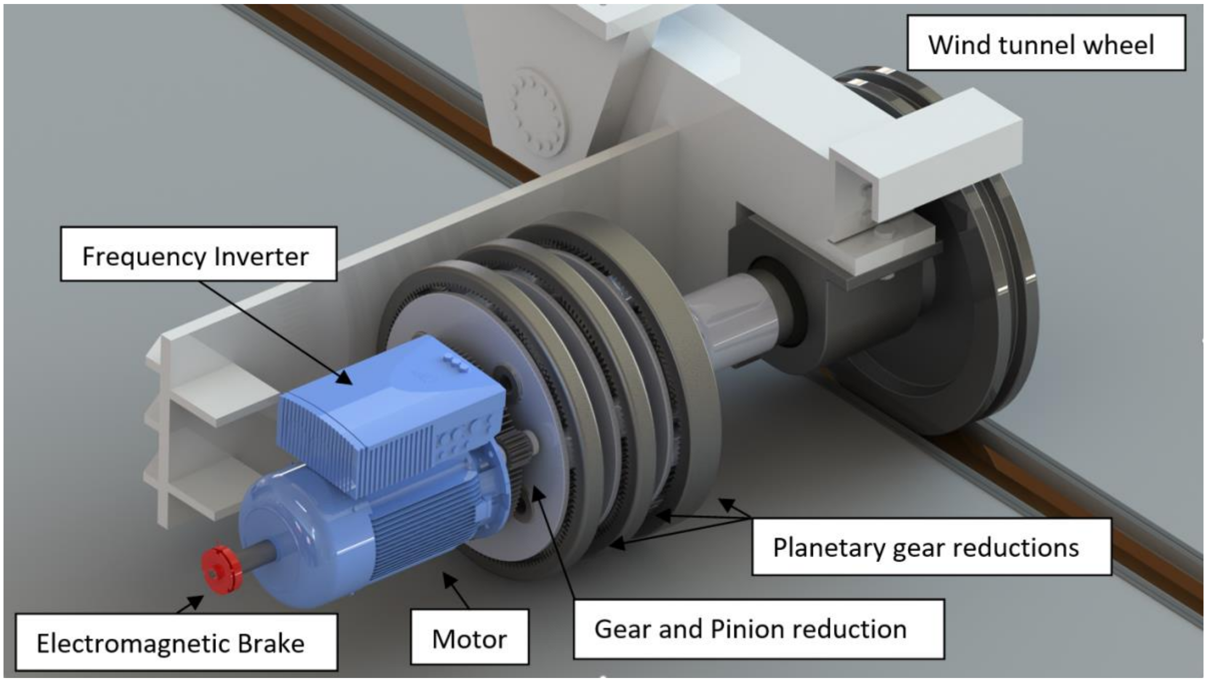


Figure 2: Overview of the Drive system with the major components labeled.

3 Detailed Design

To develop a successful solution to move the wind tunnel, five major design decisions needed to be made. They are as follows:

1. A method to power the drive system.
2. A method of delivering the power to the wheels of the wind tunnel platform.
3. A method to stop the motion of the wind tunnel platform.
4. A method of controlling the movement of wind tunnel platform.
5. An orientation of all the components of the drive system.

The design team developed several concepts for each of the five design criteria above. Details of the concept development process can be found in Appendix B. The complete final drive system design is described in the following sections.

3.1 Drive System Orientation

In the concept selection phase of the project, the design team considered several drive system arrangement options. For the final design, the orientation selected was a two-wheel drive system using two independent and identical setups. The two-wheel drive orientation was selected for several reasons. Firstly, two setups are more cost-effective than the unimplemented four-wheel drive system. Secondly, the wind tunnel platform is not stressed in a significant way because the drive systems propel both sides of the platform equally. Finally, the two-wheel drive system eliminates the need for excessively long shafts that accompany a mitre-box style drive system, where one drive setup resides in the middle of the wind tunnel platform, equidistant from the two rails. A complete orientation selection process is outlined in Appendix B.

3.2 Motor

This section will discuss the final motor selection and the additional motor features that are required to accommodate for braking and controls. The forces considered for the calculations of the motor will also be explained.

3.2.1 Concept Selection

The concept selection procedure described in Appendix B gives a detailed explanation of the different motors that were considered and the reason for the specified scores in TABLE *III* below:

TABLE III: Drive System Input Concept Scoring Matrix

		Concepts							
		DC Brushed		DC Brushless		AC Synchronous		AC Induction	
Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Cold weather operation	0.19	5	0.972222222	5	0.972222222	5	0.972222222	5	0.972222222
Operable after 6-month idle time	0.19	5	0.972222222	5	0.972222222	5	0.972222222	5	0.972222222
Noise	0.01	2	0.02	5	0.05	5	0.05	5	0.05
Potential Torque	0.19	4	0.777777778	5	0.972222222	3	0.583333333	4	0.777777778
External Components needed	0.03	5	0.138888889	4	0.111111111	3	0.083333333	3	0.083333333
Ease of maintenance	0.14	3	0.416666667	5	0.694444444	4	0.555555556	5	0.694444444
Cost	0.11	5	0.555555556	3	0.333333333	3	0.333333333	4	0.444444444
Operational at Variable Speeds	0.06	3	0.166666667	5	0.277777778	5	0.277777778	5	0.277777778
Simplicity of design	0.08	4	0.333333333	5	0.416666667	4	0.333333333	4	0.333333333
	Total Weighted Score	4.353333333		4.8		4.161111111		4.605555556	
	Rank	3		1		4		2	
	Develop?	No		Yes		Yes		Yes	

The DC brushless motor was initially assumed to be the better option in reference to the listed criteria in TABLE *III*. After further investigation, the DC brushless motor was not able to meet the specified motor speed of 550rpm that was needed. The voltage applied to the DC motor would not be able to be altered in an cost-effective manner. The reason the voltage would need to be lowered is because DC brushless motors off the shelf have standard speeds around 1300rpm. The AC motor does have the ability to be used with a variable frequency drive to reduce the speed of the motor to meet the 550rpm goal. The power supply that the GE TRDC has available is AC power, so there would need to be external components to convert that AC power into DC power in order for the DC motor to function.

The initial concern of the AC induction motor being able to meet the high required torque value was disregarded after determining that AC motors with double the necessary power required for this drive system are readily available from many manufacturers. AC induction motors have grown in popularity in recent years, so their cost is lower than the DC brushless motor. The AC motor does have a torque overload system built in, so when the design torque of the motor is exceeded, it will shut off to avoid any damage to the motor and the gearbox. On the contrary, the DC motor does not have this advantage and would keep trying to rotate even if it could not handle the extra load. This persistence could lead to failure of any of the components on the drive system.

The AC induction squirrel cage motor has been selected as the motor that will be used for this drive system.

3.2.2 Motor Specifications

The first step to designing a drive system is to determine the amount of torque required for the motor to output to move the wind tunnel. Initial calculations to determine an assumed wheel rotational speed are discussed in this section. The wheel speed is based on parameters that were chosen by the team with consideration of the clients needs.

3.2.2.1 Travel Time and Gear Ratio

The tread surfaces of the wheels supporting the platform are 36 inches in diameter. The diameter of the wheel can be used to determine how many revolutions of the wheel are required to move the platform from one end to the other, a distance that measures 95 feet in length. The number of wheel revolutions is critical in determining the overall gear ratio (OAGR) and the rotational velocities of the motor.

The clients require that the platform move at one speed. The metrics determined in Phase I of the project include an ideal travelling speed of 1m/min, or 3.281ft/min. However, an acceptable travelling speed is anything less than 5m/min, so a travelling speed of 3m/min, or 6.562ft/min, was used to determine the amount of time it will take the platform to move the 95-foot distance. The motor speed was chosen to be 550rpm to achieve the required traveling speed. The travelling time can be used to determine the wheel RPM and the OAGR. The detailed equation can be seen in Appendix C.

TABLE IV: Initial calculation results

Initial Specifications	
Revolutions in 95 feet	10.080 rev
Travel Time	14.48 min
Wheel RPM	0.696 RPM
OAGR (Gear Ratio)	789.9:1

The travelling speed will be used to calculate the motor torque, also the total tractive force needs to be found to determine the motor torque.

3.2.2.2 General Forces on the Wheel

This section explains all the general forces that the motor needs to overcome, such as:

- Starting Resistance
- Grade Resistance
- Acceleration Force

The average wheel load of the structure was found using load ratios that were provided by the client of the old wind tunnel, which is shown in Figure *3*:

[redacted]

Figure 3: Loads at wheel carriage locations on the wind tunnel platform

The back of the wind tunnel is where the wheels see the largest load and these will be used as the two drive locations. The larger load on those two wheel locations will result in a larger available tractive force. The error associated with the load at those locations was added to the total load. The average load was taken from location 3 and 4 and was assumed to be the load per wheel for the entire structure. Since each location has two wheels, the load was adjusted accordingly. Since the loads in Figure *3* are from the old wind tunnel (800,000 lbs), the values need to be scaled higher to account for the heavier weight of the new wind tunnel structure (1.2 million lbs). The equations to find the load ratios of each location of the old structure, new structure is found in Appendix C.

TABLE *V* shows the new loads for each wheel carriage location. The new total loads include the 120% allowance associated with each value.

TABLE V: Total loads at each location, load errors included

Total Location Loads		
#1	273.2278481 kips	273227.848 lbs
#2	259.0379747 kips	259037.975 lbs
#3	395.556962 kips	395556.962 lbs
#4	365.1772152 kips	365177.2152 lbs

The starting resistance is approximately 2-2.5 times higher than the rolling resistance [4]. The friction coefficient of railroad steel wheels on steel rails was found to be approximately 0.00164ft [5], which was used in the calculation to be more precise instead of a value with no units.

Additional tractive force effort would have been required if the ground was at an incline. The TRDC facility was constructed to have a perfectly flat foundation where the wind tunnel is situated, as specified by the client, so no incline was accounted for.

The acceleration force is an additional parameter that determines the amount of force required to reach the desired speed in a certain amount of time. The following calculation assumes the wind tunnel structure starts from rest. An acceleration time of 15 sec was assumed to be a safe and reasonable time. This value will vary since the motor is set to accelerate at a certain rate, but that was assumed to be negligible, since the acceleration force does not have a notable effect on the resistance the motor must overcome.

TABLE *VI* summarizes all the general forces on the wheel:

TABLE VI: General forces

General forces	
Average Load per Wheel	190 183.54 lbs
Rolling resistance per Wheel	207.99 lbs
Starting Resistance per Wheel	519.97 lbs
Grade Resistance	0 lbs
Acceleration Force	271.72 lbs

A more detailed analysis showing each of the calculations in this section can be found in Appendix C.

3.2.2.3 Wind Forces

The wind tunnel and platform have a large footprint and are approximately nine metres tall, so the wind force plays a role in determining the required motor torque.

The wind pressure was found according to the National Building Code of Canada in 2015 [6]. An average wind speed of 44.8km/h was chosen to be the value that the drive system could withstand (from the appendix C of the NBCC 2015 [6]). The highest wind speed that happens a few times a year was 60km/h and the lowest average wind speed was 25km/h [7]. The average was taken between those two results. The facility is in an open terrain environment, so the wind speed was assumed to be higher than the lowest average according to NBCC.

The projected area of the wind tunnel (dimensions of most unshielded side estimated from drawings) was found to be 143.78m^2.

TABLE *VII* below shows all the factors and the respective values present in the wind pressure calculation, as well as the total force caused by the wind:

TABLE VII: Factors used in Wind Pressure calculation.

Factor	Value
Importance Factor (Normal)	1
Reference Velocity Pressure	0.45 kPa
Wind Speed	44.8km/h
Exposure Factor (Open Terrain)	0.98
Gust Effect	2.0
External Pressure Coefficient	1
Total Pressure	0.44125 kPa
Total Wind Force	13292.66 lbf

*Equations for all calculations are found in Appendix C

3.2.2.4 Ice Forces

To acquire an approximation of the forces that would need to be overcome due to ice buildup on the tracks, several assumptions were made. The first was that the platform would only overcome the ice by breaking it, or the wind tunnel would not move. The overall weight of the structure was assumed to be too large for the wheels to drive over top of the ice. These assumptions mean that the ability to overcome the ice would be dependent on the compressive strength of the ice.

Other assumptions made were related to wheel dimensions, the actual compressive strength of the ice, the amount of ice, the number of wheels in contact with ice, and the percent of contact the ice had with the rail wheel. Due to so many variables being present, a medium and worst-case scenario were considered. The medium-case scenario assumes that all ice above the surface of the rail has been manually removed with only partial coverage below the wheel and ice with a moderate compressive strength. The worst-case scenario assumes one inch of ice on top of the rail, as well as increased coverage and ice that has a higher compressive strength, which is true for colder temperatures [8].

To approximate the surface area of ice that would be blocking the wheel, the wheel dimensions were assumed as shown below in Figure *4*, where "Y" is the height of ice above the rail and "Φ" is the angle of coverage. The radius of the wheel is known (r=18 inches) so that did not need to be assumed for these calculations.

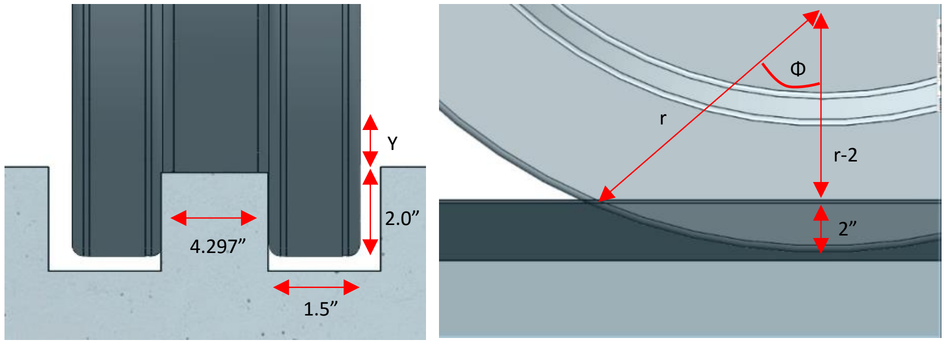


Figure 4: Assumed Wheel Dimensions

The surface area of ice was split into two sections: below the rail and above the rail. The angle of affected area above and below the rail were determined using trigonometry, then those sections were multiplied by the total circumference to determine the total length of circumference blocked by ice. The area above the rail also included the middle of the wheel, although this was assumed to be the same radius as the outer sections of the rail to simplify calculations.

To determine the required force to break the ice, an assumed compressive strength and coverage percent was used. For the medium-case scenario, approximately 1,000 psi was assumed, which was a common breaking strength found for ice at approximately -10°C, with 50% coverage [8]. For the worst-case scenario, a breaking strength of 2,000 psi was assumed, which represents ice that is around -25°C, with 75% coverage.

The results of the calculations from TABLE *VIII* are estimations and may vary greatly with coverage and the compressive strength of the ice. Compressive strength can reach 4,300 psi at -40°C [8]. The details and calculations of for the results presented in TABLE *VIII* can be found in Appendix C.

TABLE VIII: Summary of factors pertaining to ice force.

Factor	Value
Angle affected area below rail	27.3 degrees
Angle affected area above rail	6.3 degrees
Surface Area Medium Case	25.70 in^2
Surface Area Worst Case	40.12 in^2
Ice Forces Medium Case	12849 lbf
Ice Forces Worst Case	65453 lbf

Since ice has such a detrimental effect on the available tractive force and the resistive force, assumptions had to be made after consulting with the client for appropriate levels of ice that the drive system could overcome. Ice on top of the track was assumed to be only 0.1 inches and coverage would be only 20 percent on the top and sides of the rail, as it can be manually removed. Additionally, a breaking strength of 1500 psi was used as it reflected the strength of ice at a more realistic temperature of between -15°C to -20°C. Using the same equations as above with the new ice coverage values, there is a required breaking strength of 8181 lbf per wheel, or a total of 65453 lbf of resistive ice. As this is well below the worst-case conditions for ice on the rails, it is very important to keep the track clear if the drive system will be used in winter.

The ice force was based on the available tractive force and dry rail conditions. Anything above those forces will result in wheel slip or the motor won't have enough power.

3.2.2.5 Required Tractive Force and Motor Torque

The total tractive force required was determined by adding all the resistances and forces on the wind tunnel structure. The starting resistance was multiplied by eight because there is a total of eight wheels contributing to the friction force. The total tractive force was then multiplied by a friction resistant factor of 1.15, which takes into account bearings and axle friction, as well as other unspecified mechanical resistances.

There are two driving wheels to move the structure, so the tractive force required per wheel is equally distributed between those two driving wheels. The motor torque was found by using the overall gear ratio and gear transfer efficiency, which is found in Appendix C. A summary of the values used in determining the required motor horsepower is presented in TABLE *IX*, detailed equations and calculations for these values can be found in Appendix C.

TABLE IX: Variables determined in the process of calculating the required motor horsepower based on tractive force required.

Variable	Value
Total Tractive Force	94503.69 lbf
Total Tractive force per wheel	47251.85 lbf
Required Torque at each Wheel	70877.88 lbf
Required Motor Torque	97.28 ft lbs
Required Motor Horsepower	10.19 Hp

The final drive system motor must have the required power found in TABLE *IX*. The total tractive force in TABLE *IX* is based on all the forces that the wind tunnel will experience. To ensure that the drive system can propel the wind tunnel, the total available tractive force must be equal to or greater than the required tractive effort.

3.2.2.6 Available Tractive Force and Motor Torque

The available tractive effort is based on the dead load of the entire structure. The dead load was broken down to determine the dead load on each wheel. The adhesion coefficient is a factor that will constantly change based on the weather conditions. The introduction of water, snow, or ice on the rail system will drastically change the adhesion coefficient, which will greatly reduce the available tractive force and thus, the potential of wheel slip is present. TABLE *X* shows three adhesion coefficients that are assumed values based on rail conditions [9].

TABLE X: Adhesion Coefficients based on rail conditions

Rail Conditions	Adhesion Coefficient
Dry Conditions	0.3-0.5
Wet	0.2
Ice	0.05-0.1

An adhesion coefficient of 0.3 was taken as the value used to determine the available tractive force in dry conditions. The client specified that the wind tunnel would be moved mostly in fall and spring and rarely during winter. Also, ice would be removed as completely as possible, so the conditions will be mostly dry to wet in the worst of cases.

The available tractive force represents the force past which the wheels will slip, which will prevent the wind tunnel from moving and possibly cause damage. In comparison with the required tractive force, the available tractive force per wheel is higher; this means with the current assumed conditions, the wind tunnel will be able to be moved safely without any wheel slip. The available tractive force result is the maximum amount of traction a wheel will have with the current weight of the wind tunnel. The motor that will be purchased will be based on the horsepower and torque required for 57055.06lbs of tractive force. TABLE *XI* summarizes the values calculated in determining the required horsepower of the motor based on the available tractive force.

TABLE XI: Variables determined in the process of calculating the required motor horsepower based on tractive force available.

Variable	Value
Available Tractive force per wheel	57055.06 lbs
Required Torque per wheel	85582.595 lbs
Required torque at motor	117.4583 ft lbs
Required Horsepower at motor	12.3 Hp

3.2.3 Nord Inc Motors

NORD Drive Systems is a North American company that offers a variety of asynchronous induction motors that can be used for many applications. They have many options for motor add-ons and have variable frequency drives that function with each of their AC motors [10].

The AC motor will be a 3-phase design because the single-phase motors produce a maximum of two horsepower, which does not meet the need of 12.3HP. The 3-phase motor offers a range of 0.75HP to 75HP and it gives the ability to have the motor also function in reverse, which is a necessary feature for the drive system.

The option for motor frequency is either 50Hz or 60Hz. The 50Hz range offers the ability for the motor to function at a range of voltages, whereas the 60Hz has a specific working voltage for the motor. The 60Hz frequency results in a lower max torque output for the motor at the same horsepower rating as the 50Hz motor. A frequency of 60Hz is the most common frequency used across North America, so both frequencies could be used.

There are three different efficiency classes specified by NORD drive systems: IE1, which is standard efficiency, IE2, which is high efficiency, and IE3, which is premium efficiency. The differences in the efficiencies are the power factors associated with each class of efficiency. The high and premium efficiencies offer similar power factors of 0.86 and 0.87, respectively. This small difference does not justify the extra cost of the premium efficiency motor. The standard efficiency has a power factor of 0.81. Having a lower power factor will increase the current in the power lines and will also increase electricity cost. The standard efficiency motor will cost less and weighs only 122kg compared to the 155kg high efficiency motor. To reduce cost, the standard efficiency was deemed sufficient for the drive system since it is only being used 2-3 times a year.

The maximum tractive force that is available is 57055.06lbs per wheel. This force requires 12.3HP to be generated by the motor. The motor selection was based on these values. An AC motor has one operating speed, and to change that output, a variable frequency drive system needs to be implemented. Typical off the shelf motors do not have a low operating speed combined with a high torque and high horsepower. The only option was to add the VFD as an external system. Also, a motor cannot output 159Nm of torque and only output 12.3HP, so a larger horsepower motor was required to achieve the required torque output of 159Nm.

Since a VFD was used to lower the frequency of the motor to the frequency for which the gear train was designed, the effects on horsepower and torque had to be considered. When lowering the rpm, the horsepower also drops at the same rate as the rpm, as shown in Figure 5. This consideration was taken into account when choosing a motor:

Figure 5: HP and torque change with motor speed for a rated 50 hz motor.

The forces acting on the wind tunnel show that with a 44.5km/h wind and 20% ice coverage on the bottom portion of the face of the wheel, the horsepower needed is around 10.19hp.

TABLE XII: Specifications for the selected Motor.

Motor specifications	
Model	IEC SK 180LX
Power	22kW
Torque	144Nm
Efficiency	90.3%
Speed	1460 rpm
Power factor	0.81
Voltage	380-420/655-725 V

The voltage provided by the GE TRDC is only up to 208V. The VFD is only capable of being used at the 300V level, so the motor selected was the higher voltage. This is further discussed in the next section of the report.

The brake will be attached to the opposite side of the motor, so a second shaft must be added to the motor to accommodate for braking; this is the most efficient way to incorporate braking. When the motor speed is decreased, more heat is generated, so the motor will need additional cooling. There is an option to add an additional fan with an enclosure to further cool the motor.

The weather conditions in the fall, winter and spring possess multiple different humidity, temperature and moisture conditions. The motor will have a IP66 enclosure to combat water and dust.

3.2.4 Frequency Inverter and Control System

The control system selected was a NORDAC FLEX SK200E AC vector drive with a handheld “Simplebox” for control and parametrization. This frequency inverter can be mounted directly to the motor, or it can be wall mounted nearby. The ambient temperature range for the operation of this device is listed as -25 to 50°C, so the wall mounting in a slightly protected area should be considered.

The frequency inverter offers more possibilities as to the speed that the drive will move at, effectively making it a variable speed drive as opposed to a single speed drive. This system is also offered by NORD which simplifies compatibility issues with the motor and makes use of only one supplier for two major components. Unfortunately, the size of the motor (30HP due to required torque) meant that a model operating at 380-500V (3-phase) had to be selected. This selection will require the addition of a transformer to operate the drive from a 208V (3-phase) power source.

The SK200E model includes a built-in 24V power supply, a built-in Incremental encoder input for SERVO-MODE, an absolute encoder input and a PLC (programmable logic controller) functionality.

Additionally, a safe stop function can be added, which may be beneficial to prevent the motor from being operated at a torque higher than it can handle.

The inverter comes with two types of enclosure ratings, IP55 and IP66. Due to the rough conditions that the entire drive system will encounter, a protection of IP66 is recommended because the dust and small particles that might be present around the facility. However, with some sort of alternate shielding, IP55 should be sufficient, as it still protects against water from all directions. More information on the IP rating can be found in Appendix C. The handheld device allows for simple setting of values as well as a switchover to parameter mode.

3.3 Braking

This section will outline the final recommendation for braking for the wind tunnel and platform, including the final braking requirement calculations, design considerations, as well as an estimated cost.

3.3.1 Concept Selection

Throughout the concept design phase of this project, multiple viable braking options were considered, including hydraulic and pneumatic braking, linear rail braking, electromagnetic braking, and plugging motor braking. After screening and weighing each braking system against the clients' requirements, electromagnetic braking was determined to be the most viable option and pursued further. Electromagnetic brakes most accurately met the most important requirements: simple design, operable in extreme conditions, and exhibiting multiple functionality as a primary, emergency, and stationary brake. The full extent of the concept screening and evaluation can be found in Appendix B.

3.3.2 Braking Requirements

The total braking force requirement for the system was initially determined from the deceleration required to stop the platform and wind tunnel from a top speed of 1.93 meters/minute (a 15-minute travel time for the max distance of 95 feet) over one foot. The calculation was performed for a braking system placed at the output shaft right before the wheel. The total torque required to stop the system was calculated to be approximately 1000 lb-ft.

While sourcing the brakes from suppliers, it was recommended to mount the brake off the rear of the motor, as this produced several advantages: rear-motor mounting would substantially decrease the amount of required torque due to the large gear reduction, thus decreasing the cost of the brakes. Rear-motor brake mounting would also help remove the limitation of the output shaft size. The output shaft currently has large amounts of drive torque, again due to the gear reduction, thus forcing it to be a large diameter. The only brakes that were available with that bore size were very costly and had over double the necessary torque for the system.

Calculating the required braking torque at the motor was done using total inertial loads instead of the required system deceleration, as the brake is responsible for slowing the inertia of more systems. The inertia of the motor and gearbox would also have to be stopped, as well as the platform, although at an incredibly reduced amount due to the gear reduction. The inertial loads of the gear reduction and motor were obtained from appropriate supplier data sheets to acquire some level of accuracy, while the equivalent inertia of the platform was calculated using the reduction of horizontal speed to brake rpm. Using the initial assumptions for platform speed and a motor speed of 500rpm, the total inertial load was found to be 23.97lb-ft^2. To determine the braking torque required, the stopping time was first needed. From the initial assumptions for speed and stopping distance, a time of 9.5 seconds was found, and from this time, the required static torque was determined to be 10.24 lb-ft with a safety factor of two.

A complete summary of the calculations used to determine the braking force and torque requirements can be found in Appendix C.

3.3.3 Design Considerations and Product Availability

A few items had to be considered before electromagnetic braking could be selected as the final recommendation for the braking system. The first consideration involved the braking mounting options, as electromagnetic braking typically mounts inline with the shaft. Braking units are usually positioned nearest to the moving element to reduce stress on the rest of the system, therefore a mounting position at the output shaft was chosen, off the gearbox. The output torque is very high in the output shaft due to the high weight and low required speed, which forced the output shaft to be quite large. Electromagnetic braking stock bore sizes typically range from 1.0 to 5.5 inches, so a material that could handle the appropriate torque at that maximum size shaft would have to be chosen to fit this design constraint. As previously mentioned, this mounting option was substituted for a brake mounted on the motor. While rear-motor brake mounting does put the braking stress through the motor and gear reduction, it is preferred in industry applications due to the ability of amplifying the effective torque through the gearbox, greatly reducing the cost. The brake rests on a shaft diameter of 1.875in, while also having room to install a reduction if necessary, vastly increasing the number of suppliers available.

Ensuring the appropriate braking torque required was available in an off-the-shelf product was also critical. Typical torques for electromagnetic brakes range from 0.2 to 4000 lb-ft, exceeding the required braking force substantially per braking unit.

After sourcing, an applicable unit manufactured by SEPAC Incorporated was found. They sell two power-off (engage when power is lost) series of brakes that fit the constraints of the designed drive system: the TSEB and SEB models. Both are frictional disk brakes, while the TSEB is simply a more compact version of the two.

The final braking system chosen was the TSEB friction electromagnetic brake made by SEPAC industries. The model number 460 comes with a 0.748in bore diameter and a static torque rating of 11.80 lb-ft. It has a recommended max rotational speed of 3500rpm, a 4.61in diameter, and 1.351in length [11]. Unfortunately, the bore size is substantially smaller than the output shaft on the side of the recommended motor option. To compensate for this, the motor shaft would need to be machined down or a reduction created to accommodate the brake. Since the motor shaft is approximately 1.875in, the brake that would have to be purchased to fit without modifying the shaft would produce 250 ft-lbs of torque, making it far larger and more costly than necessary [12].

The TSEB 460 system was quoted at 492.78 USD per unit with a 4-6-week lead time.

3.4 Gearbox

The design process of the gearbox began with specifying the gears using a force and stress analysis. Once the gears were specified, the shafts were designed. This section of the report outlines the final concept selection for the gears, discusses the final gear arrangement, and provides details on the shafts that support the gears and transfer the power from the motors to the drive wheels.

3.4.1 Concept Selection

Following the concept selection process, the optimal speed reduction solution to proceed to the design stage was a gear-and-pinion-style gearbox. Typically, the maximum gear ratio of a gear-and-pinion is 10:1 [13]. Therefore, for the overall gear ratio of 789.9:1 (which was determined as part of the motor calculations), a minimum of three reduction stages were required. However, if three reduction stages were used, the gear ratio of each stage would have to be 9.24:1 to achieve the 789.9:1 overall gear ratio. Since 9.24:1 is approaching the maximum attainable gear ratio of a gear-and-pinion, the gears would be large and may not fit within the design space. Consequently, a four-stage reduction system was selected as the final arrangement. With four stages, the gear ratio per stage could be 5.30:1 on average, which allows the gears to be more reasonably sized. Figure *6* shows a schematic of the four-stage gear-and-pinion-style gearbox:

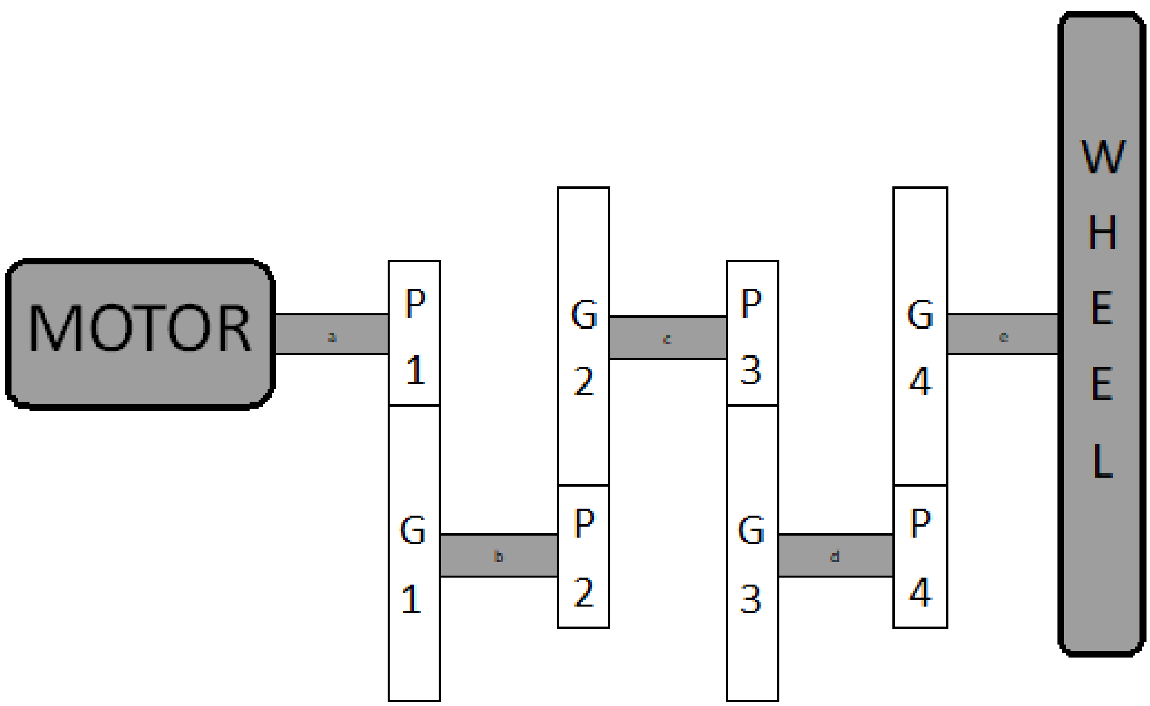


Figure 6: Four-stage gear-and-pinion-style reduction, showing general configuration between motor and wheel

The four-stage gear-and-pinion-style design shown in Figure *6* was selected to proceed to the next stage of design, which was specifying the gears using a force and stress analysis. After determining the gear ratio per stage, the calculations for determining the stresses in the teeth were performed.

Once the preliminary calculations for the four-stage gear-and-pinion-style reduction system were completed, the conclusion was that the forces on the pinion teeth in all stages but the first exceeded the bending strength and contact strength of any common material. A summary of the of the preliminary analysis is shown in TABLE *XIII* and TABLE *XIV*:

TABLE XIII: Four-stage gear-and-pinion design variables

Design Item	Value	Unit
Assumed Input Power	12	HP
Diametral Pitch	8	teeth/in
Pressure Angle	20	degrees
Teeth on Pinion	16	teeth
Teeth on Gear	85	teeth
Diameter of Pinion	2	in
Diameter of Gear	10.625	in
Input shaft velocity (Shaft a)	550	RPM
Intermediate shaft velocity (Shaft b)	103.53	RPM
Intermediate shaft velocity (Shaft c)	19.49	RPM
Intermediate shaft velocity (Shaft d)	3.67	RPM
Output shaft velocity (Shaft e)	0.69	RPM

TABLE XIV: Four-stage gear-and-pinion load and stress results

	Gear Pair 1	Gear Pair 2	Gear Pair 3	Gear Pair 4
Pitch-line Velocity [ft/min]	287.98	54.21	10.20	1.92
Transmitted Load [lbf]	1077	5720	30388	161436
Pinion Bending Stress [psi]	56661	263913	1315738	6790084
Contact Stress [psi]	193750	418148	933651	2120985

The contact stress and bending stress values for the most critical component (pinion 4) in TABLE *XIV* are on the scale of 10^6, while the allowable contact strength and bending strength of any reasonable material is on the scale of 10^5 [13]. The disparity of a factor of ten indicated that a gear-and-pinion design would not be able to tolerate the large torques and stresses that occur with a two-wheel-drive system. Therefore, a design alteration was required to reduce the forces acting on the gear teeth.

In the concept selection process, a planetary gear system was one of the design considerations. With the spur gear-and-pinion and helical gear-and-pinion systems effectively ruled inadequate due to the high tooth stresses, the use of a planetary gear system was the next most feasible design. A planetary gear system using multiple planets distributes the forces among several gears, thus reducing the stresses. Therefore, the design process proceeded with specifying the gearbox using a planetary gear arrangement.

3.4.2 Gear Specification

The final gearbox design consists of four stages: the first stage is a spur gear-and-pinion reduction and the final three stages are spur gear planetary reductions. Figure *7* shows a rendering of the planetary gear arrangement:

Figure 7: Rendering of gearbox stages 2 to 4

The first stage of the gearbox was allowed to remain a gear-and-pinion design as originally planned since the stresses were acceptable. The final three stages required a planetary gear system, identical to the one shown in Figure *7*. For a complete analysis of the gearbox final design, see Appendix C.

TABLE *XV* and TABLE *XVI* show the input variable and results of the force and stress analysis for the final gear design:

TABLE XV: Final gear design input variables

Design Item	Value	Unit
Assumed Input Power	12	HP
Diametral Pitch	5	teeth/in
Pressure Angle	20	degrees

TABLE XVI: Final gear design analysis results

	Stage 1	Stage 2	Stage 3	Stage 4	Note
Reduction Type	Gear and pinion	Planetary	Planetary	Planetary	
Number of Planets	N/A	4	4	4	
Input Speed (Sun Gear) [RPM]	581.72	94.98	18.80	3.72	
Output Speed (Carrier) [RPM]	94.98	18.80	3.72	0.74	
Gear Ratio	6.125:1	5.052:1	5.052:1	5.052:1	
Pinion Diameter [in]	3.2	N/A	N/A	N/A	
Gear Dameter [in]	19.6	N/A	N/A	N/A	
Pinion Teeth	16	N/A	N/A	N/A	
Gear Teeth	98	N/A	N/A	N/A	
Sun Diameter [in]	N/A	7	7	7	
Planet Diameter [in]	N/A	10.8	10.8	10.8	
Ring Diameter [in]	N/A	28.6	28.6	28.6	
Sun Teeth	N/A	35	35	35	
Planet Teeth	N/A	54	54	54	
Ring Teeth	N/A	143	143	143	
Pitch-line Velocity [ft/min]	487.34	174.05	34.45	6.82	Sun-planet for stages 2, 3,, 4
Transmitted Load [lbf]	812.6	568.8	2873.8	14519.5	Sun to one planet for stages 2, 3,, 4
Face Width [in]	2.5	2.5	2.5	4	
Pinion/Sun Tooth Contact Stress [psi]	111050	63164	123075	196103	
Pinion/Sun Tooth Bending Stress [psi]	17686	7953	33061	94722	
Material Selection	Grade 1 through-hardened steel	Grade 1 through-hardened steel	Grade 2 through-hardened steel	Grade 3 carburized and hardened steel	
Allowable Contact Stress	101550	101550	156450	275000	
Allowable Bending Stress	30193	30193	52100	75000	
Hardness Value [Brinell]	225	225	350	See note	See Table 9 of ANSI/AGMA 2001-D04
Factor of Safety for Wear	1.06	1.91	1.66	2.06	
Factor of Safety for Bending	2.22	4.94	2.21	1.35	

As shown in TABLE *XVI*, each stage requires a different material or hardness. Also of note, the ring gears are large with a 28.6-inch pitch diameter, which was a requirement of the system due to the exceptionally high torque that is required in a two-wheel-drive system.

3.4.3 Shaft Specification

The following shafts were designed based on the gear designs and the DE-ASME elliptic failure criterion to determine the minimum shaft diameters at the critical locations. Equation (1) shows the rearranged version of the DE-ASME equation to find the diameter [13]. Additional information on shaft calculations can be found in Appendix C and drawings of the shafts can be found in Appendix D.

$$d = \left\{ \frac{16n}{\pi} \left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} \quad (1)$$

Where:

n is the factor of safety

K_f is the fatigue stress concentration factor

K_{fs} is the fatigue concentration factor due to shear

M_a is the alternating bending stress

M_m is the midrange bending stress

T_a is the alternating torque

T_m is the midrange torque

S_e is the fatigue stress

S_y is the yield tensile stress

The diameters of the shafts were based off the minimum acceptable diameter for the selected materials and the available sizes. Fillet radii (r) are 10% of the smaller diameter. Some adjustments were required to accommodate bearing size and the size of the input and output shafts. All gears are attached to the shafts via splines and located with retaining rings 0.05in wide.

3.4.3.1 *Shaft a*

Shaft a, as seen in Figure *8*, connects the output of the motor to the gearbox. This shaft uses AISI 4130 steel normalized to 1600°F

D_1 is diameter of the shaft section that is splined onto the motor output and has a value of 2.76in and a length of 4.5in.

D_2 is the diameter of the section holding the stage 1 pinion, measuring 2in. This section has a length of 2.52in.

D_3 is the diameter of the section of shaft holding the bearing, measuring 1.25in.

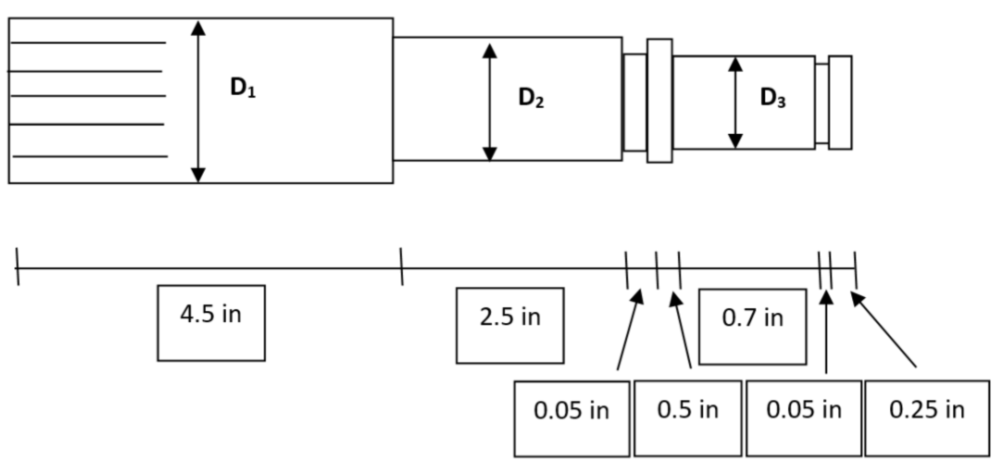


Figure 8: Shaft a

3.4.3.2 *Shaft b*

Shaft b, as seen in Figure *9*, connects the stage 1 gear to the sun gear of the first planetary gear set (stage 2). This shaft uses AISI 4130 steel normalized to 1600°F.

D_1 is the diameter of the section of shaft holding the bearings, measuring 1.25in. The length of this section is 2in.

D_2 is the diameter of the sections holding the stage 1 gear on one side and the stage 2 sun on the other, measuring 1.5in. This section has a length of 2.52in on both sides.

D_3 is the diameter of the midsection of shaft locating the gears on either side (using shoulders), measuring 2in. This section is 0.5in long:

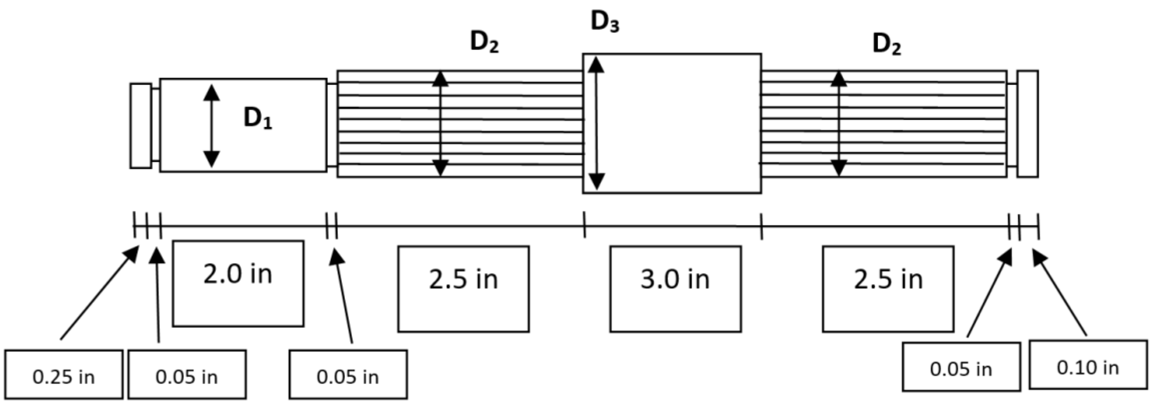


Figure 9: Shaft b

3.4.3.3 *Shaft c*

Shaft c, as seen in Figure *10*, connects gearbox stages 2 and 3, having the planet carrier of stage 2 splined onto one side and the sun of stage 3 on the other. This shaft uses AISI 4130 steel normalized to 1600°F.

D_1 is the diameter of the sections holding the stage 2 carrier on one side and the stage 3 sun on the other, measuring 2.2in. This section has a length of 2.52in on both sides.

D_2 is the diameter of the midsection of shaft locating the gears on either side (using shoulders), measuring 2.75in. This section is 0.5in long.

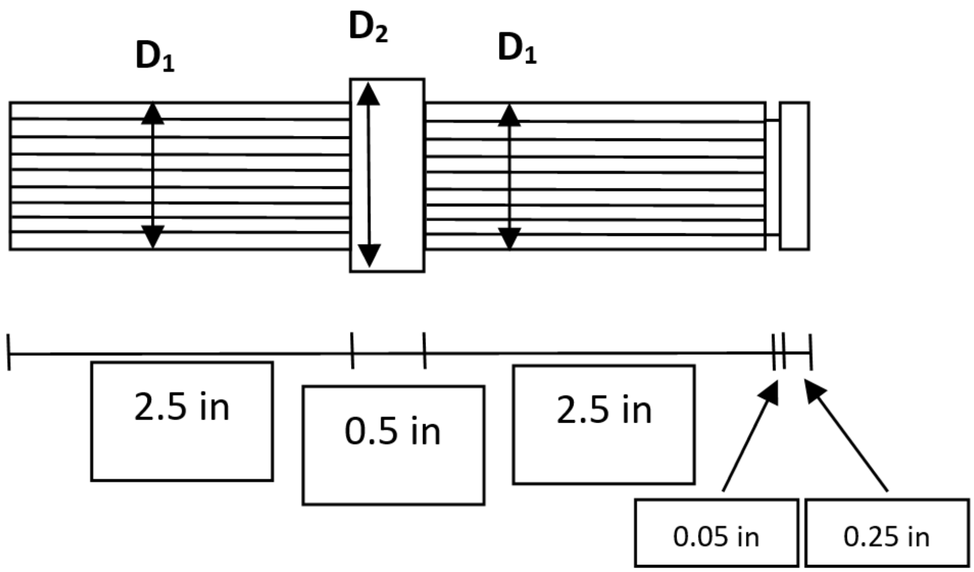


Figure 10: Shaft c

3.4.3.4 *Shaft d*

Shaft d, as seen in Figure *11*, is very similar to shaft c in its layout and serves the same purpose. This shaft uses AISI 4130 steel normalized to 1600°F.

D_1 is the diameter of the sections holding the stage 3 carrier on one side and the stage 4 sun on the other, measuring 4in. This section has a length of 2.52in on the stage 3 side and 4.00in on the stage 4 side.

D_2 is the diameter of the midsection of shaft locating the gears on either side (using shoulders), measuring 4.8in. This section is 0.5in long.

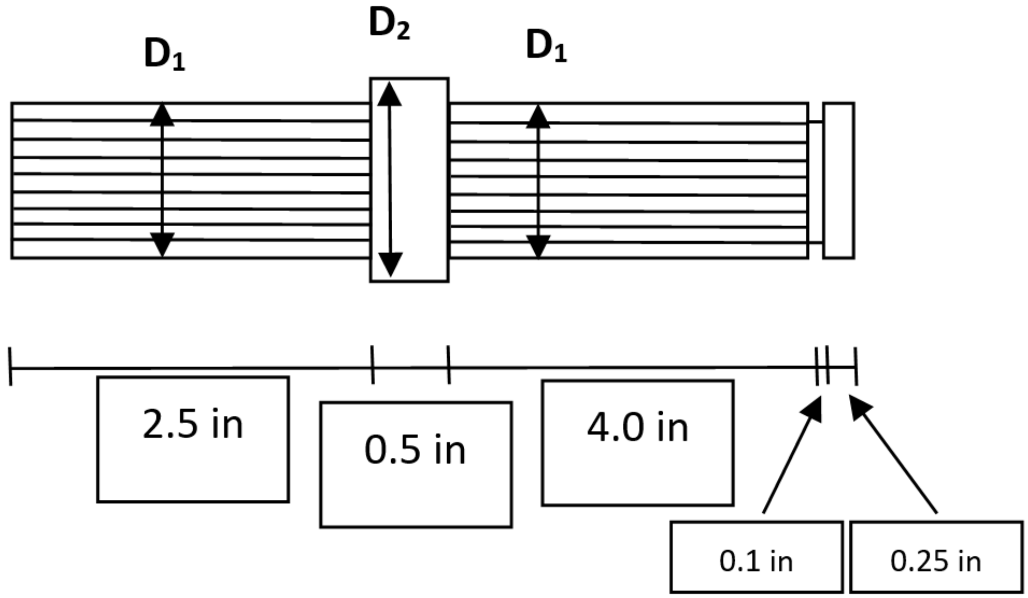


Figure 11: Shaft d

3.4.3.5 *Shaft e*

Shaft e, as seen in Figure *12*, is the final shaft and connects the output of the gearbox (stage 4) to the drive wheel of the wind tunnel platform. This shaft uses AISI 4150 normalized to 1600°F.

D_1 is the diameter of the sections holding the stage 3 carrier (3.15in) measuring 5.5in. This section has a length of 2.75in.

D_2 is the diameter of the section of shaft locating the stage 4 carrier and holding the bearing. It measures 6in, with a length of 1in.

D_3 is the diameter of the connection to the output (wheel) shaft. This measures 8in, having a length of 11.45in:

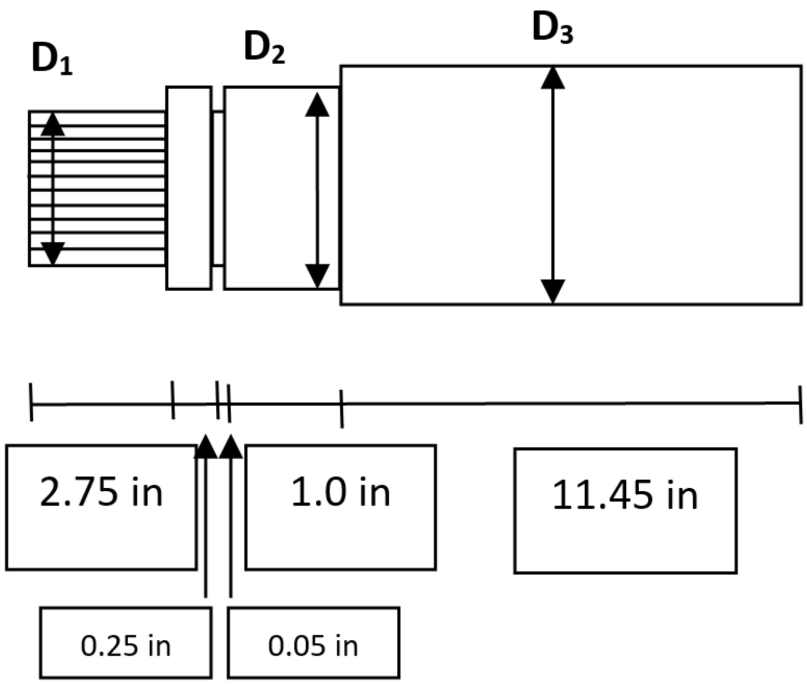


Figure 12: Shaft e

3.4.4 Bearings

The bearing selection for this drive system was selected based on availability. The system is not subjected to any thrust loads, but may be subject to some heavy shocks. Even in this case, since the wheels are on rails, and the motor is mounted to a support coming from the vicinity of the wheel, the thrust load is not expected to be high. Since the shafts are not rotating quickly or continuously, the established threats to the bearings were the risk of deformation due to high static loads, and low temperatures negatively affecting the lubricants. Heavy duty ball bearings were used, although, if further optimization were desired, cylindrical roller bearings should be examined. Many suppliers have comprehensive guides on bearing selection, here as a guide to check that the selected bearings would meet basic loads, the NTN bearing catalogue was used [14].

3.4.5 Lubrication

To prevent metal to metal contact within the gearbox, gear-specific lubrication must be used. For an enclosed gearbox that will only be used sparingly and resting in the cold outdoor weather, a synthetic gear oil should be used. Synthetic gear oils perform much better than conventional oils in cold operation temperatures [15]. Using oil with a low-temperature pour point, such as Duolec Syn Gear Lubricant 9815, will allow the gearbox to function properly. This lubricant meets the AGMA 9005-E02 standard and will cling to the gears, providing a film of protection against metal to metal contact.

4 Drive System Limitations

The final drive system has a few limitations imposed by several factors. Weather condition limitations occur due to available tractive force that is present. Three cases were explored: the maximum force the drive system can handle under dry conditions, wet conditions, and icy conditions. The wind speed and ice build-up maximums were determined for their respective cases. The starting resistance and acceleration forces did not change drastically, so it was assumed that they do not change, since a 120% allowance was already taken into account.

TABLE *XVII* is a summary of the drive system approximate limitations:

TABLE XVII: Limitations of the Drive system in Dry, Wet, and Icy conditions

	Dry Conditions	**Wet Conditions**	**Icy Conditions (extreme Conditions)**
Adhesion Coefficient	0.3	0.1	0.05
Wind Speed (km/h)	96	96	21.33
Ice Build-up (% Coverage on all 8 Wheels)	0	0	1.75% with 0.1” ice layer on top of rail
Temperature of ice (Degree C)	N/A	N/A	-20
Required Tractive Force (lbf)	18926.55	18926.55	9480.86
Available Tractive Force (lbf)	57055.063	19018.354	9509.177
Required Motor Torque (Nm)	48.73	48.73	24.41
Spec’d Motor Torque (Nm)	144	144	144

The dry conditions represent moving the wind tunnel during the fall and the wet conditions represent the rails flooded in water. During the dry conditions, there is room for additional weight to be added to the wind tunnel if it is necessary. The adhesion coefficient drops in the wet conditions since the friction is reduced, which results in the available tractive force to decrease drastically.

The ice build-up factor is the percentage of ice coverage that is on the wheel surface. In the case of the icy conditions, a 0.1in layer of ice is also on the top surface of the rail. This ice layer represents the same conditions as if a vehicle was driving on an ice patch. There is a minimal amount of friction, so the resulting available tractive force is reduced. If the ice is not cleaned off the rail at all, there is a large risk of wheel slip. The approximate strength of the ice is based on a -20°C temperature.

Also, the case of the ice is based on the structure being stationary, so momentum of the structure while moving is not taken into account. The structure will have an easier time breaking the ice while in motion. This consideration is also not taken into account for the special case below in TABLE *XVIII*.

TABLE *XVIII* highlights a case where there is a certain amount of ice coverage on the wheel, but the ice on the top of the rail has been scraped off, so the friction between the rail and the wheel is best represented by a dry condition:

TABLE XVIII: Special case when the top of the rails has been scraped free of ice.

	Dry Top Surface of Rail (Scraped Ice)
Adhesion Coefficient	0.3
Wind Speed (km/h)	44.8
Ice Build-up (% Coverage on all 8 wheels)	18
Temperature of frozen Ice (Degree C)	-25
Required Tractive Force (lbf)	52746.55
Available Tractive Force (lbf)	57055.063
Required Motor Torque (Nm)	135.8
Spec'd Motor Torque (Nm)	144

It is to be noted that the overall structure weight has been given a 120% allowance, so the drive system could potentially handle higher forces caused by the ice.

4.1 Finite Element Analysis

The dimensions of each component of the drive system were determined using appropriate theoretical methods. To supplement the theoretical calculations, a finite element analysis can be used to approximate if selected components are capable of handling the applied loads.

The structures that handle the greatest stresses are the critical components of this design. By theoretical calculation, the largest stresses were determined to occur in the 4th stage of the gearbox and the final shaft that attaches to the wheel, Shaft e. A finite element analysis using SolidWorks software was conducted on these components to determine if the theoretical calculations are valid.

4.1.1 Shaft e

Shaft e is made from AISI 4150 steel, normalized to 1600°F. The density of this material is 0.284 lb/in$_3$, the tensile yield strength is 107000 psi, and Poisson's ratio is 0.29 [16]. The torque that the shaft experiences was calculated to be 1047470 lbf-in at the spline connection between the stage 4 planet carrier and shaft e. Figure *13* shows that the face on the right side of the shaft is fixed.

Figure 13: Fixation of shaft e for finite element analysis.

The torque is then applied to a single face of each tooth on the left side as shown in Figure *14*.

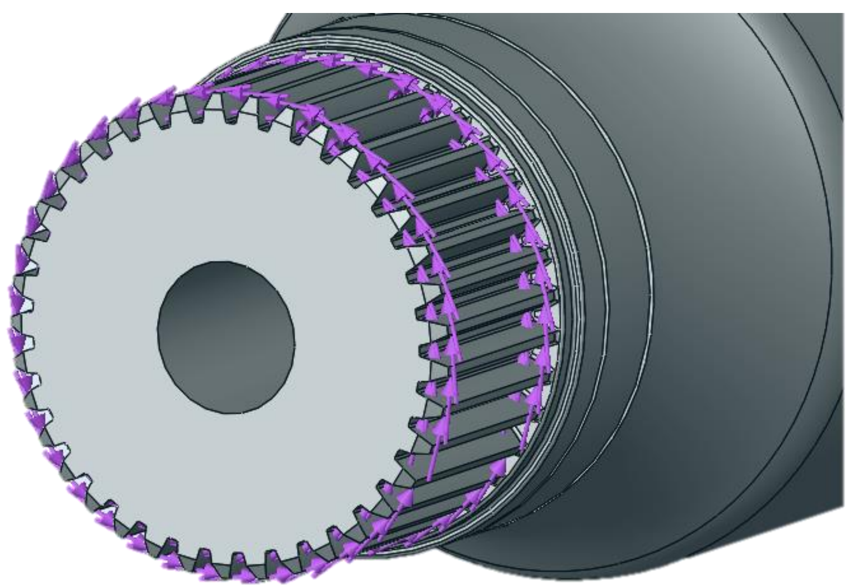


Figure 14: Torque application for the finite element analysis of shaft e.

The first simulation run was done with a coarser mesh to determine the higher stress areas and the model was adjusted to distribute the stresses throughout the shaft. The stress concentrations were also eliminated by adding fillets at every 90-degree angle. The resulting simulation with the adjustments to the model are shown in Figure *15*.

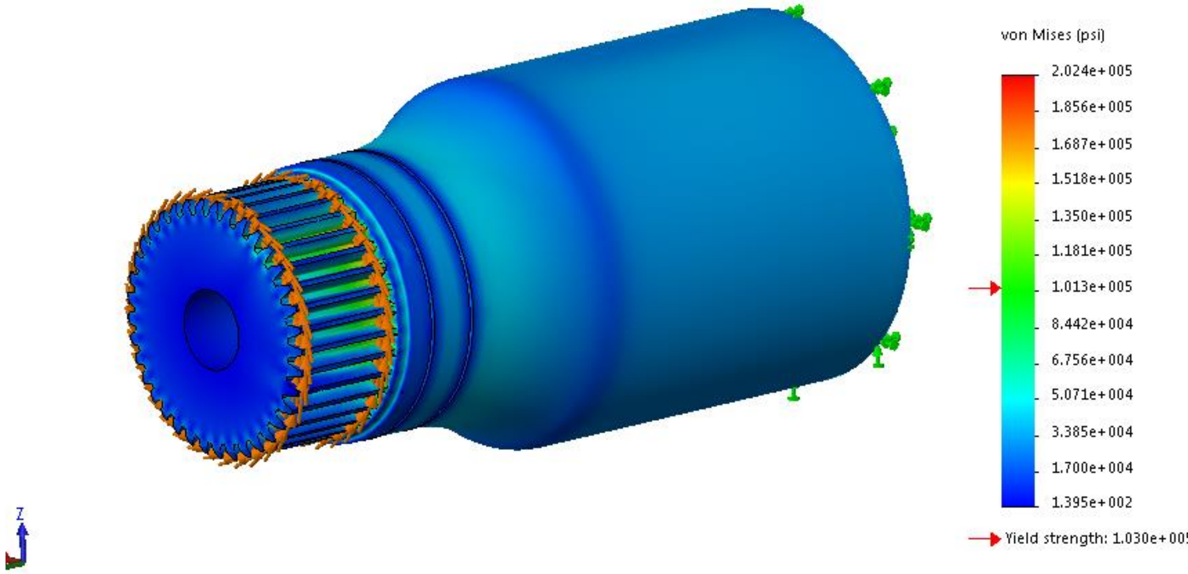


Figure 15: Coarse-mesh finite element analysis of shaft e to determine high stress concentration areas.

It is to be noted that the yield strength shown in Figure *15* is the strength for AISI 4130 steel. AISI 4150 steel has a slightly higher yield strength as 107000psi. The shaft has smooth transitions to evenly distribute the stress. The highest stress was found to be at the spline where the torque is applied. This is the most problematic spot out of the entire shaft. Figure *16* shows a closer view of the teeth of the spline.

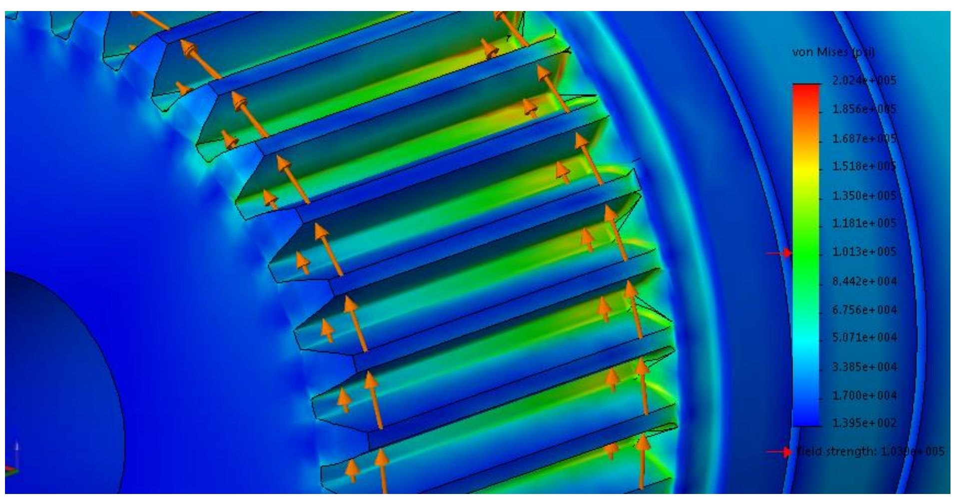


Figure 16: Finite element analysis of Shaft e spline.

The stresses are larger where the end of the spline and the shoulder meet. A larger fillet to transition between the features more smoothly is recommended. There are small stress concentrations at the bottom of each tooth. There are some areas that are higher than the yield strength of the material, these are places that need to be further examined to try to eliminate the higher stresses.

In order for these results to be valid, the stress value should converge as the number of nodes of the mesh increase (more detailed simulation). Figure *17* shows that the stress does converge.

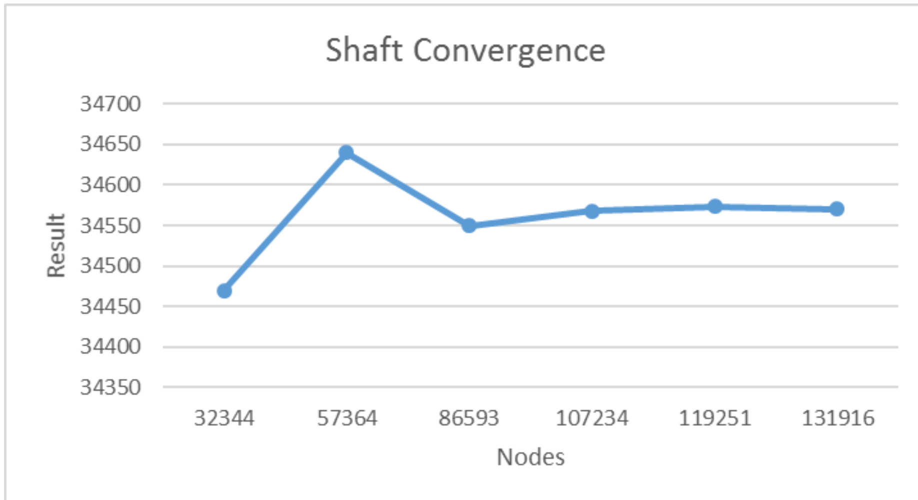


Figure 17: Convergence of the finite element analysis results on shaft e.

The probe point (sensor location) was placed in an area where no stress concentrations would be present in order to get valid results. Figure *18* shows the location of the probe.

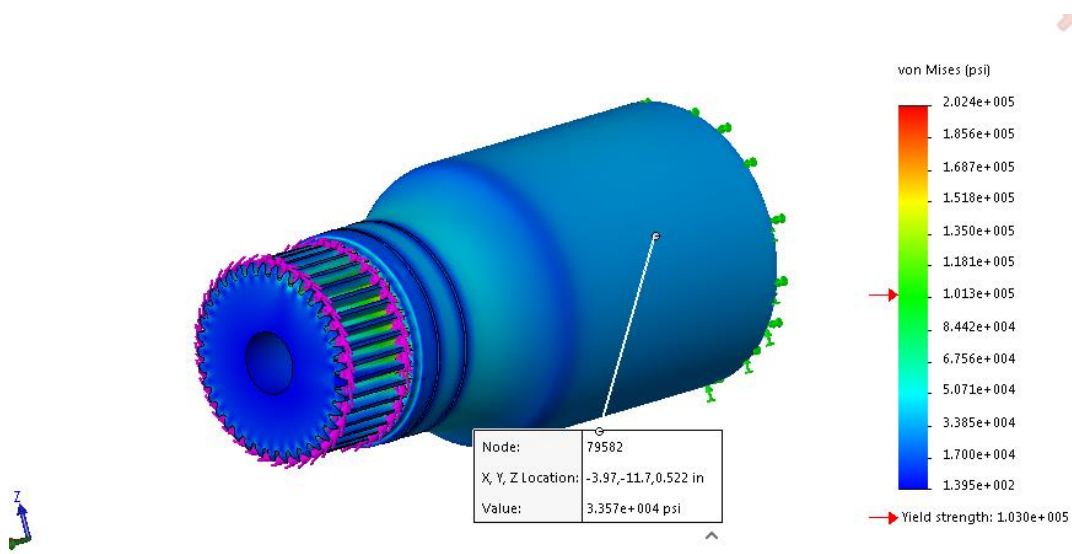
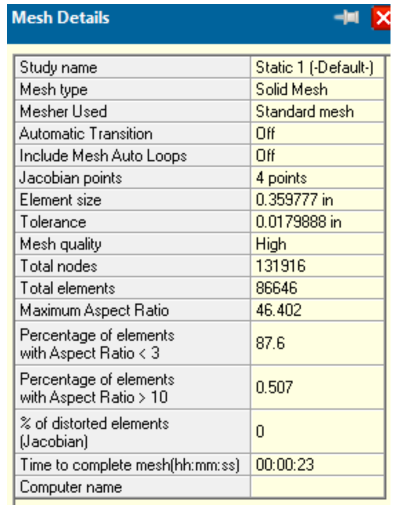


Figure 18: Location of probe used to establish convergence.

This is the location where the values in Figure *17* were derived from to achieve convergence.

The mesh details are shown in Figure *19*that were used for the final iteration of the simulation.



Mesh Details

Study name	Static 1 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.359777 in
Tolerance	0.0179888 in
Mesh quality	High
Total nodes	131916
Total elements	86646
Maximum Aspect Ratio	46.402
Percentage of elements with Aspect Ratio < 3	87.6
Percentage of elements with Aspect Ratio > 10	0.507
% of distorted elements (Jacobian)	0
Time to complete mesh(hh:mm:ss)	00:00:23
Computer name	

Figure 19: Mesh details for the final finite element analysis iteration.

A standard mesh was used with a total node count of 131916 nodes. The maximum stress of 2.024 x 10^5 psi was found in the teeth of the spline near the shoulder. This location should include a more gradual fillet or shoulder to blend the spline teeth with the rest of the shaft. The shaft itself is experiencing no more than 5.071 x 10^4 psi, which is below the yield strength of AISI 4150.

4.1.2 Stage 4 planet Carrier

The carrier is linked to the four planet gears via a pin connection. The carrier will always rotate with the planets and the torque will be transmitted through the spline of the carrier and onto the spline of the shaft. The pins are 4” in outside diameter and are supported by round plates on either side. The surfaces that the pin connections are in contact with are 0.75” wide. The pin connection section of the carrier has been overbuilt to overcome any stresses. The more problematic area is the spline. This is where the finite element analysis will be completed, to ensure the spline can handle the 1047470lbf-in of torque. Figure *20* shows the initial setup of the simulation, where the pin locations are fixed on the inside and the torque is applied to the inner spline.

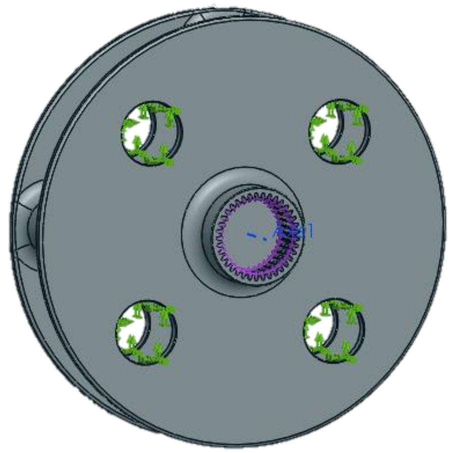


Figure 20 Torque application and fixed geometry of Finite Element Analysis of Stage 4 Carrier

A large fillet is placed between the round plate and the spline to gradually distribute the stresses coming from the pins onto the spline. A similar mesh was used for the carrier like with shaft e. The final mesh gave a stress distribution through the carrier shown in Figure 21.

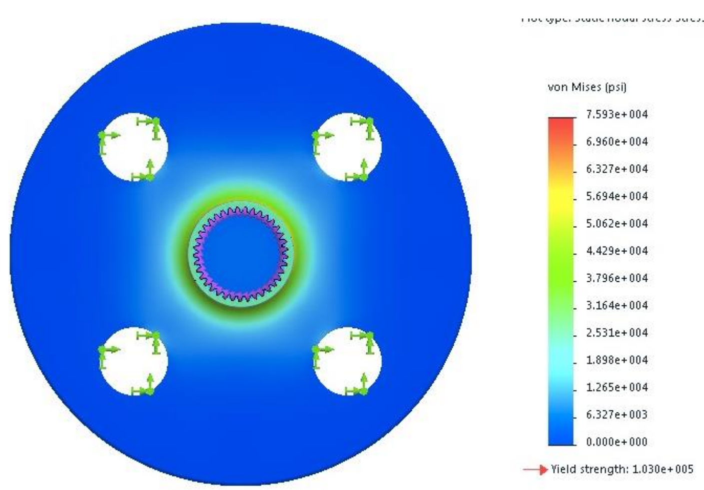


Figure 21 Stress distribution through the stage 4 carrier

The location of the highest stress is near the transition point of the plate to the spline. It can be seen that the yield strength of AISI 4340 is never exceeded throughout the entire carrier, which means that the carrier will be able to handle the large torque from the wheel. Figure 22 shows the highest stress area of the carrier.

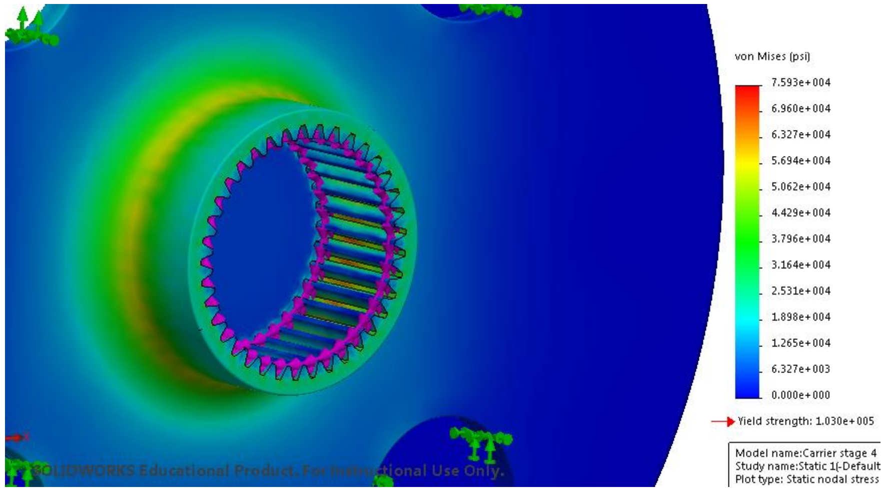


Figure 22 Stress transfer from round plate to spline

The transition point has a max stress of 5.694 x 10^4 psi. All the torque from the spline is being transmitted to the plate in an efficient manner that doesn't cause any stress concentrations. Figure 23 shows the stress distribution in the teeth.

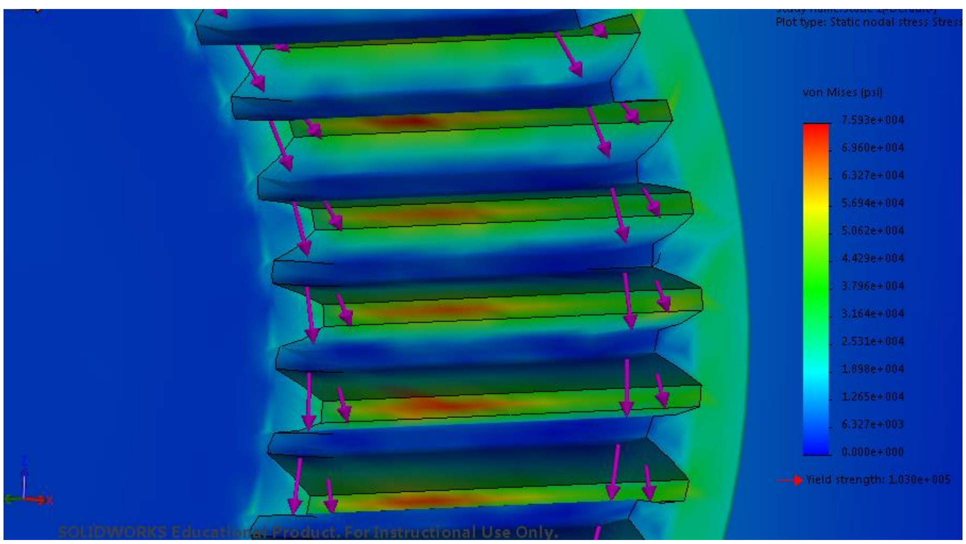


Figure 23 Stress concentrations in the teeth of the spline

The location where the spline meets the plate on the inside doesn't have a fillet to relieve the stress concentrations. This would cause the higher stresses as seen on the teeth. Also, the teeth don't have any fillets, which would also create the high stresses near the bottom of the teeth. These can be eliminated by adding fillets, which would result in a more realistic stress distribution.

In order for these results to be valid, the stress at any point on the carrier has to converge to one value. A probe (sensor) was placed on the carrier away from the spline and any stress concentrations as shown in Figure 24.

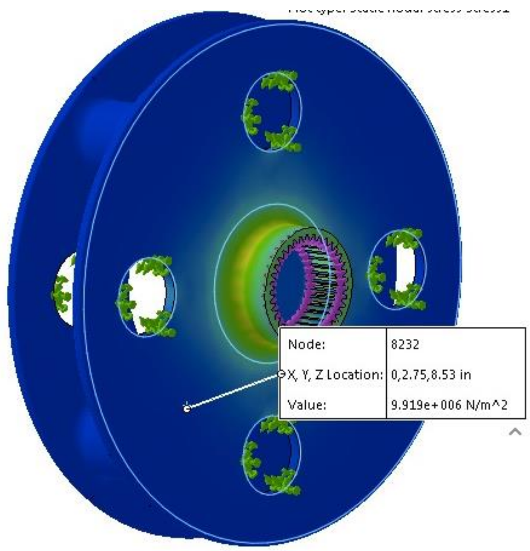


Figure 24 Location of probe to establish convergence

The mesh was increased multiple times to obtain a more detailed and accurate finite element analysis. A plot was created showing the number of nodes versus the stress at a specific location. The following figure shows that the stress converges.

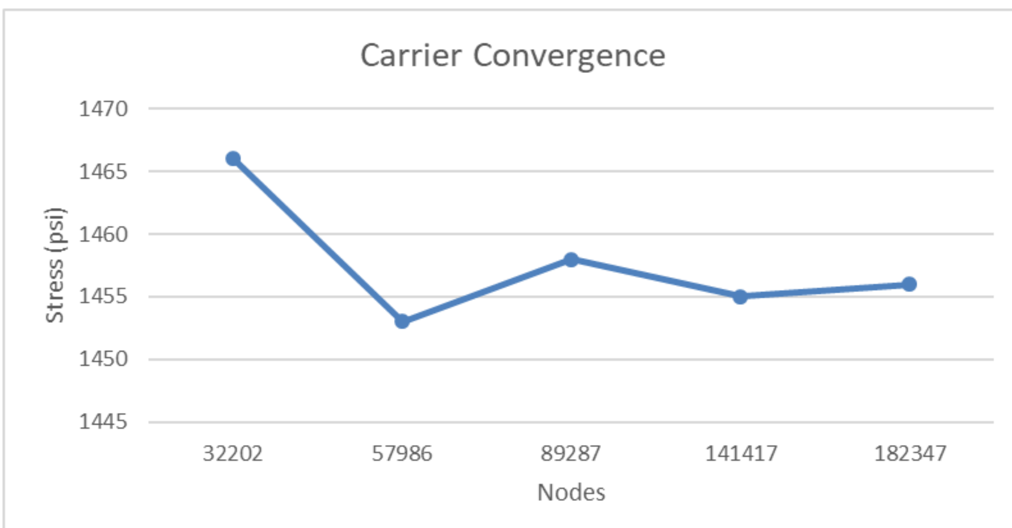


Figure 25 Convergence of the finite element analysis of stage 4 carrier

The geometry of the carrier is able to handle the torque that it experiences. This carrier is overbuilt and can be reduced in weight significantly to achieve a lighter design.

4.1.3 Stage 4 sun gear

The sun gear is the most important component of the planetary gear set because if that gear can't handle the forces then the gear box will fail and the wheels will stop spinning. The analysis of the stresses was conducted on a single tooth since the sun gear and planet gear interact through contact with a single tooth. The gear was remodeled as shown in Figure 26 to obtain a flat surface for the gear to be fixed.

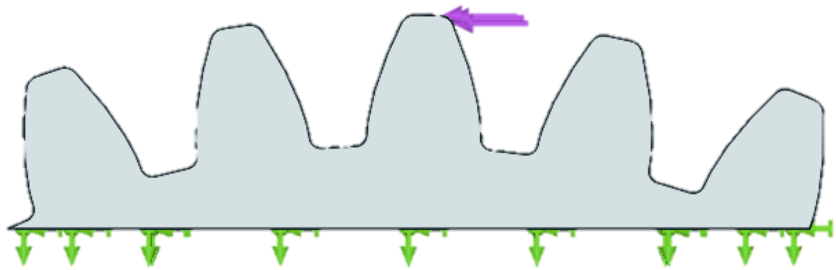


Figure 26: Contact Force (tangential force) application on one tooth for finite element analysis of sun gear

The geometry of the gear is consistent through five teeth but the force is only applied to one tooth. The contact force between the sun and planet gear is approximately 14519.51 lbf. This force is applied at the tip of the tooth tangentially. This will imitate a beam in bending and be the most accurate way to recreate the contact force through the tooth.

Thea standard mesh was used for the simulation but was optimized to have a higher node count near the bottom of the tooth to increase the accuracy since this will be the highest stress area. When a beam is in bending the highest stress area that is closest to the fixed location. Figure 27 shows how the mesh becomes finer near the bottom of the tooth.

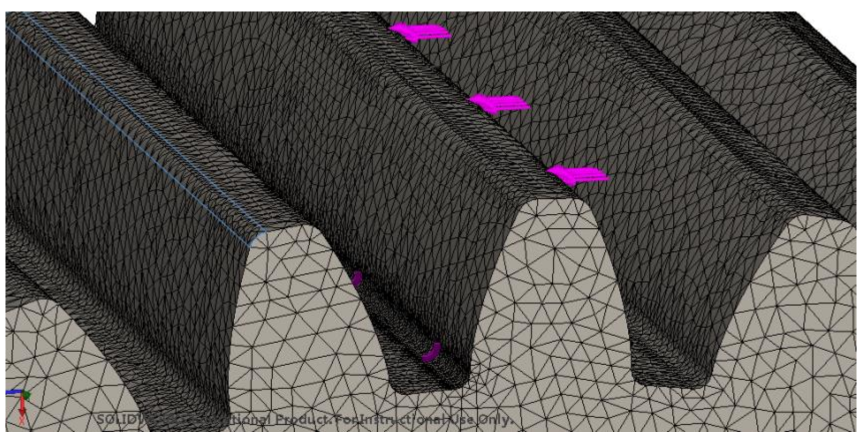


Figure 27: Mesh control applied to the gear with finer mesh near bottom of tooth

The radial forces are not included in this model, which should be validated to make sure the radial forces don't affect the tooth significantly. Figure 28 shows the stress distribution through the tooth and onto the body of the gear.

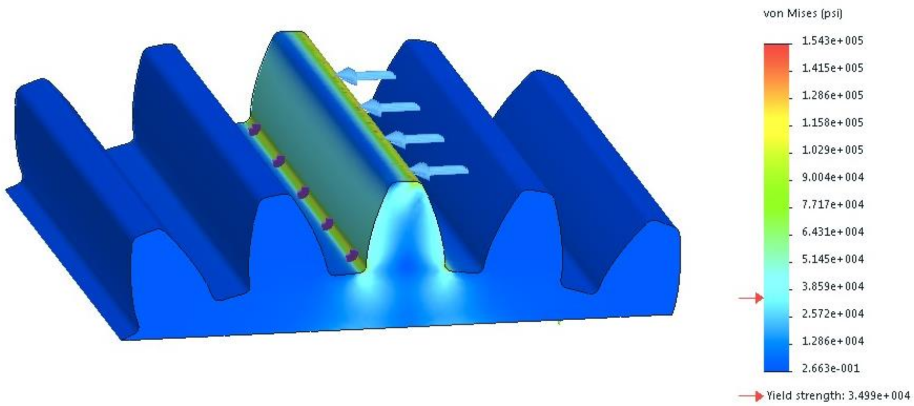


Figure 28: Stress distribution though one tooth and the face of the gear

The yield strength of the material in this analysis is not the correct yield strength. The actual value is 70 000 psi for allowable bending stress and 275 000 for allowable contact stress. The areas of high stresses are at the contact point and at the bottom of the tooth. This is to be expected. The highest stress area is at the contact point, which has a value of 154300psi. This is well under the allowable contact stress. Near the bottom of the tooth where a bending stress will be present, the stress is at a maximum of 90 004psi. This is above the 70 000psi allowable bending stress value so it is recommended to do a more accurate analysis with the proper material. The gear will be carburized and hardened to increase the toughness of the outer layer of the gear to reduce wear from those high stresses and to reduce pitting. This analysis should be completed with a software that can handle custom inputs to imitate a material with grade 3 carburized and hardened steel. There are no stress concentrations present in the model so this is an accurate representation of the stress distribution.

In order for this representation to be valid, a probe (sensor) has to placed far from any edges. Figure 29 shows where the probe was placed to determine if the stress value converges.

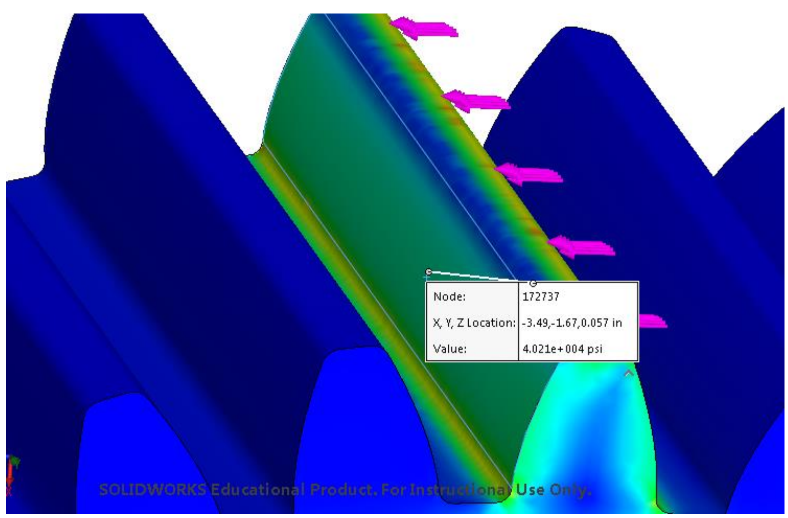


Figure 29: Location of probe to establish convergence

The mesh was increased in increments until the stress converged at that location on the tooth. Figure 30 shows the convergence plot.

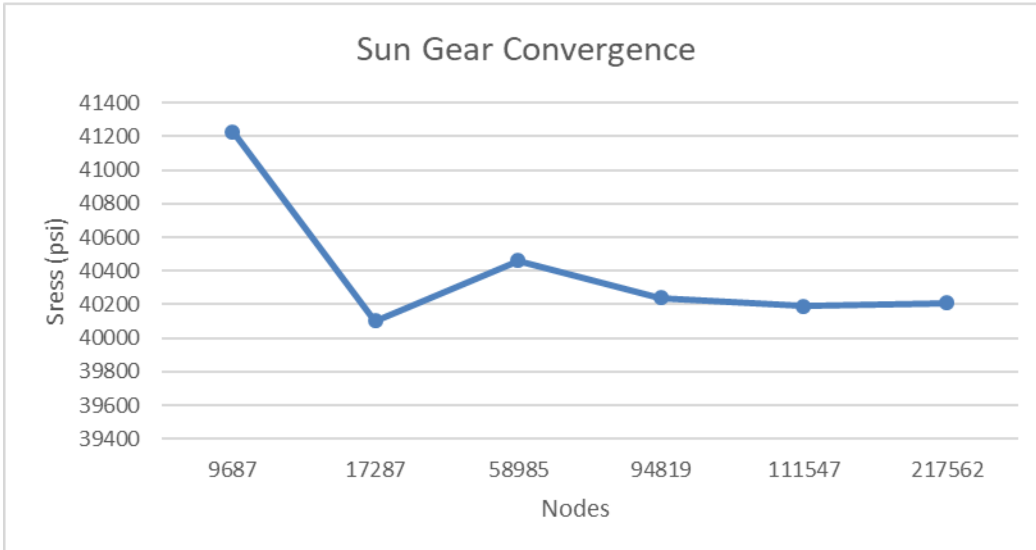


Figure 30: Convergence plot of finite element analysis of stage 4 sun gear

4.2 Estimated cost

The estimated cost will be based on motors, gearboxes and other components that are readily available from companies that can provide the necessary specifications that were determined in the calculations process. A similar gearbox to the custom one that was outlined in detail in this report will be purchased in order to reduce the cost.

The motor and frequency inverter were selected from NORD Drivesytems Inc. [10]. This selection was based on the fact that the products seemed reliable, based off of the company's reputation and that the available enclosures for both were available to the standard required for a drive system operating in harsh conditions.

The vendor that was selected to provide the gearbox was Bonfiglioli [17]. The gears and shafts within the gearbox were completely designed from scratch for the purposes of this project, but Bonfiglioli can provide an optimal system using pre-constructed gearboxes. The Bonfiglioli website suggests that their 300M Planetary Gearboxes can withstand torques from 1250 Nm to 1286000 Nm and gear ratios from 3.4 to 5234. Unfortunately, at the time of publication of this report, a quote was requested, but not received from the supplier.

The brake system was quoted from SEPAC Industries due to their power off electromagnetic brakes with a large bore selection. The system was initially intended to be mounted at the large output shaft, and SEPAC was the only supplier found that could supply large enough brakes. When the mounting placement was changed to behind the motor, it was too late to request a quote from another manufacturer, but SEPAC was able to provide a quote for a smaller brake on at a last moments notice.

TABLE *XIX* highlights the cost for all the parts with the vendor that is providing those components.

TABLE XIX: Sourced parts, supplier and cost if available.

Component	Vendor	Cost
Motor*	Nord Drivesystems	
Frequency inverter*	Nord Drivesystems	
Brakes	SEPAC Industries	$492.78 per unit (2 required)
Gear train*	Bonfiglioli	
Shaft (Gearbox to Wheel)	Possibly provided by gearbox supplier	

*Quotes were requested, but not received at the time of report publication

5 Design Summary

To begin the third phase of the project, preliminary calculations needed to be made to determine what size of motor was required. Items such as wind force, adhesion, and force due to ice on the rails were all calculated prior to defining the motor.

The overall design consists of two separate but identical drive systems. Each drive system attaches to one of the existing mounting plates and the existing wheel shaft. The motors that were selected are brushless, 3-phase, AC induction squirrel cage motors. The motors are capable of delivering 30HP, 144Nm of torque, and have built-in torque overload protection. The motors also require frequency inverters to control the speed.

The braking system uses electromagnetic power-off brakes, so the brakes engage automatically if power is lost when the wind tunnel is in motion. These brakes are simple to maintain and operate, are reliable in extreme weather, and are able to completely immobilize the wind tunnel platform.

The power transfer system is a four-stage gearbox.

The first stage of the gearbox is a gear-and-pinion design, while the last three stages are planetary spur arrangements with four planets each. The input to each planetary gear stage is the sun gear, while the output is the planet carrier. The speed reduction ratio of all four gear stages combined is 790:1. To transfer the power from stage to stage, and ultimately to the wheel, splined shafts were designed to tolerate the torque applied.

5.1 Client Needs Evaluation

To evaluate the design, project specifications were developed as a means of providing quantitative evidence that the drive system functions as requested by the client. This section of the report summarizes the project specifications and how the proposed design measures up to the desired results. However, many of the metrics cannot be effectively measured until the drive system is installed, so only the specifications that can currently be measured are included.

Metric 1: Corrosion resistance
Despite not obtaining values for the corrosion resistance of the components directly, the design team selected enclosed and protected components to prevent corrosion as much as possible.

Metric 3: Existing rail system is used
The design team was adamant on eliminating any complicated or outside-the-box methods to move the wind tunnel. The clients requested a drive system that was simple and making use of the existing rail system was the simplest design.

Metric 4: Drive system can perform two full passes of rail at -40°C
Lubrication was selected to perform well in low temperatures. Obstruction by ice was taken into account.

Metric 5: Year of operation
The gears were design to have a design life of 20 years based on the number of expected cycles that they would encounter over that time. The motor and frequency inverter were sourced to be reliable and have a long service life in harsh conditions.

Metric 6: Number of start and stop methods
The frequency inverter selected provides a reliable method of starting and stopping the wind tunnel, the electromagnetic brake can be connected to the inverter so that all control can be conducted from one point. Additionally, a handheld controller is available to allow the operator to stand a short way away from the wind tunnel during operation.

Metric 8: Obstructions that will prevent wheels from rolling
Ice was determined to be a very limiting factor when it came to evaluating what the drive system could handle. Limitations are largely due to the available tractive force when ice is present and not the strength of the drive system. As such, a maximum amount of snow and ice resistance, as well as wind resistance, was calculated so the client would be aware of what the system was realistically capable of overcoming.

Metric 9: Compliance with test site safety policies
Using an enclosed design prevents exposure of employees to moving parts. Power-off type brakes ensure that structure cannot move unless it is being operated by an employee.

Metric 10: Overload protection engagement
The sourced motor is complete with built-in torque overload protection, so overload protection was not incorporated into other design components.

Metric 11, 12: Speed test
The drive system was designed to move the wind tunnel at a speed of 2m/min, which is between the ideal and marginal acceptance values. This speed allows an employee to move the wind tunnel from one end of the rails to the other in 15 minutes.

Metric 13, 14: Damage to structure
By eliminating the forklift method that was used to move the wind tunnel before, the platform will have a significantly longer life. Also, the drive system was designed to slowly accelerate and decelerate, thus further reducing stresses on the platform.

Metric 17: Drive system power requirements
The available voltage is 208 V 3-phase, however the configuration of frequency inverter and motor that would satisfy the power and torque requirements for the drive system required 400 V, thus a transformer would be required to power the drive system. The AC motors found needed at least 230V to run but with the inverter to reduce the motor rpm, a transformer is needed.

Metric 18: Footprint in excess of original platform
The drive system was designed to be compact, using two independent drive systems to drive two wheels. The existing mounting plates were measured and used to gauge how large the system could be without interfering with the rest of the structure. The original mount does need to be move back 4.5in to fit the gear box.

Metric 19: Percentage of parts with quotes
Out of the three total quote requests that were sent out (one for brakes, one for motor and VFD, one for gearbox), only the brake vendor returned a quoted price. The quote requests for the gearbox and motor were awaiting response at the time of report publication.

5.2 Recommendations

The use of a drive system using a planetary gear train and electric motor looks promising, however, the potential exists for significant amounts of further optimization. This section outlines a few further optimizations that were identified but not able to be implemented due to time constraints or knowledge limitations.

1. The gears themselves can be reduced in size and weight by the use of spoked and rimmed gears. Late in the design finalization, the team also determined that 30-teeth sun gears may be viable based on theoretical calculations. Fewer teeth would reduce the size of the planetary gear-sets, lowering the weight and lowering the cost of the system.
2. The bearings and bearing/gear lubrication should be revisited, due to low thrust loads, cylindrical roller bearings/needle bearings may be viable options. Further research should be conducted in appropriate lubricants that are intended for the wide temperature range experienced by the drive.
3. If a simpler motor operating at the available voltage were to be found, this would remove the need for a transformer to step up the voltage, reducing the cost of the drive. For the given design, the supplier only offered the selected frequency inverter at the higher voltage used in the design and no alternate frequency inverter was found.
4. The information on the effects of ice on the rails with regards to the adhesion coefficient and force prevention motion would require further research, as data currently available to the design team relating to this application is not sufficient.
5. The suppliers of the optimal drive system could change once the cost of the components is determined, the selection here was made on the basis of component capability and construction.

6. Required brake force should be reviewed, specifically as to the effects of braking through a large reduction as well as the stress experienced by the gearbox (vs. the driving stress it was designed to handle).
7. The shaft coming out of the motor is currently too large for the recommended brake size. While stepping this shaft size down or machining it down is an option, it increases complexity and/or failure points into the system. Upsizing to a larger brake would be simpler, although may be costlier and excessive.
8. The AC motor generally had a working rpm between 1450-1760rpm. The speed that the team wanted at the input was 550rpm. Since ac motors don't go that low, a frequency inverter needed to be used, which raised the cost. It is recommended to further research if having the input run at the rated motor speed would decrease overall costs. It would mean that the gearbox would have to be redesigned but would eliminate the inverter and possibly eliminate the transformer since a 208V motor could be found.

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Appendix A – Project Management

Table of Contents

List of Tables

Client Needs

The client needs are what the stakeholders view as outputs and results of the project. TABLE I lists prioritized client needs that were determined at the outset of the project. The "Importance" column is based on a scale from 1 to 5, where a '1' represents a low priority need and a '5' represents a high priority need. The priority levels were determined by the design team and approved by WestCaRD and StandardAero based on information from the project kickoff meeting:

TABLE I: WEIGHTED NEEDS LIST

Need ID	Need	Importance	Need ID	Need	Importance
A	Drive system is operable after long periods of inactivity	4	M	Drive and braking systems operate without damage to platform or wind tunnel systems	5
B	Drive system is simple to maintain	3	N	Platform remains stationary unless being moved by drive system	5
C	Drive system propels platform using existing rail system (171# Crane Rail)	5	O	Platform can stop in an emergency	5
D	Drive system is operable in extreme weather	5	P	Drive system can be operated from control room	2
E	Drive system has long working life	3	Q	Drive system requires no more than 120/208 single/3-phase power	4
F	Drive system has control redundancies (multiple ways to stop/start)	3	R	Drive system does not obstruct working area	3
G	Drive system can start/stop at any point over the length of the track	5	S	Drive system is contained on platform	4
H	Drive system can start regardless of obstruction	5	T	Vendors will be contacted and components sourced/priced	3
I	Drive system is safe for operator	5	U	Drive system is simple to operate	4
J	Drive system is safe for bystanders	5	V	Total cost of drive system is kept to a minimum	3
K	Drive system has a form of overload protection if design torque is exceeded	5	W	Drive system is aesthetically pleasing	1
L	Drive system moves at a safe speed	5			

To clarify each of the needs in TABLE I, a brief explanation of each requirement is provided as follows:

Need ID A: Drive system is operable after long periods of inactivity.
The drive system will only be used one or two times a year for the tests that require the wind tunnel to be moved. Assuming that the drive system will be operated twice a year, the components should be able to operate successfully after being unused for six to eight months.

Need ID B: Drive system is simple to maintain.
If the final drive system design requires lubrication, then lubrication must be the only maintenance item that StandardAero operators will need to perform. The other components of the drive system and braking system will be designed to last the life of the wind tunnel platform; components will be robust and, if they do fail, be easily replaceable.

Need ID C: Drive system propels platform using existing rail system (171-lb crane rail).
Using the existing rail system will reduce the cost of the drive system. The wind tunnel is currently positioned correctly on the rails in relation to the turbine engine mounting rig and the rest of the facility.

Need ID D: Drive system is operable in extreme weather.
The TRDC is an outdoor facility that operates year-round in temperatures ranging from -40°C to +40°C and in windy conditions. The harsh climate and large temperature range must not be an issue when the wind tunnel must be moved.

Need ID E: Drive system has long working life.
The drive system needs to be designed with a long endurance limit because replacing the components will become expensive and will delay the engine tests that need to be performed.

Need ID F: Drive system has control redundancies (multiple ways to stop/start).
If one set of controls fails during the movement of the wind tunnel, there must be another way or set of controls to stop the wind tunnel. Multiple control locations will also make moving the wind tunnel more convenient for the operators.

Need ID G: Drive system can start/stop at any point over the length of the track.
The tests that are conducted at the facility require the wind tunnel to be positioned at various points along the length of the track. Additionally, the wind tunnel must be able to stop at any point on the track in the case of an emergency.

Need ID H: Drive system can start regardless of obstruction.
Since the test facility operates in winter, ice buildup can occur on the rails. The drive system must be able to overcome the additional force encountered when unremovable ice is in front of the wheels.

Need ID I: Drive system is safe for operator.
The operation of the drive system must not pose a risk to the person using the equipment. Considerations include that any long-term risks (such as hearing impairment) be addressed so that reasonable safety measures can be implemented when using the drive system.

Need ID J: Drive system is safe for bystanders.
The operation of the system must be safe for bystanders. Everyone in the testing facility must be able to easily identify that the system is about to move, or is already in motion.

Need ID K: Drive system has a form of overload protection if design torque is exceeded.
Unforeseen situations are possible and if the system were to undergo an unexpected power surge, an overload protection system would be crucial in preventing damage to the drive system.

Need ID L: Drive system moves at a safe speed.
When moving such a large and heavy structure (1.2 million pounds), a slow speed will ensure the safest possible working environment for all bystanders.

Need ID M: Drive and braking systems operate without damage to platform or wind tunnel systems.
Since 1.2 million pounds in motion can produce a large amount of momentum, having a brake system that is designed to account for such momentum will help avoid any unwanted stresses in the testing platform or associated structures.

Need ID N: Platform remains stationary unless being moved by the drive system.
The platform must be at a standstill unless the drive system is being operated to move it. The platform's weight, a slight incline, or other external forces such as wind should not be able to move the structure.

Need ID O: Platform can stop in an emergency.
While the wind tunnel is in motion, it is possible that the motion must be stopped quickly, therefore, the drive system must have an option to stop the wind tunnel from moving as soon as the emergency stop sequence is initiated.

Need ID P: Drive system can be operated from control room.
Providing redundant controls will increase the convenience of the system for the operators. Remote control of the drive system will allow operators to remain in the control room to move the wind tunnel. Operators will not necessarily have to leave the control room to move the wind tunnel to the correct position.

Need ID Q: Drive system requires no more than 120/208 Single/3-phase power.
The available power supply is 120/208 Volt single/3-phase outlet, so if electrical power is required, using the current setup will simplify the design and minimize design costs.

Need ID R: Drive system does not obstruct working area.
There are multiple points around the wind tunnel that the employees need to access for maintenance and setup of the wind tunnel. The drive system will not obstruct these locations around the wind tunnel.

Need ID S: Drive system is contained on platform.
Housing the drive system outside the original footprint of the platform would increase costs and weight. The drive system needs to be able to move with the wind tunnel as it is moving along the crane rail.

Need ID T: Vendors will be contacted and components sourced/priced.
The components for the final design may be specialized and difficult to source. Finding vendors that are willing to supply the required components for a reasonable price will be a large part of creating a viable design.

Need ID U: Drive system is simple to operate.
A drive system design that is simple to operate will minimize training requirements. Simplicity will also reduce the chance that errors in operation could occur.

Need ID V: Total cost of drive system is kept to a minimum.
The previous design that was conceptualized for the previous version of the wind tunnel was too expensive. Keeping the cost to a minimum is ideal, since the system will only be used one or two times a year.

Need ID W: Drive system is aesthetically pleasing.
Ideally, the drive system will look like it is a part of the finished platform that is currently in place. All external drive system components that are visible will be painted the same color as the platform.

Project Specifications

To measure the performance of the design of the drive system, the needs from **Error! Reference source not found.**, each require a means to measure how effective the drive system design is in each respective category. TABLE II compiles a list of the metrics created to test the design against the customer's needs:

TABLE II: METRICS, MARGINAL VALUES, AND IDEAL VALUES

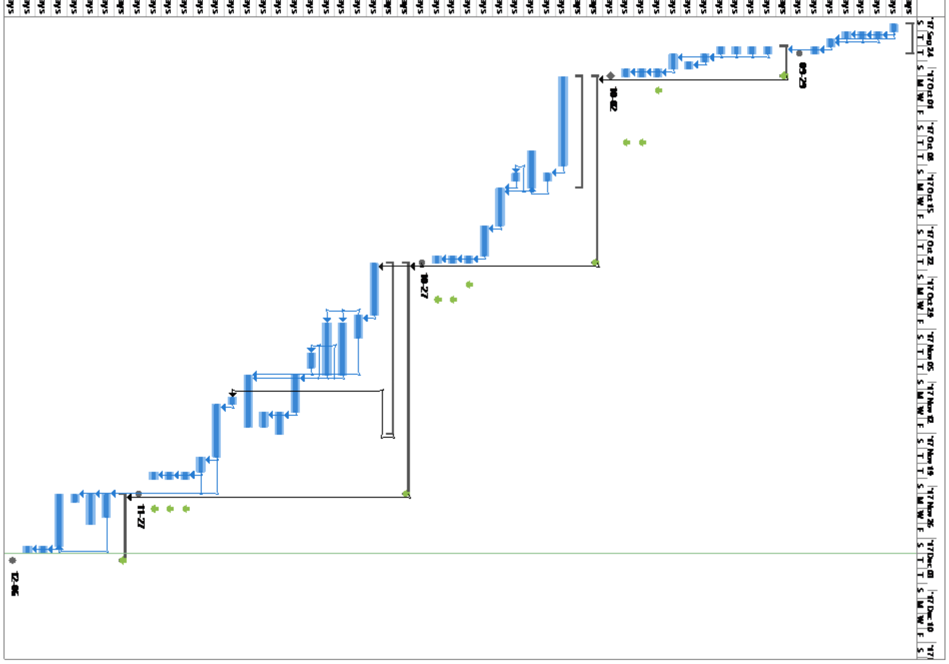
Metric #	Need ID	Metric	Units	Marginal Value	Ideal Value
1	A	Corrosion resistance	mm/year	1 to 3	<1
2	B	Time to perform annual maintenance of drive system	Minutes	<120	<60
3	C	Existing rail system is used	Binary	Pass	Pass
4	D	Drive system can perform two full passes of rail system at -40C	Binary	Pass	Pass
5	E, O	Years of operation before major overhaul/replacement of drive system	Years	10	>20
6	F, P	Number of start and stop methods	Number	1	3
7	G	Minimum distance travelled	cm	30	1
8	H	Height/mass of obstruction that will prevent wheels from rolling	cm/kg	TBD	TBD
9	I, J	Compliance with test site safety policies for moving equipment	Binary	Pass	Pass
10	K	Percentage of the maximum design torque at which overload protection engages	Percentage	90%	85%
11	L	Minimum speed test	m/min	1	0.1
12	M	Maximum speed test	m/min	<5	1
13	N	Damage occurs during drive system initiation	Binary	Pass	Pass
14	M, O	Damage occurs during braking system initiation	Binary	Pass	Pass
15	N	Static brake system is operational	Binary	Pass	Pass
16	O	Distance travelled between emergency stop activation and platform coming to a stop	m	<1	0.5
17	Q	Drive system power requirements do not exceed available power	Binary	Pass	Pass
18	R, S	Footprint in excess of original platform	m^2	<5	0
19	T	Percentage of total parts that are sourced with vendor confirmation	Percentage	>50	100
20	U	Operator training course grade	Percentage	>80	100
21	U	Time to explain drive system to stakeholders	Minutes	<120	60
22	V	Quoted costs	$	TBD	TBD
23	W	Stakeholder opinion	subj	3 out of 5	5 out of 5

Project Schedule

The Gantt chart is the tool that the project team used throughout the project to track progress and stay on schedule. TABLE III is the completed project Gantt chart. The task table shows all project task relationships and durations:

TABLE III: PROJECT GANTT CHART

ID		Task Mode	% Complete	Task Name	Duration	Start	Finish	Predecessors	Finish Slack	Free Slack
1			100%	**Problem Definition and Client Approval**	4 days	Mon 17-09-25	Fri 17-09-29		0 days	0 days
2			100%	Identify customer needs	1 day	Mon 17-09-25	Tue 17-09-26		0 days	0 days
3			100%	Identify preliminary risks	1 day	Tue 17-09-26	Wed 17-09-27	2	0 days	0 days
4			100%	Identify preliminary assumptions	1 day	Tue 17-09-26	Wed 17-09-27	2	0 days	0 days
5			100%	Identify preliminary constraints and limitations	1 day	Tue 17-09-26	Wed 17-09-27	2	0 days	0 days
6			100%	Create scope description	1 day	Wed 17-09-27	Thu 17-09-28	2,3,4,5	0 days	0 days
7			100%	Create "Project Scope for Client Approval"	1 day	Thu 17-09-28	Fri 17-09-29	6	0 days	0 days
8			100%	Obtain customer approval	0 days	Fri 17-09-29	Fri 17-09-29		0 days	0 days
9			100%	**Project Definition Report**	4 days	Thu 17-09-28	Mon 17-10-02	6	0 days	0 days
10			100%	Develop problem statement and background information	1 day	Thu 17-09-28	Fri 17-09-29		0 days	0 days
11			100%	Develop project objectives	1 day	Thu 17-09-28	Fri 17-09-29		0 days	0 days
12			100%	Identify target specifications and metrics	1 day	Thu 17-09-28	Fri 17-09-29		0 days	0 days
13			100%	Develop report template	1 day	Thu 17-09-28	Fri 17-09-29		0 days	0 days
14			100%	Write and compile report	1 day	Fri 17-09-29	Sat 17-09-30	10,11,12,13	0 days	0 days
15			100%	Edit and format report	1 day	Sat 17-09-30	Sun 17-10-01	14	0 days	0 days
16			100%	Develop oral report slides	2 days	Fri 17-09-29	Sun 17-10-01	12	0 days	0 days
17			100%	Complete peer evaluations	1 day	Sun 17-10-01	Mon 17-10-02	16	0 days	0 days
18			100%	Compile meeting minutes for advisor	1 day	Sun 17-10-01	Mon 17-10-02	16	0 days	0 days
19			100%	Develop presentable Gantt chart	1 day	Sun 17-10-01	Mon 17-10-02	16	0 days	0 days
20			100%	Hand in PDR	0 days	Mon 17-10-02	Mon 17-10-02		0 days	0 days
21			100%	**Concept Design Report**	25 days	Mon 17-10-02	Fri 17-10-27	9	0 days	0 days
22			100%	**Develop concepts**	15 days	Mon 17-10-02	Tue 17-10-17		0 days	0 days
23			100%	Brainstorm	12 days	Mon 17-10-02	Sat 17-10-14		0 days	0 days
24			100%	Site visit	1 day	Sun 17-10-15	Mon 17-10-16	23FS+1 day	0 days	0 days
25			100%	Evaluate most promising concepts	5 days	Thu 17-10-12	Tue 17-10-17	24FF+1 day	0 days	0 days
26			100%	Develop report template	1 day	Sun 17-10-15	Mon 17-10-16	25FS-2 days	0 days	0 days
27			100%	Write report	5 days	Tue 17-10-17	Sun 17-10-22	25,26	0 days	0 days
28			100%	Edit and format report	4 days	Sun 17-10-22	Thu 17-10-26	27	0 days	0 days
29			100%	Complete peer evaluations	1 day	Thu 17-10-26	Fri 17-10-27	28	0 days	0 days
30			100%	Compile meeting minutes for advisor	1 day	Thu 17-10-26	Fri 17-10-27	28	0 days	0 days
31			100%	Develop presentable Gantt chart	1 day	Thu 17-10-26	Fri 17-10-27	28	0 days	0 days
32			100%	Hand in CDR	0 days	Fri 17-10-27	Fri 17-10-27		0 days	0 days
33			100%	**Final Design Draft Report**	31 days	Fri 17-10-27	Mon 17-11-27	21	0 days	0 days
34			100%	**Final design work**	23 days	Fri 17-10-27	Sun 17-11-19		0 days	0 days
35			100%	Perform preliminary design calculations	7 days	Fri 17-10-27	Fri 17-11-03	32	0 days	0 days
36			100%	Determine drive system input (motor)	3 days	Fri 17-11-03	Mon 17-11-06	35	0 days	0 days
37			100%	Determine drive system output (gearbox and shafts)	7 days	Sat 17-11-04	Sat 17-11-11	36SS+1 day	0 days	0 days
38			100%	Determine braking system	7 days	Sat 17-11-04	Sat 17-11-11	36SS+1 day	0 days	0 days
39			100%	Determine control system	2 days	Wed 17-11-08	Fri 17-11-10	37FS-3 days,	0 days	0 days
40			100%	Develop CAD models of components	5 days	Sat 17-11-11	Thu 17-11-16	37,38	0 days	0 days
41			100%	Create engineering drawings	3 days	Thu 17-11-16	Sun 17-11-19	40	0 days	0 days
42			100%	Perform FEA on critical components	2 days	Thu 17-11-16	Sat 17-11-18	40	0 days	0 days
43			100%	Contact vendors to obtain quotes on parts	7 days	Sat 17-11-11	Sat 17-11-18	36,37,38,39	0 days	0 days
44			100%	Develop draft report template	1 day	Tue 17-11-14	Wed 17-11-15	34FS-5 days	0 days	0 days
45			100%	Write draft report	7 days	Wed 17-11-15	Wed 17-11-22	44	0 days	0 days
46			100%	Edit and format draft report	2 days	Wed 17-11-22	Fri 17-11-24	45	0 days	0 days
47			100%	Compile meeting minutes for advisor	1 day	Fri 17-11-24	Sat 17-11-25	46	0 days	0 days
48			100%	Develop presentable Gantt chart	1 day	Fri 17-11-24	Sat 17-11-25	46	0 days	0 days
49			100%	Evaluate another team's report	1 day	Fri 17-11-24	Sat 17-11-25	46	0 days	0 days
50			100%	Hand in FDDR	0 days	Mon 17-11-27	Mon 17-11-27		0 days	0 days
51			100%	**Final Design Report**	9 days	Mon 17-11-27	Wed 17-12-06	33	0 days	0 days
52			100%	Edit and update draft report	3 days	Mon 17-11-27	Thu 17-11-30	46	0 days	0 days
53			100%	Create presentation slides	4 days	Mon 17-11-27	Fri 17-12-01	46	0 days	0 days
54			100%	Submit presentation attendance	1 day	Mon 17-11-27	Tue 17-11-28	45	0 days	0 days
55			100%	Develop poster	7 days	Mon 17-11-27	Mon 17-12-04	52FF-2 days	0 days	0 days
56			100%	Complete peer evaluations	1 day	Mon 17-12-04	Tue 17-12-05	55	0 days	0 days
57			100%	Complete mechanical engineering survey	1 day	Mon 17-12-04	Tue 17-12-05	55	0 days	0 days
58			100%	Hand in FDR	0 days	Wed 17-12-06	Wed 17-12-06		0 days	0 days



Risk Management Plan

As with every project, risk is inevitable. This section of Appendix A, which was created at the start of the project, contains a list of events that could have either caused problems with the design (threats), or could have benefited the project team and stakeholders (opportunities). Understanding the possible risks was crucial to being able to consciously mitigate, avoid, enhance, or exploit them and successfully complete the project.

Threats

- The project may deviate from the schedule.
    - Mitigate this risk by maintaining an up-to-date Gantt chart and ensure that internal deadlines are met.
- Poor project team dynamics.
    - Mitigate this risk by maintaining courteous communication between team members and execute professional conflict resolution when a problem or disagreement arises.
- Project sponsor / stakeholders may not approve of selected design.
    - Mitigate this risk by designing the drive system within the customer needs and constraints as closely as possible, as well as communicating progress made throughout the project to acquire customer input.
- Availability of project sponsor / stakeholders may not align with project team, as needs arise.
    - Mitigate this risk by proactively and effectively communicating with the customer, as well as following the customer's request of informing the customer if responses become extensively delayed.
- An element of the design may be controlled by another company or entity, which would prevent the chosen design from being used.
    - Mitigate this risk by researching applicable patents and the patent system to ensure infringement does not occur.
- Existing vendors may not be willing to quote or may take too much time to quote on drive system components.
    - Mitigate this risk by acquiring quotes for each part from multiple sources, as well as follow up on requested quotations early and frequently.
- Design could be more expensive than expected.
    - Mitigate this risk by incorporating simplicity into the design wherever possible, as this often translates to lower costs.
- A focus on reports and course deliverables may distract from project success for the customer.
    - Accept and mitigate this risk by understanding that presentations and reports are a core requirement for the course while still focusing as much as possible on the drive system by implementing appropriate time management and planning ahead.
- Unforeseen weather conditions or stresses may decrease the effectiveness of the design.
    - Mitigate this risk by holding brainstorming sessions and conducting extensive research to develop a thorough understanding of all forces involved.

- Concentration on design by the project team or on business by the stakeholders may interfere with communication between the two parties.
    - Mitigate this risk by having both parties send recurring information updates along with questions and concerns.

Opportunities

- Limited experience in some areas allows the project team to gain experience in those areas, resulting in more competent future engineers.
    - Exploit this risk by distributing work packages in which each person has minimal prior experience.
- Drive system could be patented to apply to other heavy objects (cranes, trains, other wind tunnels).
    - Enhance this risk during concept brainstorming by thoroughly considering design concepts that are unique.

Appendix B – Concept Design

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Concept Development

The design team decided to split the project into five manageable systems: drive system orientation, drive system input, drive system output, braking system, and control system. The five systems each have several concepts that are explained in the following subsections. Within each system, the individual concepts will first be described, then compared and ranked using a matrix format to determine the most suitable options. The control system does not have a scoring matrix because the final control system depends heavily on the other four systems. It will be determined in the design optimization following the methods and considerations outlined in this report.

Drive System Configuration Concepts

The wind tunnel and its platform create a pre-defined design space for the drive system. Figure 1 illustrates the design space in a schematic form that will be used to present the various concepts throughout this report:

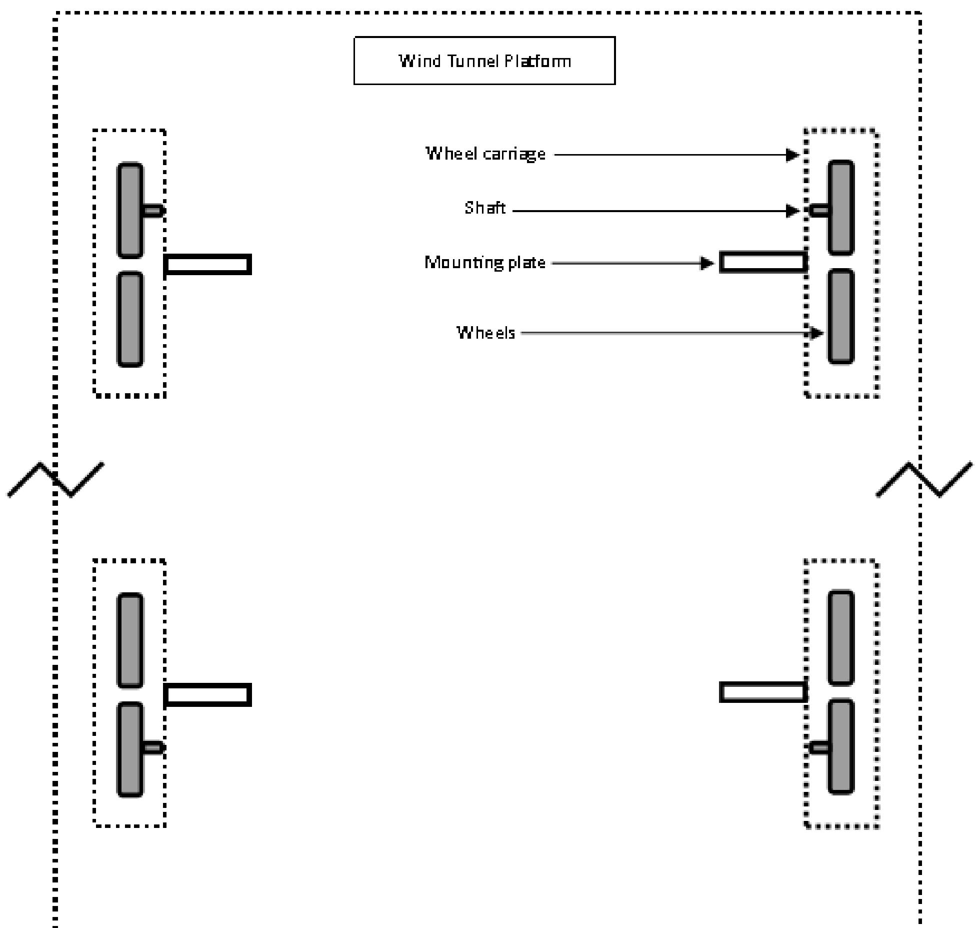

Figure 1: Drive system design space

The labeled parts in Figure 1 are identical at all four of the wheel locations. Based on the existing setup of the wind tunnel and its platform, three different options exist for the physical location of the drive system input and drive system output: the options are single wheel drive, dual wheel drive, and mitre box drive.

1. Single Wheel Drive

The single wheel drive option is the simplest of the three drive system setup configurations. Essentially, the drive system input and drive system output will both be mounted a mounting plate attached to one of the four wheel carriages. Figure 2 is an image of the mounting plate and wheel carriage arrangement:

Figure 2: Single wheel drive design space

The mounting plate provides a location where the drive system can be mounted directly to the platform, which avoids any contact with the ground. Figure 3 shows a schematic of where the drive system will be mounted:

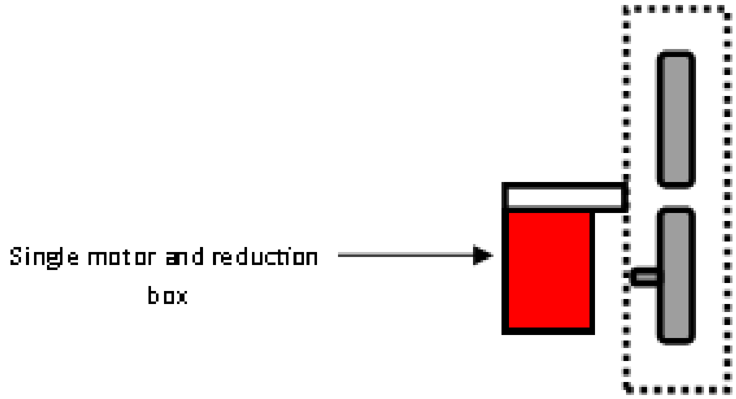

Figure 3: Single wheel drive schematic

This concept is compact and eliminates the need for any long shafts to transfer the torque from the drive system input to the wheel. However, the downside to this concept is that a single wheel drive system will create a moment arm that will strain the platform. If this concept is selected, structural analysis of the platform will be required to determine if the platform can handle the applied moment. The platform structural analysis is beyond the scope of this project, so StandardAero or GE engineering will conduct the structural analysis.

2. Dual Wheel Drive

The dual wheel drive option is similar to the single wheel drive option, but consists of two drive systems applied to two wheels on opposite sides of the wind tunnel platform. Figure 4 illustrates the dual wheel drive concept:

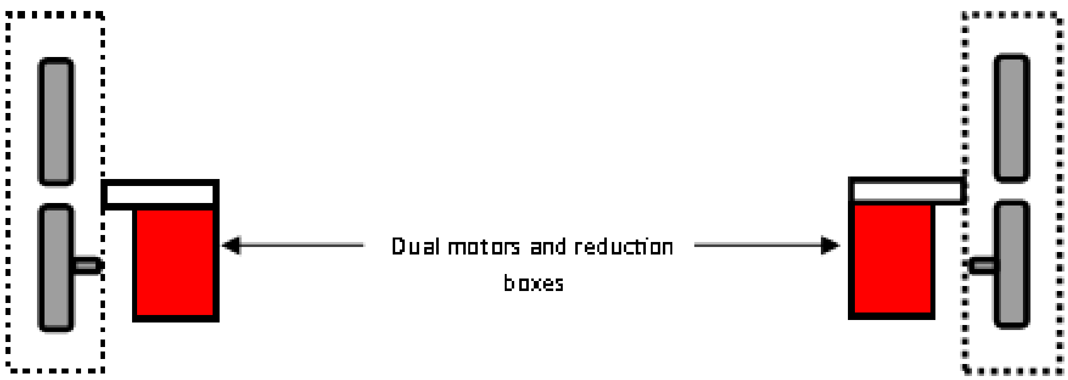

Figure 4: Dual wheel drive schematic

There are two benefits that the dual wheel drive concept has over the single wheel drive concept. The first benefit is that the platform will be driven equally from each side, eliminating the platform-straining moment of the single wheel drive. The second benefit is that smaller motors can be selected because there are two motors doing the work. Alternatively, this system may cost more because there are twice as many components as the single wheel drive concept.

3. Mitrebox Drive

The mitrebox drive setup is a combination of the single wheel drive and dual wheel drive arrangements. The mitrebox transfers the work from one drive system input to two drive system outputs. Figure 5 illustrates this concept:

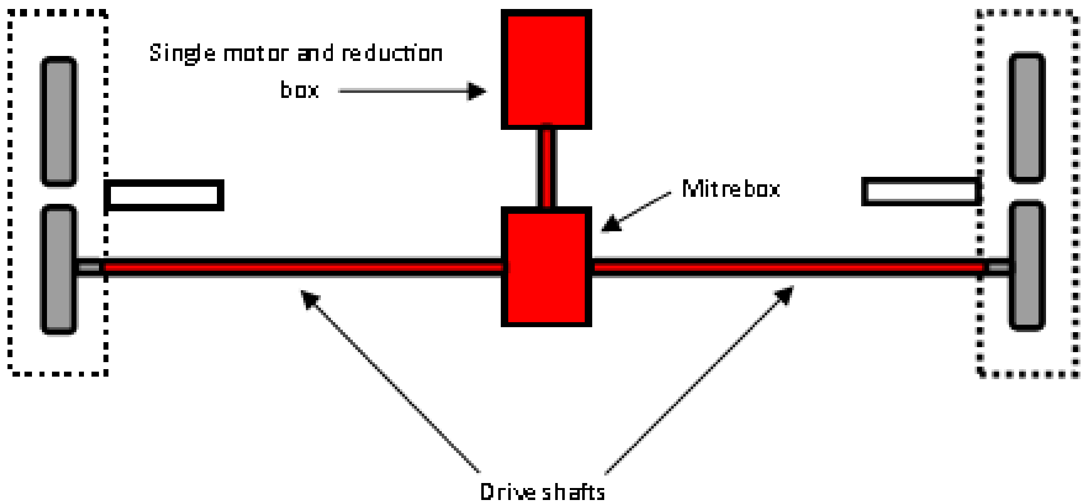

Figure 5: Mitrebox drive schematic

The mitrebox makes use of gears to transfer the rotation from the reduction to the drive shafts and then to the wheels. The mitrebox concept eliminates the need for a second drive system and removes the straining moment created by a single wheel drive system. However, the platform is 40 feet wide from wheel to wheel, so long shafts would be required to transfer the torque. Due to the weight of the structure, it may not be possible to design long shafts that could withstand the required torque without adding too much excess weight, cost, or safety risk.

4. Concept Scoring

Prior to scoring and ranking the drive system orientation concepts, the criteria must be weighted appropriately. The criteria are ranked against each other using the following matrix:

TABLE I: CRITERIA WEIGHTING MATRIX FOR THE DRIVE SYSTEM CONFIGURATION

	Criteria	A Ease of maintenance	B Simplicity of design	C Cost of motor components	D Cost of reduction components	E Strain on structure	F Ability to provide power
A	Ease of maintenance		B	C	D	E	F
B	Simplicity of design			B	B	B	F
C	Cost of motor components				Equal	E	F
D	Cost of reduction components					E	F
E	Strain on structure						F
F	Ability to provide power						
	Total Hits	0	4	1	1	3	5
	Weightings	0.05	0.28	0.06	0.06	0.20	0.35

Using the weightings from the table above, the concepts can be compared against criteria using a score from 1 to 5:

TABLE II: WEIGHTED SCORING MATRIX FOR THE DRIVE SYSTEM CONFIGURATION

		Concepts					
		Single Wheel Drive		Dual Wheel Drive		Mitrebox Drive	
Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Ease of maintenance	0.05	5	0.25	4	0.20	4	0.20
Simplicity of design	0.28	5	1.40	4	1.12	3	0.84
Cost of motor components	0.06	4	0.24	1	0.06	4	0.24
Cost of reduction components	0.06	5	0.30	1	0.06	5	0.30
Strain on structure	0.20	1	0.20	5	1.00	4	0.80
Ability to provide power	0.35	2	0.70	4	1.40	1	0.35
Total Weighted Score		3.09		3.84		2.73	
Rank		2		1		3	
Develop?		No		Yes		No	

Based on the results of the concept scoring table, the dual wheel drive arrangement is the ideal choice for the drive system configuration.

Drive System Input

The motor will provide the power and torque required to overcome the opposing forces to move the wind tunnel in a forward and reverse direction. The client specified that the drive system must be functional in cold weather and have enough power to overcome ice buildup along the track. Three different motor types will be explored, including the electric motor, the hydraulic motor, and a diesel/gas motor. These types will be compared based on many factors, which will be discussed later, to show the best motor that meets all the important needs of the client.

1. Electric Motors

Electric motors convert electrical potential to mechanical motion. Most electric motors operate using the interaction between the motor's magnetic field, which is generated by current moving through an electrical winding.

a) DC Motors

A simple DC motor has a stationary stator that is a permanent magnet and it provides a constant magnetic field. There is an armature, which is a simple coil loop of wires, in between the north and south pole magnets as shown in Figure 6 below.

Figure 6: Electromagnetic force induced on the coils, rotates armature coil *[5]*

The armature is connected to a DC power source; the current starts flowing through the loop of wires and creates an electromagnetic force that is induced on the armature [5]. This results armature rotation, as shown by the red arrows in Figure 6. There is a set of commuter rings that connect with the power source of opposite polarity to ensure that the current always flows in a uni-directional way.

In the real world, there are multiple armatures so the torque will always be constant without any interruptions. Additionally, the armature loops are placed inside slots of a highly permeable steel cylindrical layer. This simply enhances the magnetic flux interaction. There is a large range of different types of motors that have specific purposes and functions.

Generally, DC motors are:

- Initially cheaper than AC motors.
- They have a linear speed-torque curve. The speed-torque curve allows for the DC motor to have a high starting torque and there is potential for a high torque output at low speeds. This could allow the gearbox to have a lower gear ratio.

- The DC motor is much simpler
- The DC motor is easy to install and they are readily available.
- If the force is greater one day because of the ice build-up, the DC motor won't sacrifice speed when the load is greater and will continue to try and rotate.

The DC motor is generally available at many different rated RPM's and torque values. Since the torque and speed are directly related to the amount of voltage supplied and the amount of current passed through the armature, the speed can be varied by having a controller that can vary the supplied voltage. This could provide the benefit of moving the wind tunnel at multiple speeds, which would successfully achieve the specified marginal and ideal speed values that were established in the project needs.

One factor to keep in mind is that a redundancy should be put in place due to a potential for torque overload. The DC motor does have the ability to go in reverse, but an additional controller must be implemented to automatically switch the polarities inside the motor. The main two types of DC motors are brushed and brushless with sub categories in the brushed motor section.

(1) Brushed

The brushed DC motor has brushes to charge the commutator inversely in polarity to the permanent magnet, in turn causing the armature to rotate as shown in Figure 7. The commutator is a set of rings that make contact with the brushes, which connect to opposite poles of a power source to deliver positive and negative charges to the commutator [6].

Figure 7: Split ring that wraps around the axle, called the commutator *[7]*

The following list contains the advantages of the brushed DC motor:

- These motors are very common, which makes them inexpensive.

- They also have a simple two-wire control and require simple controls to operate the motor.
- Since these motors are simple and reliable, but the brushes may need to be replaced if they wear out. This would require additional maintenance, but since the system will only be used once or twice a year, this may not be a massive issue. The brushed DC motor is very robust and can reliably handle rough environments.

There are many disadvantages that come with brushed DC motors, for this project they overpower the advantages, which makes these motors less appealing than the brushless design. The list is all the disadvantages that apply to the brushed design.

- The main objective is to move the wind tunnel so the amount of torque required should be what the motor can output. Since there are brushes that rub against metal inside the motor, friction is produced. When the speed of the motor increases, the brush friction will also increase and result in a decrease of viable torque. This happens in high torque situations, which is exactly the scenario that is presented.
- The brushes limit the speed range of the motor and this could potentially pose a problem if the wind tunnel is wished to have a fast and low speed setting.
- The next potential issue is the electromagnetic interference caused by brush arcing. This creates electromagnetic waves (noise) that travel away from the motor. If any digital system or sensor exists nearby, this “noise” can potentially give false readings to these external systems by “adding” voltage to certain components. If automatic braking is implemented or control systems near the motor, this could potentially hinder these systems and could cause them to fail.

The following four types of brushed motors are discussed below.

Sections I.A.1.a)(1)(a) to I.A.1.a)(1)(d) below discuss four types of different circuitry arrangements possible in a brushed DC motor.

(a) Series Wound

The field winding is connected in series with the rotor winding, which is shown in Figure 8 below:

[redacted]

Figure 8: Series wound DC motor circuit and speed-torque curve *[8]*

This type of brushed motor is used in many heavy-duty applications like with moving cranes, lifts, and hoists. It has a high starting torque, as seen in the Figure 8 above [8]. As the torque increases, the speed falls, which means that if the estimated torque is lower than the actual torque because of ice build up, the wind tunnel will move slower and result in taking longer to get from one end to the other. The moving time is listed in the project needs as an ideal time of 30 minutes from one end of the rail to the other, but the client doesn't have a specified time and it is not a high-prioritized item.

A disadvantage of the series-wound type is that it has poor speed control, which wouldn't be beneficial if a range of speed to move the wind tunnel is required. Also, if the drive system needs to move a small amount to obtain a certain distance away from the engine, being able to have an accurate control over the speed of the wind tunnel would be ideal.

(b) Shunt Wound

Shunt-wound circuitry, shown in Figure 9 below, involves the field winding being connected in parallel with the rotor winding. This means that it can deliver an increased torque without a reduction in speed by increasing the motor current. The starting torque is not as high as the series wound but it is better at dealing with a sudden increase in torque.

[redacted]

Figure 9: Shunt wound DC motor and speed-torque curve *[8]*

(c) Compound Wound

This is a combination of shunt and series wound, where the polarity of the shunt winding is such that it adds to the series fields as shown in Figure 10. Combining these two types creates a unique motor. It has a high starting torque and it runs smoothly if the load varies slightly, which allows for a variance in ice build up on the track. If the opposing force on the wind tunnel is higher than expected, the DC motor will still be able to push through the obstruction. An example of this setup would be elevator conveyors, the total load on the system always varies but the motor has to stay at a constant speed.

Figure 10: Compound wound DC motor and speed-torque curve *[8]*

(d) Permanent Magnet

A permanent magnet is for low torque and precise control applications. The need of high torque eliminated this option without considering any other features.

Figure 11: Permanent Magnet DC motor and speed-torque curve *[8]*

(2) Brushless

The brushless DC motor works differently than the brushed dc motor because it doesn't have the commutator to create that reversal in polarity and this results in creating the torque. The permanent magnet is now the rotor and not the stator as it is in the brushed setup as shown in Figure 12 [8]. The operation of a brushless DC motor is based on the simple force interaction between the permanent magnet and the electromagnet. It works by energizing one of the coils on the rotor, which rotates the rotor because the opposites poles of the rotor and stator are attracted to each other. Each coil is energized in a certain order and the rotor continues to spin.

Figure 12:The stator and rotor of brushless DC motor.

The brushless design has many advantages over the brushed DC motor.

- In the absence of brushes, Hall effect sensors are used to detect the rotor's position, which results in regulating the speed of the motor via the motor current. This means that the brushless design offers a more accurate positioning technique, so if the wind tunnel needs to be positioned at a specific distance, the brushless motor will be able to accomplish that.
- It can produce a higher torque than the brush type and has the ability to maintain or increase the torque at various speeds. This flexibility allows for complete control over the speed and torque needed to overcome any opposing forces.
- Since there are no brushes, this immediately increasing the reliability of the motor and requires less or no maintenance at all. Additionally, it means that they have a longer life expectancy.
- Brushless motors have a high efficiency (85%-90%). Since there would be no power loss across the brushes, this increases the effectiveness of the motor and better heat dissipation. The amount of heat dissipation is directly related to the amount of torque produced.

- The brushless motor has low rotor inertia allowing for the motor to accelerate, decelerate and reverse directions quickly but since the mass of the wind tunnel is so large, the braking system will be more important in stopping the entire structure.
- Another advantage is that the motor has a higher power density because it is more compact and physically smaller than the brushed motor.
- As mentioned previously, the brushes create an electromagnetic interference, which can hinder other electronics near the motor. Since the brushless motor is brushless, it has a very low noise level mechanically and electrically while in operation.

The biggest disadvantage is that this type of DC motor is more expensive to purchase. In the long run, they could potentially cost less due to the low maintenance of the design. Another issue is that they require more complex controllers to run the motor efficiently. These controllers can be potentially more expensive than the motor itself.

b) AC Motors

The biggest difference between AC and DC is that AC waveforms are time varying or oscillatory whereas DC power is a more direct source that is constant and doesn't vary with time. In other words, in a DC motor, the magnetic field is time invariant due to the applied signal. A term that is associated with an AC motor is frequency. Since the AC waveform is a sinusoidal and has amplitudes like shown in Figure 13, the motor will run at a certain frequency. The basic operation of the ac machine is similar in concept to the dc machine where the interaction of magnetic fields will be tied to the rotation of the machines rotor.

Figure 13: Sinusoidal waveform demonstrating an AC power source behavior.

General advantages that the AC motors offer are:

- The ability to have variable operational speeds that the motor functions at.
- They make controlling the acceleration, speed and reversing easier than the DC motor.
- Since they use AC power, it can be connected to a simple wall outlet. The TRDC offers 120V or 208V, single and three phase power outlets. This provides a range of options when picking AC motors, while the DC motor needs an external system to switch the DC power to a sinusoidal value.

There are two main types of AC motors: synchronous and asynchronous (induction). These both function a little differently and both have advantages and disadvantages that will be discussed in detail.

(1) Synchronous

In a synchronous AC motor, the rotation of the rotor is synchronized with the frequency of the supply current and the speed remains constant under varying loads. In other words, the stator and the rotor rotate at the same speed this achieving the synchronization. This works well for moving something on constant speed and high precision. Synchronous motors aren't common in the industry for drive applications. They are generally used as a power factor correction device. The power factor is described as the real power over the total power in the circuit as seen in Figure *14* [9].

Figure 14: Graph showing the power factor and its relationship *[9]*.

So, they can be used in parallel with another system just to increase that power factor, which could result in a higher efficiency overall system. When the motor is faced with a load beyond its limit, the motor just comes to a halt. This can be an advantage and disadvantage because if there is more ice build up at a certain point along the track, the motor would just shut off. Another the hand, this is a torque overload safety system so nothing in the drive system would fail. Some of the advantages and disadvantages related to synchronous motors are explained below:

Advantages:

- In synchronous motors, the speed remains constant, irrespective of the load. This results in a consistent system that can move the wind tunnel in a specific amount of time every operation.
- These motors operate at a higher efficiency than induction motors, usually around 90% at very low speeds. This could potentially eliminate a gearbox if the motor can provide enough torque at low speeds but a high-powered electronic converter is required to generate very low frequencies.
- The synchronous motor is move mechanically stable then induction motors.
- As mentioned previously, load doesn't affect the speed; similarly, torque doesn't get affected as well.

Some disadvantages:

- The synchronous motor is not self-starting and needs an external excitation from a dc motor to start working. This would require another system just to start the motor, which would raise the cost of the drive system. Also, another system to just synchronizes the rotor and stator.
- These types of motors are usually expensive and more complicated compared to the induction motor.
- They can't be used for variable speed applications, as there is no possibility of speed adjustment unless a variable frequency drive system is implemented.
- Collector rings and brushes are required in the motor, which means that it needs more maintenance. As seen above, the client needs are to have a simple system with low maintenance.
- Synchronous motors can't be used for high a starting torque application.

(2) Induction (Asynchronous)

An induction electric motor works on the principle of induction. It uses an electromagnetic field from the magnetic fie stator winding to produce an electric current in the rotor. They are widely used throughout the world in many different industrial applications. This results in a very cheap motor since they have been manufactured and optimized throughout many years. The client wants a cheaper drive system then there previously designed hydraulic motor system, which was expensive. Thus, the induction motor being very common could reduce the cost of each motor. In general, these motors are rugged in construction, which is beneficial since they will be put through harsh Winnipeg winter weather, with temperatures dropping to approximately -40 degrees Celsius.

Induction motors come in either single phase or three-phase configuration. The main difference is that one uses a single-phase power source and the other uses a three-phase power source. The other advantages include:

Single Phase:

Advantages

- Less complicated
- Less expensive for small load applications

Disadvantages

- Lower starting torque
- Used for smaller loads
- Not self-starting

Three-Phase:

Advantages

- Used for high loads
- More efficient
- Can self-start

Disadvantages

- More complicated construction
- More Expensive

There are two distinctive types of AC motors, Squirrel Cage and slip-ring induction motors.

c) Squirrel Cage Induction Motor

The squirrel cage is the most common in industry compared to the slip-ring induction motor. The same working principles as previously mentioned in the introduction paragraph of AC motors are how the squirrel cage induction motor functions. Figure 15 shows the design of the rotor, which looks like a cage.

Figure 15: The rotor inside of the squirrel cage motor *[10]*.

The rotor consists of a cylindrical laminated core having semi-closed circular slots and short circuit at each end by copper or aluminum rings [10]. The rotor slots are not parallel but are skewed. This alone provides some advantages compared to the slip-ring motor. The advantages of this type of motor are as follows:

- This cage reduces the humming of the motor and it provides a smooth torque curve, results in handling heavy loads more effectively
- It increases the rotor resistance due to the increase length of the rotor bar connectors
- They are very rugged and robust in construction; this is beneficial to be in operation during extreme weather conditions.
- Since there is no rings or brushes, the motor requires no frequent maintenance. This makes the system very simple.
- The cost is lower than the slip-ring induction motor but it does have a higher efficiency

There are a couple disadvantages:

- It has a lower starting torque than the slip-ring motor and it can be significantly improved with external systems.
- Speed control by rotor resistance is not possible so another way has to be implemented in order for this to happen.

d) Slip-ring Induction Motor

In the slip ring induction motor, the main difference is that there are some rings and brushes, as in the squirrel cage, that doesn’t exist. The rings are attached to the motor and the brushes are attached to the stator. As the rotor spins, the brushes are always in contact with the rings when they rotate, which is the main area where the electrical power is converted to mechanical work, a diagram can be seen in Figure 16:

Figure 16: Inside view of all the components of a slip-ring AC induction motor.

The main advantages over the squirrel cage induction motor are:

- It can have a high starting torque by adding an external resistance in the rotor circuit.
- Speed control by rotor resistance is possible
- They have a very low starting current
- They have a high over loading capacity and smooth acceleration under heavy loads

These characteristics are appealing because they meet the high ranked needs that were specified by the client in the beginning of the report. The slip-ring induction motor does have disadvantages, which include:

- The initial and maintenance cost are higher due to the presence of rings, brushes and short-circuiting devices.
- It can be used as a variable speed motor but using external resistances makes the motor have a poor speed regulation.
- The efficiency and power factor are lower than the squirrel cage induction motor.

The AC induction motors have many advantages that meet the needs of the client. They meet the electrical requirements that WestCaRD can provide. The slip-ring motor can fill the issue with the low starting torque AC motors usually have. Other than an electrical motor, the next option would be to use a hydraulic motor, which is what the old drive system design had encompassed.

2. Hydraulic Motor

The hydraulic motor converts hydraulic pressure and flow into torque and rotation. It relies on a fluid flow that passes through the motor, which results in the rotation of the output shaft. There are many main types of hydraulic motors but they all have the same working principle as mentioned above.

The old drive system design was using a hydraulic motor setup and it was going to be driving 4 wheels. The client had an issue with this design because it was very expensive and complex. Since the wind tunnel will only be moved once or twice a year, this was not a cost-efficient method. Along with the cost of the motor itself, an additional pump would need to be purchased and powered using an additional power supply. This pump would provide fluid to all of the four motors or there would be a pump for each individual motor. This adds a lot of complexity to the system. Additionally, hydraulic lines would have to be routed throughout the entire wind tunnel structure, which again adds to the total cost of the drive system. There would be more maintenance that would have to be done because now there are more external systems involved. The hydraulic pumps have many advantages but the cost, complexity and client need to simplify the drive system became the more important factors that needed to be considered.

In conclusion, an electric motor would eliminate the extra pump and hydraulic lines needed and that would decrease the cost of the drive system to a reasonable price.

3. Diesel/Gas Motor

Using a diesel or gas motor would be unrealistic and wouldn't work without constant maintenance. Since the drive system needs to operate in extreme weather conditions, this may be an issue when trying to start the engine. Also, multiple external systems would need to be implemented in order for the engine to run correctly. This would cost the most money in the longer run and is the most complicated system out of all of the different types of motors. This system is also not reliable because anything could go wrong since there are so many moving parts inside of an engine. This would cause many delays in the operations at the TRDC, which would potentially push back engine testing.

4. Concept Screening

This initial concept screening was completed to get rid of the concepts that will not be beneficial to further research in any way. It was clear that these concepts had many disadvantages that wouldn't be able to overcome if it were further designed.

TABLE III: DRIVE SYSTEM INPUT CONCEPT SCREENING MATRIX

	Drive System Input Concepts					
Selection Criteria	DC Brushed	DC Brushless	AC Synchronous	AC Induction	Hydraulic Motor	Deisel Motor
Cold weather operation	0	0	0	0	0	-
Operable after 6-month idle time	0	0	0	0	0	-
Noise	-	+	0	0	0	-
Potential Torque	0	+	0	+	+	0
External Components Needed	0	0	-	0	-	-
Ease of maintenance	-	+	0	0	-	-
Cost	+	-	0	0	-	-
Operate at Variable Speeds	-	+	+	+	+	0
Complexity of Controlling	+	0	-	-	-	-
Positives	2	4	1	2	2	0
Neutrals	4	4	6	6	3	2
Negatives	3	1	2	1	4	7
Net	-1	3	-1	1	-2	-7
Rank	3	1	3	2	4	5
Proceed with evaluation?	Yes	Yes	Yes	Yes	No	No

The results show that the hydraulic and diesel motor were the weakest concepts when comparing to the electric motors. The hydraulic was simply too expensive and would require many external systems to operate the motor, for instance an external pump to make the liquid to flow through the motor to get it to produce torque. This extra system would require maintenance. Additionally, this design was previously designed for a four-wheel drive system. The hydraulic was too expensive for WestCaRD to implement so in the case of our design of a two-wheel drive system, the pumps and motor would need to be bigger to overcome the torque required to move the wind tunnel. This would increase the cost and it would potentially cost the same as the old system.

The next motor concept is the diesel engine. This was the worst concept out of all of them. Firstly, it wouldn't be reliable in cold weather; it could potentially not start up when it is needed. It would need to warm up before it could be used; this is to reduce the wear on the engine since every engine has an optimal operating temperature. Secondly, the diesel motor would have many more things to maintain, the employees would have to deal with the fuel, changing oil and keeping an eye on the coolant. The engine would need external control systems to keep it running efficiently and if the speed needs to be varied, an Engine control unit would have to be implemented in order to control the throttle readings and adjust the motors RPM accordingly. The engine would be very load and is unrealistic when the drive system needs to be reliable even after months without use. The client also specified that it has to be simple to use, the diesel engine could potentially add complexity to the system that is not needed.

5. Concept Scoring

This decision matrix will take into account weightings for each specified criteria as seen in the screening matrix.

TABLE IV: DRIVE SYSTEM INPUT CRITERIA WEIGHTING MATRIX

		Cold weather operation	Operable after 6-month idle time	Noise	Potential Torque	External Components needed	Ease of maintenance	Cost	Operational at Variable Speeds	Simplicity of design
	Criteria	A	B	C	D	E	F	G	H	I
A	Cold weather operation		B	A	A	A	A	A	A	A
B	Operable after 6-month idle time			B	D	B	B	B	B	B
C	Noise				D	E	F	G	H	I
D	Potential Torque					D	D	D	D	D
E	External Components needed						F	G	H	I
F	Ease of maintenance							F	F	F
G	Cost								G	G
H	Operational at Variable Speeds									I
I	Simplicity of design									
	Total Hits	7	7	0	7	1	5	4	2	3
	Weightings	0.19	0.19	0.01	0.19	0.03	0.14	0.11	0.06	0.08

TABLE IV shows that the capabilities of the motor to operate in cold weather conditions, to be able to fluently operate after a 6-month idle time and to provide enough torque to be able to move the wind tunnel. Now these criteria weightings are used in the actual scoring matrix of the concepts that were determined to be potential candidate after the screening process.

TABLE V: DRIVE SYSTEM INPUT CONCEPT SCORING MATRIX

		Concepts							
		DC Brushed		DC Brushless		AC Synchronous		AC Induction	
Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Cold weather operation	0.19	5	0.972222222	5	0.972222222	5	0.972222222	5	0.972222222
Operable after 6-month idle time	0.19	5	0.972222222	5	0.972222222	5	0.972222222	5	0.972222222
Noise	0.01	2	0.02	5	0.05	5	0.05	5	0.05
Potential Torque	0.19	4	0.777777778	5	0.972222222	3	0.583333333	4	0.777777778
External Components needed	0.03	5	0.138888889	4	0.111111111	3	0.083333333	3	0.083333333
Ease of maintenance	0.14	3	0.416666667	5	0.694444444	4	0.555555556	5	0.694444444
Cost	0.11	5	0.555555556	3	0.333333333	3	0.333333333	4	0.444444444
Operational at Variable Speeds	0.06	3	0.166666667	5	0.277777778	5	0.277777778	5	0.277777778
Simplicity of design	0.08	4	0.333333333	5	0.416666667	4	0.333333333	4	0.333333333
Total Weighted Score		4.353333333		4.8		4.161111111		4.605555556	
Rank		3		1		4		2	
Develop?		No		Yes		Yes		Yes	

Based on the scoring matrix, it was determined that the DC brushless and the AC induction motor were the two viable options that met the most important needs stated by the client. These will both be taken into account when doing more detailed calculations and from those calculations of torque required if one comes out on top.

Drive System Output

Providing enough torque to the wheels to overcome the opposing forces on the wind tunnel is the primary job of the drive system input. The drive system output is what will transfer the provided torque to the wheels, propelling the wind tunnel forward and backward. The drive system output has two primary purposes: a) the drive system output must transfer the torque from the motor to the wheels with minimal energy losses, and b) transfer said torque at a speed that is acceptable to the client.

The purpose of this section of the report is to first introduce and describe the concepts for the drive system output, along with their strengths and weaknesses. Second, this section will define criteria that the concepts can be measured against, and third, the concepts for the drive system output will be compared and scored against the criteria determined by the client needs.

1. Belt Drive

The first option that will be considered is a drive system created entirely with belt-driven pulleys, or sheaves. Figure *17* shows a representation of a single stage belt reduction:

Figure 17: Simple v-belt drive single stage reduction *[11]*

In the case of a speed reduction, the input will drive the smaller pulley in Figure 17. The belt, or belts, transfer the power from the smaller pulley to the larger pulley. During the power transfer, speed is reduced and torque is increased. To provide enough torque to move the 1.2-million-pound wind tunnel, multiple stages of speed reduction must be used. Figure 18 shows multiple stages of speed reduction using belts:

[redacted]

Figure 18: Multi-stage belt drive reduction *[12]*

The two pulleys in the center of Figure 18 are attached to the same shaft and have the same rotational velocity. Multiple stages of speed reduction will allow the input speed to be slowed down enough to move the wind tunnel at a safe speed.

A belt drive system has several advantages [13]:

- Belt drives are simple, economical, and have low weight.
- The drive shafts do not need to be parallel to one another, due to the flexibility of most belts.
- Belt drives are relatively quiet when compared to other types of drive systems.
- Vibration is damped in a belt drive, thus increasing machinery life because load fluctuations are shock absorbed.
- Belt drives do not require lubrication and cost less to maintain.
- Flat belt drives can have efficiencies of up to 98% [14].

There are also limitations to using a belt drive setup to move the wind tunnel. Disadvantages of belt drives include the following:

- Power transmission of belt drives is typically limited to 370 kilowatts, or 496 HP [13].
- Belts can slip and stretch, which causes inconsistent angular velocities.
- Operating temperatures for belts are typically restricted to -35 to 85°C.
- Performance of belt drives decreases when the belt speed is less than 1000 ft/min [15].
- The recommended maximum reduction ratio for a v-belt drive is 6:1 and 7:1 for a cogged belt.

A drive system consisting of strictly belt drives would need to have some specialty parts implemented to account for the low temperatures that the drive system will experience. Also, the power transmission capability of a belt drive system may be too low for the purposes of this project.

2. Chain Drive

A chain drive system is very similar to a belt drive system. The reduction concepts work in the very same manner as a belt drive, with the major exception being that the chain drives a sprocket via interaction between a roller and the sprocket tooth. Figure 19 shows a simple chain drive:

Figure 19: Simple single stage reduction chain drive *[15]*

Like belt drives, chain drives have a tight side and a slack side. However, there are certain advantages that chain drives have that belt drives do not:

- Chain drives do not slip like belt drives, so velocity ratios remain constant and power transmission efficiency is high.
- Velocity ratios can be as high as 8:1 [13].
- Chain drives do not require a lot of maintenance.
- Power transmission capability is higher in chain drives than it is in belt drives.

There are also weaknesses to implementing a chain drive concept:

- Chain drives require lubrication, which add to the complexity of the design. [15]
- Chain drives can be noisy at high speeds.
- Chain drives can occupy a lot of space due to the size of the sprockets needed for large velocity reduction ratios.

Finding suitable heavy-duty chain to move the 1.2-million-pound wind tunnel could prove to be a difficult process, but the advantages of this system could plausibly outweigh the disadvantages if a suitable option can be determined.

3. Spur Gear Drive

Just as a belt or chain concept would need multiple stages of speed reduction for this project, so would a spur gear drive system. A simple single stage spur gear drive system looks similar to Figure 20:

Figure 20: Simple single stage spur gear interaction *[15]*

A multiple stage spur gear drive system would be congruent with Figure 18, in the sense that the large gear from the first stage would share a shaft with a small gear from the second stage. Some of the advantages of using a spur gear reduction drive are as follows:

- A spur gearbox can be a compact design and take up a small amount of the design space, while providing the required torque.
- Reduction ratios as high as 60:1 can be obtained [13].
- Only radial forces are generated during gear movement. The lack of thrust forces reduces the cost and complexity of bearings because thrust bearings are not required.
- Very high efficiencies and constant velocity ratios are attainable.

As with all systems, there are weaknesses to using a spur-gear-type system as well. The weaknesses are summarized below:

- Gears require a housing and a form of lubrication. This design consideration increases the cost and complexity of the system.
- Spur gears are noisy at high speeds. If the input to the drive system spins at a high velocity, the movement of the wind tunnel could be a noisy process due to the gears.
- Gear reduction design is an iterative process and includes many calculations. The time spent developing a gear reduction could be used more efficiently on a simpler design.

Spur gears are the simplest gear design and are commonly used in industry to transfer large amounts of power. A spur gearbox may provide the best opportunity to fit within the design constraints of this project.

4. Helical Gear Drive

A helical gear drive design is similar to a spur gear drive in some respects. Figure 21 shows how helical gears engage:

Figure 21: Helical gear interaction *[15]*

Both spur and helical gear drive types have high speed reduction ratio capabilities, but helical gears provide advantages over spur gears in the following ways:

- Helical gears are quieter than spur gears due to gradual tooth interaction.
- Since the teeth engage more gradually, stress concentrations are reduced.
- Smaller helical gears can be designed for a given power-transmitting capacity when compared to spur gears.

Despite being stronger in some areas, helical gears have some drawbacks that spur gears do not have:

- Helical gears are slightly costlier to manufacture than spur gears.
- Helical gears are more difficult to machine than spur gears and require some advanced machinery, so some vendors may not be able to produce helical gears.
- Helical gears create thrust forces during operation, thus require thrust bearings.

5. Planetary Gear Drive

A planetary gear drive concept can be created from either spur gears or helical gears. Essentially, a planetary gear drive is created by spinning a sun gear, around which revolve planet gears, which drive a ring gear at a slower speed. A planetary gear system is illustrated in Figure 22:

Figure 22: Single stage planetary gear reduction *[16]*

Planetary gear systems can be highly advantageous if a large gear reduction is needed in a small space. This type of system is typically used when multiple output speeds are required because certain stages can be locked, allowing only the remaining stages to reduce the speed [17]. Some strengths of planetary gears drives include:

- High power transmission efficiency

- Coaxial – all gear stages are centred along one axis. This arrangement is ideal for vehicle transmissions.
- Typical ratio of 9:1 per stage

However, there are weaknesses associated with planetary gears, as well. The following items are the most prominent disadvantages of using planetary gears:

- Noisy at high speeds
- Very high design and manufacturing cost
- Difficult to assemble
- Long gearboxes (due to stacked stages).

Ultimately, a planetary gearbox could serve a purpose if complexity does not interfere with the functionality of the design. High cost may be an issue, but the ability to transmit power at a high efficiency could be very useful for the wind tunnel drive system.

6. Concept Screening

In order to develop a method to compare concepts to one another, some grading criteria must be determined. At the outset of the project, client needs were developed based on input from the client, as well as suggestions from the design team. The following list is a set of customer needs that are of critical importance and apply specifically to the drive system output. These needs will be used as criteria to measure the plausibility of the drive system output concepts:

- The drive system output must be operable after being unused for a minimum of six months.
- The drive system output must be simple to maintain.
- The drive system output is a simple design.
- The cost of the drive system output is kept to a minimum.
- The noise produced by the drive system output is minimal.
- The drive system output transmits the power from the input in an effective manner.
- The rotational speed reduction per reduction stage is maximized.

Taking into consideration the needs of the client and the additional needs that are required for only the drive system output, a screening process can be conducted to eliminate the least viable concepts. TABLE VI is a screening matrix that uses the spur gear concept as the reference concept; all other concepts are compared to the spur gear concept as better or worse in each of the selection criteria:

TABLE VI: DRIVE SYSTEM OUTPUT CONCEPT SCREENING MATRIX

	Drive System Output Concepts				
Selection Criteria	Belts Only	Chains Only	Helical Gear Only	Planetary Gears	Spur Gear Only: Reference
Cold weather operation	-	0	0	0	0
Operable after 6-month idle time	-	0	0	0	0
Noise	+	+	+	-	0
Power transmission efficiency	-	+	+	+	0
Obtainable speed ratio	-	-	0	0	0
Ease of maintenance	+	-	0	0	0
Cost	+	+	-	-	0
Complexity of design	+	+	-	-	0
Positives	4	4	2	1	0
Neutrals	0	2	4	4	8
Negatives	4	2	2	3	0
Net	0	2	0	-2	0
Rank	4	1	2	5	2
Proceed with evaluation?	No	Yes	Yes	No	Yes

The results of the screening process indicate that the chain only concept, the helical gear only concept, and the spur gear only concept are the most viable options and will proceed to the scoring evaluation process further on in this section of the report. The planetary gear concept is too complex and will be too costly compared to the other concepts. Despite scoring well in the screening matrix, the belt concept was eliminated for a few reasons:

- The operation temperature of the drive system may be below -35°C, and a belt cannot operate below that temperature.
- Belts can stiffen and operate improperly if neglected. Since the wind tunnel may remain stationary for 6 months or more at a time, a belt design would run the risk of failure to due lack of use.
- The obtainable speed ratio that belts can achieve, combined with the possibility of slippage, does not measure up to that of the other concepts.

7. Concept Scoring

Prior to scoring and ranking the remaining drive system output concepts, the criteria must be weighted appropriately. The criteria are ranked against each other using the following matrix:

TABLE VII: CRITERIA WEIGHTING MATRIX FOR THE DRIVE SYSTEM OUTPUT

	Criteria	A	B	C	D	E	F	G	H
		Cold weather operation	Operable after 6-month idle time	Noise	Power transmission efficiency	Obtainable speed ratio	Ease of maintenance	Cost	Simplicity of design
A	Cold weather operation		B	A	A	A	A	A	A
B	Operable after 6-month idle time			B	B	B	B	B	B
C	Noise				D	E	F	G	H
D	Power transmission efficiency					E	D	G	D
E	Obtainable speed ratio						F	G	E
F	Ease of maintenance							G	F
G	Cost								H
H	Simplicity of design								
	Total Hits	6	7	0	3	3	3	4	2
	Weightings	0.20	0.25	0.05	0.10	0.10	0.10	0.12	0.08

Using the weightings from TABLE VII above, the concepts can be compared against criteria using a score from 1 to 5:

TABLE VIII: WEIGHTED SCORING MATRIX FOR THE DRIVE SYSTEM OUTPUT

		Concepts					
		Spur Gears Only		Helical Gears Only		Chains Only	
Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Cold weather operation	0.20	5	1.00	5	1.00	3	0.60
Operable after 6-month idle time	0.25	4	1.00	4	1.00	3	0.75
Noise	0.05	3	0.15	4	0.20	4	0.20
Power transmission efficiency	0.10	4	0.40	5	0.50	4	0.40
Obtainable speed ratio	0.10	5	0.50	5	0.50	4	0.40
Ease of maintenance	0.10	3	0.30	3	0.30	4	0.40
Cost	0.12	3	0.36	2	0.24	4	0.48
Simplicity of design	0.08	4	0.32	2	0.16	4	0.32
Total Weighted Score		4.03		3.9		3.55	
Rank		1		2		3	
Develop?		Yes		Yes		No	

Based on the results of the concept scoring table, the spur gear and helical gear designs are the most ideal choices for the drive system output. However, the chain-only drive system is not far behind. This concept analysis only included isolated systems, but the optimal solution may be a combination of any of the top three concepts. Therefore, the spur gear and helical gear designs will be analyzed further in an attempt to keep the system simple, but a chain drive could be implemented at one or more stages as well. Only further design of the drive system output can fully rule out any of the top three options.

Braking System

To address several of the needs set by WestCaRD and StandardAero, the final system design is required to incorporate a method to bring the platform and wind tunnel to a complete stop from maximum velocity. This also entails preventing the platform from moving during intentional periods of rest, as well as an emergency braking option. The specific needs for the braking portion of this design are:

- Operable after 6-month idle time
- Simple to maintain
- Simple to operate
- Operable in extreme weather
- Long working life
- Braking control (accuracy)
- Platform immobilization
- Emergency braking
- Cost
- Efficiency

To gain a better understanding of the braking options available, research was conducted and the various options were compared against the needs of WestCaRD and StandardAero to evaluate the viability of each system. The types of braking have been categorized into two types: electrical (non-contact) and friction braking.

1. Electrical (Non-Contact) Brake Systems

These systems do not rely on frictional contact between two surfaces, instead relying on either a magnetic or electrical drag force that creates a braking torque.

a) Eddy Current Braking

This type of braking utilizes the drag force created between an electromagnetic field and a nearby conductive object in motion [18]. As the conductive material passes through the stationary magnets, the magnetic flux through the sheet changes. As the section of metal passing through approaches the magnet, the magnetic field increases and creates a counter-clockwise eddy current due to Faraday's Law of Induction. Once it passes under and moves away from the magnet, the opposite takes effect creating a clockwise eddy current in the material. This effect is demonstrated below in Figure 23:

Figure 23: A stationary magnet creating an eddy current in a rotating disc

Due to Ampere's Circuital Law, each of these currents creates a counter magnetic field (blue arrows) which opposes the change in magnetic field, creating a braking force exerted on the material at velocity. Certain advantages come from a lack of contact between the rotating and braking material:

- No friction surface to wear out and require replacing
- Minimal to no maintenance as the only moving part is the conductive material
- Can adjust the power of the braking force simply by adjusting the current through an electromagnet
- Creates a smooth braking force free of shaking

While these properties are very desirable, the nature of eddy current braking relies on the frequency or velocity of the object in motion. This translates to disadvantages such as:

- Ineffective at low speeds, unable to completely brake platform
- Ineffective as an emergency brake
- Unable to provide immobilization at zero velocity

This method of braking may be able to be used near the motor as it will be moving at much higher frequency than the platform wheel, but could only be used as the initial method of braking and an auxiliary system would be required for emergency braking.

b) Regenerative Motor Braking

Once the electrical supply driving an electric motor is shut off, the motor continues to spin due to the inertia and kinetic energy of the object that it is attached to [19]. The inertia of the system is translated to the motion of the motor, generating electricity that can be transferred and stored back into the electrical system. Figure 24 demonstrates the flow of energy in regenerative braking:

[redacted]

Figure 24: The flow of electricity in a regenerative braking system

Some advantages of regenerative braking are:

- Utilizing the braking force to create and store energy for later use
- Integrated braking with use of the motor
- No friction surfaces to replace
- Less likely to seize due to severe weather

This translation of inertia and energy is what causes the braking force, creating a very efficient system since little energy is wasted in comparison to other braking methods. While being very efficient, its effectiveness relies on the inertia of the overall system, decreasing its capability as the speed of the platform decreases. This creates disadvantages similar to eddy current braking such as:

- Ineffective at low inertia, unable to completely brake platform
- Ineffective as an emergency brake
- Unable to provide immobilization at zero velocity
- Increased wear on the motor

Also, since the system is used very infrequently, the regenerative capability of the system would be heavily minimized.

c) Rheostatic (Resistance) Motor Braking

Resistance braking works under very similar principles as regenerative braking. The motor has its supply turned off, turning it into a generator from the inertia of the object in motion. In this case though, the energy generated by the system is dissipated through a series of large resistors in the form of heat. This system is simpler because it does not require storage for the recuperated energy, which makes it far less efficient. The strengths and weaknesses of regenerative braking are reflected in resistance braking.

Strengths:

- Integrated braking with use of the motor
- No friction surfaces to replace
- Less likely to seize due to severe weather

Weaknesses:

- Ineffective at low inertia, unable to completely stop platform
- Ineffective as an emergency brake
- Unable to provide immobilization at zero velocity
- Increased wear on the motor

These weaknesses indicate that a secondary braking system would be required if this was used as an initial stopping method.

d) Plugging Motor Braking

As another form of motor braking, plugging braking relies on the reverse of voltage across the armature to create a torque opposite to the motion of the system, creating a braking effect [20]. With plugging braking, instead of shutting off the supply voltage, it is reversed. This in turn makes the motor want to turn in the opposing direction to the inertia, thus creating the reverse torque. A simple depiction of the difference between motoring and plugging braking is shown in Figure 25 and Figure 26:

Figure 25: Typical motor circuit

Figure 26: A plugging braking circuit with reversed polarity

To limit the current to a safe amperage moving across the armature, a resistor is incorporated into the system when the supply voltage is switched direction. Some strengths of plugging braking include:

- Not relying on inertia to create braking force
- Can bring platform to a complete stop
- Double the effective braking torque compared to regenerative or resistance braking
- Integrated braking system
- No friction surfaces to replace
- Less likely to seize due to severe weather

This method is much more likely at being applicable to the wind tunnel platform, as it still works at extremely low speeds. It is not without weaknesses though, which include:

- Requires additional power input to perform braking, decreasing efficiency
- Puts extra strain on the electric motor
- Cannot immobilize platform at rest

The major downside of this system is that the motor must be disengaged right before or at standstill otherwise it will reverse the platform in the opposite direction. As such, it will also require an additional method of braking to supplement it.

2. Friction Braking

These systems rely on the contact of two surfaces to induce a braking torque and slow the system to a stop, or to hold the platform in place.

a) Electromagnetic Braking

This braking system works off the principle of electromagnetic force. A current is passed through a wire generating a magnetic field, attracting a ferritic material [21]. This can be used in electromagnetic brakes in a few different configurations using the magnetic force to either release the braking material away from the armature or force it to make contact. In either scenario, the brake can be forced in the opposing direction, sometimes using a mechanical force such as springs to achieve the opposite effect. As shown in Figure 27 below, when current passes through the coil and the magnet is powered, it depresses the spring and keeps the armature away from the rotating friction braking material. When the current is stopped, the magnetic force disappears, the spring force takes over, and pushes the armature against the friction material. This is known as a Power Off system and can be used as a failsafe as when the power is shut off, the brake engages.

Figure 27: A power-off electromagnetic friction brake

This braking system presents many great advantages, including:

- High braking efficiency
- High torque output for small configurations
- High adaptability
- Relatively simple
- Minimal maintenance required
- Can be used for immobilization

However, electromagnetic braking is typically better suited for high cycle applications, although it can be fitted to meet the current requirements. Other disadvantages include:

- Higher cost
- Requires replacement of friction material
- Braking strength also limited by input voltage

Ultimately, the viability of this braking system will depend on cost and overall braking capability at the voltage that we have supplied.

b) Hydraulic Braking

Hydraulic braking operates using a liquid system that is put under pressure to force friction plates against a surface in motion [22]. Most commonly known in use in automotive vehicles, hydraulic brakes are known for many benefits:

- Versatility
- Relatively inexpensive
- Reliable if properly maintained
- Capable of high braking torques
- Can perform platform immobilization

They consist of a hydraulic pump or generator to produce the pressure, a reservoir for the working fluid, hydraulic lines, and a caliper to depress the friction surface (disc or drums) as shown in Figure 28 below.

Figure 28: Typical automobile hydraulic braking system

Hydraulic systems also have a few disadvantages such as:

- Subject to failures due to leaking hydraulic lines
- Require a method of producing hydraulic thrust if none are present
- Complicated
- Require regular maintenance
- Friction pads to replace

Due to being so common, many options exist which could lead to a cost effective and well performing hydraulic option.

c) Air Braking

This is very similar to hydraulic braking, but instead of a liquid working fluid, air is used to depress the friction pads [23]. Instead of a master cylinder, a compressor is used to create the working pressure and a reservoir tank. This minimizes some of the risks of having a hydraulic system, although creates a few disadvantages as well.

Air braking strengths:

- Maintain pressure even with the presence of multiple leaks, ensuring braking system is capable
- Relatively inexpensive
- Reliable if properly maintained
- Can perform platform immobilization

Air braking weaknesses:

- Less braking power per unit size
- Longer braking distances than hydraulic
- Susceptible to acquiring moisture in the system and freezing in cold climates
- Friction pads require replacing
- Require additional pressure system (compressor)

While it is useful to have a working fluid that can be constantly replenished, the effective braking power and distance is worsened by utilizing a compressible fluid, making the system less appealing than hydraulic.

d) Band Brakes

Band brakes use a friction band or strap that can be tightened around a cylindrical object to produce a braking torque, as shown in Figure 29:

Figure 29: Band brake wrapped around a rotating drum

Strengths [24]:

- High torque output for relatively low input
- Cheap
- Simple
- Compact

Weaknesses:

- Braking power can decrease significantly with heat
- Friction band requires replacing
- Subject to surface effects (corrosion, water, oil)

Band brakes are typically used for smaller, low inertia applications such as chainsaws, go-karts, or mini-bikes. It is unlikely to be effective at stopping such a high inertia object unless it can be appropriately scaled.

e) Linear Rail Brake

This friction braking technique is a rail specific system that consists of a braking “shoe” that bears down on the rail to produce the braking torque [25]. In comparison to the other typical hydraulic or air brake options, this is suited especially for rails in a linear braking fashion as shown in Figure *30*:

Figure 30: Small rail brake system

Strengths:

- Easy installation
- Stiff, vibration free braking
- Large contact area for low footprint
- Does not require lubrication or maintenance
- Can provide immobilization and emergency stop

Weaknesses:

- Usually on small scale systems
- Friction material requires replacing
- May not be able to fit current rail system

This braking system can be controlled using either air, hydraulic, or mechanical means. It is ideal for power-off, emergency stop, and holding applications. While it is typically used for smaller application it could possibly be scaled to meet the client’s requirements.

3. Concept Screening Matrix

As there are several options that may be applicable to the final design, a screening matrix was used to remove the contenders that didn't match with the appropriate needs. As shown in TABLE IX below, eddy current brakes, both types of dynamic motor braking, and band brakes were not worth evaluating. Eddy current brakes were used as a reference. TABLE IX shows that there is a negative for eddy current brakes and both types of dynamic braking methods under the platform immobilization and emergency braking needs. This was done to demonstrate the importance of these two criteria, as these systems were completely incapable of completing these tasks themselves.

TABLE IX: BRAKING CONCEPT SCREENING MATRIX

	Braking System Output Concepts								
Selection Criteria	Eddy Current Brakes (Reference)	Dynamic (Regenerative) Motor Braking	Dynamic (Rheostatic/Resistive) Motor Braking	Plugging (Reverse Polarity) Motor Braking	Disc or Drum Brakes (Hydraulic)	ElectroMagnetic Brakes	Air Brake	Band Brakes	Linear Rail Brake
Operable after 6 month idle time	0	0	0	0	-	-	-	-	-
Simple to maintain	0	0	0	0	-	-	-	-	-
Operable in extreme weather	0	0	0	0	-	-	-	-	-
Long working life	0	0	0	0	-	-	-	-	-
Braking control (accuracy)	0	+	+	+	+	+	+	+	+
Platform immobilization	-	-	-	+	+	+	+	+	+
Emergency braking	-	-	-	+	+	+	+	0	0
Simple to operate	0	0	0	0	0	0	0	0	0
Cost	0	+	+	+	+	0	+	+	+
Efficiency	0	-	-	0	+	+	0	-	+
Positives	0	2	2	4	5	4	4	3	4
Neutrals	8	5	5	6	1	2	2	2	2
Negatives	2	3	3	0	4	4	4	5	4
Net	-2	-1	-1	4	1	0	0	-2	0
Rank	8	7	6	1	2	3	4	9	5
Proceed with Evaluation?	No	No	No	Yes	Yes	Yes	Yes	No	Yes

4. Criteria Weighting Matrix

Before the final scoring of the concepts against the criteria, the criteria had to be weighted for this section to prevent the results from being skewed due to less important client needs. The criteria were set against each other and the number of times a criterion was chosen over its competitor determined its weight in the final scoring.

TABLE X: BRAKING CRITERIA WEIGHING MATRIX

		Operable after 6 month idle time	Simple to maintain	Operable in extreme weather	Long working life	Braking control (accuracy)	Platform Immobilization	Emergency braking	Simple to operate	Cost	Efficiency
	Criteria	A	B	C	D	E	F	G	H	I	J
A	Operable after 6 month idle time		A	C	A	A	F	G	A	A	A
B	Simple to maintain			C	D	B	F	G	B	B	B
C	Operable in extreme weather				C	C	C	G	C	C	C
D	Long working life					D	F	G	H	D	D
E	Braking control (accuracy)						F	G	H	I	E
F	Platform immobilization							G	F	F	F
G	Emergency braking								G	G	G
H	Simple to operate									H	H
I	Cost										I
J	Efficiency										
	Total Hits	6	4	8	4	1	7	9	4	2	0
	Weightings	0.133	0.089	0.178	0.089	0.022	0.156	0.200	0.089	0.044	0.000

5. Concept Scoring Matrix

Here, the braking concepts selected from the initial screening phase were scored against the concepts weighted in the previous section. Each criterion was scored on a scale from one to five and multiplied by that criteria's weight. The scores were summed and totaled to determine the best candidate.

TABLE *XI*: BRAKING CONCEPT SCORING MATRIX

		Concepts									
		Plugging Motor Brakes		Hydraulic Brakes		Electromagnetic		Air Barakes		Linear Rail Brakes	
Criteria	Weight	Score	Weighted Score	Score	Weighted Score	Score	Weighted Score	Score	Weighted Score	Score	Weighted Score
Operable after long periods of inactivity	0.13	5	0.67	3	0.40	4	0.53	3	0.40	3	0.40
Simple to maintain	0.09	5	0.44	3	0.27	4	0.36	3	0.27	3	0.27
Operable in extreme weather	0.18	5	0.89	4	0.71	4	0.71	2	0.36	4	0.71
Long Working Life	0.09	4	0.36	3	0.27	4	0.36	3	0.27	3	0.27
Braking Control (Accuracy)	0.02	2	0.04	4	0.09	4	0.09	2	0.04	4	0.09
Can hold platform stationary	0.16	0	0.00	5	0.78	5	0.78	4	0.62	5	0.78
Provides emergency braking	0.20	3	0.60	4	0.80	4	0.80	2	0.40	4	0.80
Simple to operate	0.09	5	0.44	5	0.44	5	0.44	5	0.44	5	0.44
Cost	0.04	5	0.22	4	0.18	3	0.13	4	0.18	3	0.13
Total Weighted Score		3.67		3.93		4.20		2.98		3.89	
Rank		3*		2		1		4		3*	
Develop?		Yes - if it can be incorporated easily and is effective		Yes		Yes		No		Yes - if it can be scaled	

As shown in the final matrix, the most promising concepts include electromagnetic, plugging, hydraulic and linear rail brakes. As accurate calculations have not been completed to determine the magnitude of braking power and rate that will be required, some of these may not be a suitable fit for the final design. Linear rail braking must be able to be scaled to meet the braking requirements and sourced from a manufacturer. Plugging braking by itself cannot meet the requirements of the customer and may only be incorporated to complement the chosen primary braking system. Electromagnetic and hydraulic braking options demonstrate the ability to meet all the criteria and shall be thoroughly investigated in phase 3 of this project.

Control System

The development of the concepts for the control system for the drive system began with establishing needs specific to the control system itself, based on the needs of the drive system itself:

1. The design should be simple, sturdy and easily operable.

- This need assumes that the control may be operated outside, where it is also going to face the harsh conditions faced by the drive system. Additionally, if humans will be operating the system, they may be wearing gloves or other clothing that make is challenging to operate intricate controls.

2. The placement of any control system requiring direct human input should be easily accessible, but not in a way that obstructs other operations.

3. The control system must be able to perform all the following actions:

a) Enable the movement of the drive system at two speeds (fast/slow).

b) Control the function of the emergency and normal braking systems.

c) Protect the drive system from unintended operating conditions.

The final control system is heavily dependent on the drive system selected, so at this stage of development, the goal was to establish the required functions of the control system, and how they could be satisfied. The result of this section is a series of methods to control the functions that the drive system is to perform, that can then be refined when each specific component of the drive system has been decided upon. TABLE XII summarizes the various options considered for each requirement, which is listed on the left-hand side of the table.

TABLE *XII*: CONTROL OPTIONS FOR EACH REQUIREMENT.

Requirements	**Options**				
1. Simple design	*1.1* Few components	*1.2* Clear labels	*1.3* Auto locking parts		
2. Sturdy design	*2.1* Tough materials	*2.2* Structurally sound	2.3 Control covers		
3. Easily Operable	*3.1* Big switch	*3.2* Big knob	*3.3* Lever		
4. Accessible but non-obstructive placement	*4.1* On platform walkway	*4.2* Attached to stairs leading onto the platform	*4.3* Operator held wireless box	*4.4* Wireless control unit in indoors control room	*4.5* Connection via adapter

5. Two speed drive	*5.1* Control the speed of the motor	*5.2* Vary the reduction mechanism			
6. Normal braking system	*6.1* Power reduction to motor	*6.2* Engage other braking system			
7. Emergency braking system	*7.1* Movement sensors	*7.2* Weight sensor	*7.3* Position sensor		
8. System protection	*8.1* Fuse	*8.2* Protective overload relay	*8.3* Temperature sensor	*8.4* Three phase monitor relays	

The options considered for the first three requirements are not mutually exclusive and will be kept in mind during the optimization of the final design and applied on a case by case basis to the control systems various functions.

1. Designing a simple system can be accomplished by leaving out unnecessary components (1.1) and clearly labeling components and controls (1.2). Also, parts that lock into their intended position when they are within a certain range will simplify the use of the system (parts such as switches and knobs) (1.3).
2. Using materials that can resist the environment and weather conditions (2.1), as well as a structure that is sturdy and resistant to their environment (2.2) will be key to having a design that is reliable and requires little maintenance. Having covers over controls (2.3) will also prevent foreign matter from entering the control mechanism and causing issues (snow/ice).
3. Easily Operable controls such as big switches (3.1), knobs (3.2) and levers (3.3) with reasonable space between them should be used over smaller controls that are tightly spaced to reduce the risk of pressing the wrong control when operating with gloves, these commands could also contain built in timers as the wind tunnel will be moving quite slowly and setting the control to automatically "turn off" after a certain time may be beneficial in situations where an operator is responsible for multiple things at once .
4. The placement of the control system is quite flexible at this point in the design, and will be narrowed down as the rest of the drive system comes into place.
    4.1 The first location considered is on the walkway, near the stairway leading onto the wind tunnel platform. This location has the advantage of being on an already walk friendly area, close to the door leading into the wind tunnel containers. It will also move with

the wind tunnel (as it will be self-contained on the wind tunnel). A potential issue is that the user must be on the wind tunnel to operate it, which may be a safety concern.

4.2 The second location is attached to the stairs leading onto the wind tunnel. This location is easily accessible from the ground, but is still contained on the platform, relatively close to the drive system location.

Figure 31 roughly shows the locations mentioned in 4.1 and 4.2.

Figure 31: Two tentative locations for the Control unit

4.3 Another option would be to have the drive system operated by remote control, such that the operator could carry a control box that could be operated from anywhere in the vicinity of the wind tunnel, adding a large amount of flexibility to where the wind tunnel can be operated from.

4.4 The fourth considered option is the placement of the control unit in a general control room in the facility, this could be a wireless or wired control unit. Having the control unit indoors may lengthen its useful life. It would be beneficial for the operator to be able to see the wind tunnel as he is operating the drive system, because of the size of what he is moving.

4.5 The final idea is to have the control unit be able to “plug in” to the drive system whenever it is needed, this would allow for the control unit to be stored indoors for most of the time (when it is not needed) where it will be protected from the elements, and only taken out when the wind tunnel needs to be moved.

The following requirements have options that are quite simple and depend on the components selected for the drive system.

5. The speed control of the drive system can happen in two main ways, directly controlling the speed of the motor (for example by controlling the voltage supply using pulse width modulation in the case of a DC motor), which is option 5.1, and secondly to control the transmission mechanism, option 5.2.
6. Controlling the normal braking mechanism by controlling the speed of the motor, or engaging a separate braking system, and the method used here will also depend largely on the braking method selected for the drive system.
7. The emergency braking system can be an extension of the braking system above, however an automatic sensory braking system can be considered as a failsafe, these sensors will detect when the wind tunnel is within minimum braking distance from the end of the track and automatically engage the brake.
   7.1 Firstly, using an optical sensor that picks up a marker located on the wheels, and triggers the brake. In this case visibility may be an issue on days with snow.
   7.2 Secondly, A weight (pressure) sensor under the track that triggers the brake when it detects a certain pressure, since the wind tunnel is extremely heavy, not much else would be able to trigger the sensor by accident.
   7.3 Thirdly, a position sensor located in the direction that the wind tunnel is moving, triggering the brake when the wind tunnel gets within a certain distance. Here again, weather conditions may pose an issue.
8. The last requirement of the control system is to protect the drive system from operating conditions that could harm it. Solutions for this include the use of fuses (8.1), protective overload relays (8.2), temperature sensors (8.3) or three phase monitor relays to protect from

voltage loss (8.4). The selection of these systems will depend on the motor chose and how effectively they can be integrated.

A separate option would be to select an exterior manufacturer to produce the control system based on the requirements. This could result in a sturdy and reliable system; however, the cost could end up being relatively high, especially because the control requirements for this system are relatively simple.

Concept Combinations

This section provides a combination of the various concepts outlined above. Only the options selected for further development in each section were compared and the final combinations examined. TABLE XIII provides a summary of the options with the top choice listed first followed by the second and the third choices.

TABLE XIII: OVERVIEW OF THE SYSTEMS BEING CONSIDERED FOR THE DRIVE SYSTEM.

System	**Options**		
Motor	1.1 DC brushless	1.2 AC induction	1.3 AC synchronous
Reduction	2.1 Spur gears	2.2 Helical gears	2.3 Chains
Braking	31. Electromagnetic braking	3.2 Plugging motor breaking (with hydraulic backup)	

Combining the top options for each system results in a drive system consisting of a DC brushless motor driving a reduction system of mainly spur gears and an electromagnetic braking system (Drive system 1).

Other concepts, formed from a combination of one secondary option and two top choices are:

- Drive system 2: AC induction motor – Spur gear reduction– Electromagnetic braking
- Drive system 3: DC brushless motor – Helical gear reduction – Electromagnetic braking
- Drive system 4: DC brushless motor – Spur gear reduction – Plugging motor braking

None of these concept options have incompatible features and all systems can be incorporated into the dual drive configuration, which was deemed the optimal configuration. Therefore, it was not necessary to examine options that included the third best option for each system.

There is very little difference between the use of a DC motor over the AC motor, since the major detractor of the DC motor is its cost and the AC motor is its size and external components needed. Neither of these attributes were deemed significant enough to justify one motor over the other at this stage, so neither will be eliminated.

The use of a reduction system relying mainly on spur gears is considered to have enough of an advantage over helical gears, due to their simplicity (making them easier to source) and higher efficiency. The main advantage of the helical gears is their relatively quiet operation and their ability to transfer more power than a spur gear of the same size. Neither of these two factors are high priority considerations in this design and thus don’t carry much weight.

Finally, the plugging motor braking might would require an additional hydraulic brake to act as a backup, which increases the complexity and offsets cost benefits of the plugging motor brake alone over the electromagnetic break. Since both braking methods should be able to effectively brake the wind tunnel, electromagnetic brakes will be used as the braking system.

The two designs are the Drive system 1 and Drive system 2 consisting of a gearbox with spur gears and an electromagnetic braking system, with either a DC brushless motor or an AC synchronous motor. The drive system will have dual configuration, as shown in section 2, which the team found to be the optimal configuration.

The control system will be designed around the control methods available for the motor and braking system. The speed variation will be accomplished by changing the speed of the motor (a capability which both considered motor types have), rather than having a separate gearing system that costs more and introduces complexity into the system. The control for the braking system will engage the separate (electromagnetic) braking system. Other aspects of the control system, as outlined in section 0, will be incorporated into the final design so that the project needs are met.

The control system for the automatic sensory breaking system will be further discussed in conjunction with the client, and designed to be incorporated in the testing facility with minimal disturbance to testing.

Appendix C – Final Design Detailed Calculations

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1 Introduction

This appendix provides the detailed analysis of each component of the proposed wind tunnel drive system. Sample calculations are provided when necessary for the motor design, braking design, gear design, and shaft design.

2 Motor Design

The tread surfaces of the wheels supporting the platform are 36 inches in diameter. The diameter of the wheel can be used to determine how many revolutions of the wheel are required to move the platform from one end of the rails to the other, a distance that measures 95 feet in length:

$$C = \pi D = \pi(36in) = 113.10in = 9.425ft$$

$$Wheel\ revolutions = R = \frac{95ft}{9.425ft} = 10.080\ revolutions$$

The number of wheel revolutions is critical in determining the overall gear ratio (OAGR) and the rotational velocities of the motors.

The clients require that the platform move at one speed. The metrics determined in Phase I of the project include an ideal travelling speed of 2 m/min or 6.562 ft/min. The ideal speed can be used to determine the amount of time it will take the platform to move the 95-foot distance:

$$t_{travel} = dv = (95ft)\left(6.562\frac{ft}{min}\right) = 14.48min$$

The travelling time can be used to determine the wheel RPM and the OAGR. The wheel RPM can be determined as follows:

$$RPM_{wheel} = \frac{10.080rev}{14.48min} = 0.696RPM$$

Assuming an arbitrary motor speed of 550 RPM, the OAGR can be determined for the travelling speed:

$$OAGR = \frac{550RPM}{0.696RPM} = 789.9{:}\,1$$

The travelling speed will be the used to calculate the motor torque, which will result in the highest tractive force to move the entire structure.

The average wheel load of the structure was found using load ratios that were provided to the design team by the client. Figure *1* shows the loads at the wheel carriage locations, based on the mass of the old wind tunnel (800,000 lbs):

[redacted]

Figure 1: Loads at the wheel carriage locations

Locations (3) and (4) in Figure *1* see the largest load, so these will be used as the two drive locations. The larger load on these two wheel locations will result in a larger available tractive force. The error associated with the load at these locations (31,000 lbs) was added to the total load (240,000 lbs and 220,000 lb). The average load was taken from location (3) and (4) and was assumed to be the load per wheel carriage for the entire structure. Since wheel carriage has two wheels, the load was adjusted accordingly. Since the loads in Figure *1* are from the old wind tunnel (800,000 lbs), the values were scaled higher to account for the heavier weight of the new wind tunnel structure (1.2 million lbs). The load ratios of each location of the old structure were calculated as follows:

$$Location\ (1)\ Load\ Ratio = \frac{Location\ Load}{Total\ Structure\ Load} = 0.2152$$

The above calculation was repeated for each wheel carriage location. The ratios were used to determine the new location loads based on 1.2 million pounds.

$$New\ Load\ at\ Location\ (1) = (Location\ (1)\ Load\ Ratio)(New\ Total\ Structure\ Load)$$
$$= (0.2152)(1200000) = 273227.85lb$$

The above calculation was repeated for each wheel carriage location. TABLE *I* shows the new loads for each location. The new total loads also include the error loads associated shown in Figure *1*:

TABLE I: TOTAL LOADS AT EACH LOCATION

Location #	Load
1	273.23 kips
2	259.04 kips
3	395.56 kips
4	365.18 kips

Using Locations (3) and (4) as the drive wheel carriages (keeping in mind that there are two wheels per location), the average load per drive wheel can be calculated as follows:

$$Average\ Load\ per\ Wheel = \frac{395.56 + 365.18}{4\ wheels} = 190.18 kips$$

The starting resistance force per wheel was calculated to determine the force required to move the wind tunnel, excluding all other potential factors. The starting resistance is approximately 2-2.5 times higher than the rolling resistance [1]. The friction coefficient of railroad steel wheels on steel rails was found to be approximately 0.00164ft [2], which was used in the calculation instead of a value with no units:

$$Rolling\ resistance\ per\ wheel = \frac{f * L_w}{r} = \frac{0.00164 * \ 190183.544}{1.5} = 207.99\ lbs$$

where f is the coefficient of friction, L_w is the average load per wheel, and r is the radius of the steel wheel. The starting resistance per wheel is determined as follows:

$$Starting\ resistance\ per\ wheel = 207.99\ lbs * 2.5 = 519.97\ lbs$$

Additional tractive force effort will be required if the ground is at an incline. The following calculation is the grade resistance of the entire structure, where 1.2 million pounds was taken as the total load, but since the wind tunnel structure is situated on perfectly flat ground, no inclination was accounted for:

$$Grade\ resistance = T_L \sin\left(\frac{\theta\pi}{180}\right) = 1200000 \sin\left(\frac{0\pi}{180}\right) = \ 0\ lbs$$

where TL is the total load of the structure and θ is the inclination angle.

The acceleration force is an additional parameter that determines the amount of force required to reach the desired speed in a certain amount of time. The following calculation assumes the wind tunnel structure starts from rest. An acceleration time of 15 seconds was assumed to be acceptable:

$$F_{acceleration} = \frac{T_L s}{gt} = \frac{(1200000)(6.562)}{(115920)(0.25)} = 271.72\ lbs$$

Where s is the maximum traveling speed in ft/min, g is the acceleration due to gravity in ft/min^2, and t is the minimum traveling time to reach the desired speed in min. The traveling time to reach the desired speed can be varied, but 15 seconds was chosen to reduce the torque required from the motor.

2.1 Wind Forces

Since the structure has a large footprint, the wind force plays a role with the required motor torque.

$$F = A \cdot P \tag{1}$$

Where F is the force due to the wind, A is the projected area of the wind tunnel, and P is the pressure of the wind. The force acting on the structure due to the wind was approximated with the help of the section on calculating wind forces on buildings and large structures in the National Building Code of Canada 2015. According to the NBCC 2015 [3]:

$$P = I_W \, qC_o \, C_g \, C_p \tag{2}$$

Where I_w is the importance factor for the wind load: Normal (low hazard to human life in case of failure), which has a value of one. The factor q is the reference velocity pressure, which is 0.45 kPa for Winnipeg for a one in 50-year wind [3]. This represents a roughly 96 km/h wind. Since this application will not be operating continuously, the team decided to use a wind speed value of 45 km/h, which gave a reference velocity pressure of 0.21 kPa.

C_e is the exposure factor. For open terrain:

$$C_e = \left(\frac{h}{10}\right)^{0.2} \tag{3}$$

Where h is the height of the wind tunnel. The estimated height of the wind tunnel was 9 m from the provided drawings. Therefore:

$$\left(\frac{9}{10}\right)^{0.2} = 0.98$$

C_g is the gust effect factor. For the main structural members, C_g = 2.0

C_p the external pressure coefficient, which has a value of one in this case.

Using Equation (2) for the wind pressure:

$$P = 1(0.21)(0.98)(2.0)(1) = 0.44125 \, kPa$$

The projected area (dimensions of most unshielded side, estimated from provided drawings as a rectangle) can be found as follows:

$$A = height \cdot width = 9.1(15.8) = 143.78 \, m^2$$

Using Equation (1), the wind force was determined:

$$F = A \cdot P = 143.78 \cdot 0.882 = 59.13 \, kN = 13292.66 \, lbf$$

2.2 Ice Forces

To acquire an approximation of the forces that would need to be overcome due to ice buildup on the tracks, several assumptions were made. The first was that the platform would only overcome the ice by breaking it, or the wind tunnel would not move. The overall weight of the structure was assumed to be too large for the wheels to drive over top of the ice. These assumptions mean that the ability to overcome the ice would be dependent on the compressive strength of the ice.

Other assumptions made were related to wheel dimensions, the actual compressive strength of the ice, the amount of ice, the number of wheels in contact with ice, and the percent of contact the ice had with the rail wheel. Due to so many variables being present, a medium and worst-case scenario were considered. The medium-case scenario assumes that all ice above the surface of the rail has been manually removed with only partial coverage below the wheel and ice with a moderate compressive strength. The worst-case scenario assumes one inch of ice on top of the rail, as well as increased coverage and ice that has a higher compressive strength, which is true for colder temperatures [4].

To approximate the surface area of ice that would be blocking the wheel, the wheel dimensions were assumed as shown below in Figure *2*, where "Y" is the height of ice above the rail and "Φ" is the angle of coverage. The radius of the wheel is known (r=18 inches) so that did not need to be assumed for these calculations:

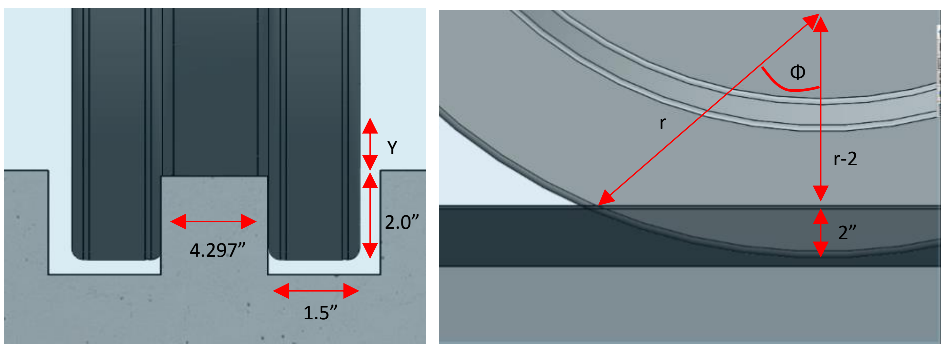

Figure 2: Assumed Wheel Dimensions

The surface area of ice was split into two sections: below the rail and above the rail. The angle of affected area above and below the rail were determined using trigonometry, then those sections were multiplied by the total circumference to determine the length of circumference blocked by ice, as demonstrated by the following calculations:

$$\phi_{below\ rail} = \cos^{-1}\left(radius - \frac{wheel\ below\ rail}{radius}\right) = \cos^{-1}\left(18 - \frac{2}{18}\right) = 27.3°$$

$$\phi_{above\ rail} = \cos^{-1}\left(radius - \frac{wheel\ below\ rail + ice\ above\ rail}{radius}\right) - \phi_{below\ rail}$$
$$= \cos^{-1}\left(18 - \frac{2+1}{18}\right) - 27.3° = 6.3°$$

$$Length\ Affected_{below\ wheel} = 2\pi r\left(\frac{27.3}{360}\right) = 8.58in$$

$$Length\ Affected_{above\ wheel} = 2\pi r\left(\frac{6.3}{360}\right) = 1.98in$$

The affected lengths were then multiplied by the width of the tread surface of the wheel to obtain the surface area. The area above the rail also included the middle of the wheel, although this was assumed to be the same radius as the outer sections of the rail to simplify calculations. Therefore, total surface area was found as follows:

$$Surface\ Area_{medium\ case} = 8.58(2 * 1.5) = 25.70\ in^2$$

$$Surface\ Area_{worst\ case} = (8.58 * 2 * 1.5) + 1.98 * \left(2 * 1.5 + 4\frac{19}{64}\right) = 40.12\ in^2$$

To determine the required force to break the ice, an assumed compressive strength and coverage percent was used. For the medium-case scenario, approximately 1,000 psi was assumed, which was a common breaking strength found for ice at approximately -10°C, with 50% coverage [4]. For the worst-case scenario, a breaking strength of 2,000 psi was assumed, which reflected ice closer to a temperature of -25°C, with 75% coverage:

$$F_{medium} = SA * coverage * Breaking\ Strength = 25.70 * 0.50 * 1000 = 12{,}849\ lbf\ per\ wheel$$

$$F_{worst} = 40.12 * 0.75 * 2000 = 60{,}179\ lbf\ per\ wheel$$

The force estimations shown may vary greatly depending on coverage and the compressive strength of the ice. Compressive strength of ice can reach 4,300 psi at -40°C [5].

Since ice has such a detrimental effect on the available tractive force and the resistive force, assumptions had to be made after consulting with the client for appropriate levels of ice that the drive system could overcome. Ice on top of the track was assumed to be only 0.1 inches and coverage would be only 20 percent on the top and sides of the rail, as it can be manually removed. Additionally, a breaking strength of 1500 psi was used, as it reflected the strength of ice at a more realistic temperature of between -15°C to -20°C. Using the same equations as above with the new ice coverage values, there is a required breaking strength of 8181 lbf per wheel, or a total of 65453 lbf of resistive ice. As this is well below the worst-case conditions for ice on the rails, it is very important to keep the track clear if the drive system will be used in winter.

The wind forces are based on average conditions that arise around 10 times a year. The ice force was based on the available tractive force and dry rail conditions. Anything above those forces will result in wheel slip or the motor won't have enough power.

2.3 Required Tractive Force

The total tractive force required was determined by adding all the resistances and forces on the wind tunnel structure. The starting resistance was multiplied by eight because there is a total of eight wheels contributing to the friction force. The total tractive force was then multiplied by a friction resistant factor of 1.15, which takes into account bearings and axle friction, as well as other unspecified mechanical resistances:

$$\begin{aligned} Total\ required\ tractive\ force &= (SR * 8 + GR + AF + WF + IC) * 1.15 \\ &= (519.97 * 8 + 271.72 + 13292.66 + 65453) * 1.15 = 94503.69\ lbs \end{aligned}$$

There are two driving wheels to move the structure, so the tractive force required per wheel is:

$$Total\ required\ tractive\ force\ per\ wheel = \frac{TTF}{2} = \frac{94503.69}{2} = 47251.85\ lbs$$

The required torque at the wheel to set the structure in motion is:

$$Required\ torque\ at\ each\ wheel = (TRTF\ per\ wheel) * r = 47251.85 * 1.5 = 70877.77\ lbs$$

The motor torque is found by treating the gear reduction system as a whole and using the total gear ratio and gear transfer efficiency:

$$Required\ Motor\ torque = \frac{TW}{OAGR * GE} = \frac{70877.77}{789.9 * 0.9224} = 97.28\ ft\ lbs$$

The required total tractive force is based on all the forces that the wind tunnel will experience. To ensure that the drive system can propel the wind tunnel, the total available tractive force must be equal to or greater than the required tractive effort.

Finally, to determine the required horsepower of the motor, the following formula was used:

$$Required\ Motor\ Horsepower = \frac{MT * RPM}{5252} = \frac{97.28 * 550}{5252} = 10.19\ HP$$

2.4 Available Tractive Force

The available tractive effort is based on the dead load of the entire structure. That dead load was broken down to determine the dead load of each wheel. The adhesion coefficient is a factor that will constantly change based on the weather conditions. The introduction on water, snow or ice on the rail system will drastically change the adhesion coefficient, which will greatly reduce the available tractive force and thus, the potential of wheel slip is present. TABLE *II* shows three adhesion coefficients that are assumed values based on rail conditions [6]:

TABLE II: ADHESION COEFFICIENTS BASED ON RAIL CONDITIONS

Rail Conditions	Adhesion Coefficient
Dry Conditions	0.3-0.5
Wet	0.2
Ice	0.05-0.1

An adhesion coefficient of 0.3 was taken as the value used to determine the available tractive force in dry conditions. The client specified that the wind tunnel would be moved mostly in fall and spring and rarely during winter. Also, ice would be removed as completely as possible, so the conditions will be mostly dry to wet in the worst of cases. The available tractive force per wheel was determined using the adhesion coefficient, *Ad*, and the average dead load per wheel, *Lw*:

$$Available\ Tractive\ force\ per\ wheel = Lw * Ad = 190183.544 * 0.3 = 57055.06\ lbs$$

The available tractive force represents the force past which the wheels will slip, which will prevent the wind tunnel from moving and possibly cause damage. In comparison with the required tractive force, the available tractive force per wheel is higher; this means with the current assumed conditions, the wind tunnel will be able to be moved safely without any wheel slip. The available tractive force result is the maximum amount of traction a wheel will have with the current weight of the wind tunnel. The motor that will be purchased will be based on the horsepower and torque required for 57055.06lbs of tractive force. Given the available tractive force per wheel, the required torque at each drive wheel, the required torque at the motor, and the required horsepower of the motor can all be finalized:

$$Required\ Torque\ per\ wheel = AT\ per\ wheel * r = 57055.06 * 1.5 = 85582.59\ lbs$$

$$Required\ Torque\ at\ motor = \frac{TW}{OAGR * GE} = \frac{85582.59}{789.9 * 0.9224} = 117.46\ ft\ lbs$$

$$Required\ Horsepower\ at\ motor = \frac{MT * RPM}{5252} = \frac{117.46 * 550}{5252} = 12.3\ HP$$

3 Braking Calculations

Another requirement of the system was to be able to bring the platform and wind tunnel to a safe stop, as well as keep the platform stationary when it is not being moved. The following calculations describe the required braking force, holding force, and the max and actual deceleration rate of which the system is capable.

3.1 Required Braking Force

Initially, the system's brake was going to be mounted as close to the wheel as possible to minimize impact on the gear reduction and motor. The following calculations were done to estimate the amount of braking force that would be required to dynamically stop and hold the platform.

An assumed top speed of 1.93 m/min, or 0.03217 m/s, from a 15-minute travel time from end to end was used, as well as an assumed stopping distance of 1 foot, or 0.3048 meters. From these assumptions, the required average deceleration was determined [7]:

$$a = \frac{v_2^2 - v_1^2}{2s} = \frac{0.03217^2 - 0^2}{2 * 0.3048} = 0.0016977 \frac{m}{s^2}$$

From the deceleration, the required braking force was found using the total mass of the wind tunnel platform, which was assumed to be 1,200,000 lbs, or 544,311 kg:

$$B_F = ma = 544311 * 0.0016977 = 924.5\ N$$

B_F is the required braking force, or friction force. To bring the platform to a stop in the required distance, a coefficient of friction must be accounted for to determine the required clamping force that must be supplied by the brakes [8]. A coefficient of friction of 0.35 was used for the braking material [9]:

$$F_{clamping} = \frac{F_{friction}}{\mu_f} = \frac{924.5}{0.35} = 2641.42\ N = 593.8\ lbf$$

To calculate the required torque from the clamping force, the effective radius at which the braking force is being applied is required. This was calculated as follows, first in inches and then afterwards converted to feet, since most torque applications in industry are foot-pounds [8]:

$$r_e = \frac{1}{3}\left(\frac{D^3 - d^3}{D^2 - d^2}\right) = \frac{1}{3}\left(\frac{18^3 - 10^3}{18^2 - 10^2}\right) = 7.19\ inches$$

Where r_e is the effective radius, *D* is the outer diameter of the brake pad, and *d* is the inner diameter for the brake pad. The effective radius value was changed continuously as different brake options were considered, with the final dimensions of the pad determined to be 7.19 inches.

Finally, the required torque was found by multiplying the required clamping force by the effective radius of the braking system, and multiplied by 3, a standard safety factor rating for braking applications [10]:

$$Torque = F_{clamping} * r_e * 3 = 593.8 * \left(\frac{7.19}{12}\right) * 3 = 1067\ lbft$$

At this point, potential suppliers were contacted and an estimated price was sourced. Brakes of this physical size and braking requirement were found to be very costly, so the design team developed a solution that could substantially decrease the applied braking torque by placing the braking system before the motor. Rear-motor mounted brakes allow the gear reduction to amplify the braking torque to the value required.

To calculate the braking requirement for this mounting placement, inertial values were used instead of required deceleration. First, the total inertia of the system that the brake must dynamically stop was determined, including the motor inertia, gear inertia, and finally, the inertia created through the horizontal motion of the platform and wind tunnel.

First, the inertial value of the platform was found by using the assumed horizontal traverse speed (1.93 m/min or 6.333 ft/mi) as well as the rotational speed of the motor (500 rpm) to determine the inertia, as it would be reduced through the gear train [11]:

$$I_{platform} = Weight * \left(\frac{V_{horizontal}}{2 * \pi * rpm_{brake}}\right)^2 = 1{,}200{,}000 * \left(\frac{6.333}{2 * \pi * 500}\right)^2 = 4.87\ lb * ft^2$$

The inertial values of the motor and gearbox were acquired from data sheets from prospective suppliers to acquire a more realistic expectation. From Nord Drive Systems, the motor's inertia was rated at 0.16 kg-m^2 or 3.80 lb-ft^2 [12]. Bonfiglioli was contacted to determine the inertial load of their planetary gearboxes, but could not provide values as it was deemed minimal for their typical applications. A value of 15 lb-ft^2 was assumed, as other gearboxes had shown to have similar inertia to the motors, although more was added as a precaution against a larger than usual gear reduction. Finally, the brakes that were quoted had an inertia of 0.3 lb-ft^2. Ultimately, a total inertia of 23.97 lb-ft^2.

An approximate braking time was required before the dynamic and static braking torque could be determined. The time was determined using the assumed horizontal speed and braking distance previously stated:

$$t = \frac{distance}{velocity} = \frac{1}{6.333/60} = 9.5\ seconds$$

From the total system inertia and the stopping time, the dynamic braking torque can be found as follows:

$$T_d = \frac{I_{total} * rpm_{brake}}{308 * t} = \frac{23.97 * 500}{308 * 9.5} = 4.10\ ft * lbf$$

Dynamic torque was then converted to static torque [13]:

$$T_s = \frac{T_d}{0.8} = \frac{4.10}{0.8} = 5.12\ ft * lbf$$

A safety factor was incorporated as well to adjust for any inaccuracies in the assumptions or any discrepancies from the actual performance of the brake. A safety factor of two was chosen to ensure the brake would function, but to not overload the gear or motor system and put unnecessary stress on the platform or its components from sudden braking. The recommended braking force required was calculated to be 10.24 ft-lbf.

4 Gear Design Calculations

To complete the spur-type planetary gear analysis and design, the American Gear Manufacturers Association (AGMA) standard was followed [14]. The gears were designed to have bending and contact strengths greater than the allowable stresses determined by the following calculations. To demonstrate the calculation process, the spur gear-and-pinion stage 1 and the spur planetary stage 2 values will be used in this appendix.

4.1 Design Inputs

The following inputs were determined from the calculations for the motor:

- Power required: 12HP
- Motor speed: 581.72RPM
- Overall gear ratio (OAGR): 789.95:1
- Average gear ratio per stage: 5.30:1
- Wheel speed: 0.736RPM

Other inputs are also required prior to beginning calculations. To simplify the process, a 20° pressure angle, full depth teeth, and a diametral pitch of 5 teeth/inch were assumed. The next step was to determine the gear specifications.

4.2 Gear Specifications – Stage 1

Stage 1 is a gear-and-pinion style speed reduction. The first step in determining the gear specifications for this stage was to determine the number of teeth on the gears. To prevent interference, the following formulas were used:

$$N_P = \frac{2k}{(1+2m)\sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)\sin^2\phi}\right) \tag{4}$$

$$N_G = \frac{N_P^2 \sin^2\phi - 4k^2}{4k - 2N_P \sin^2\phi} \tag{5}$$

The value of k is 1 for spur gears. Using the average gear ratio per stage as the value for m and 20° as the pressure angle, ϕ, the minimum number of teeth on the pinion, N_P, is 16 and the maximum number of teeth on the gear, N_G, is 101. The first reduction stage will have the lowest torque value in the system, so the ideal solution was to maximize the gear ratio in the first stage. The number of teeth for the pinion was selected as 16 and the number of teeth on the gear to be 98, giving a gear ratio of 6.125:1. Re-substituting this value into equation (4) confirms that 16 teeth on the pinion is suitable to prevent interference.

With the diametral pitch, P_d, set at 5 teeth/inch and the teeth numbers selected, the pinion and gear pitch diameters, d_P and d_G, were determined:

$$d_P = \frac{N_P}{P_d} = \frac{16}{5} = 3.2in$$

$$d_G = \frac{N_G}{P_d} = \frac{98}{5} = 19.6in$$

The gear ratio, GR, and the pinion speed, n_P, were used to calculate the gear speed, n_G, as follows:

$$n_G = \frac{n_P}{GR} = \frac{581.72rpm}{6.125} = 94.98rpm$$

Using the rotational speed of the pinion, the pitch-line velocity of the gear mesh, V, was calculated as follows: $V = \frac{\pi d_P n_P}{12} = \frac{\pi(3.2)(581.72)}{12} = 487.34fpm$

From the pitch-line velocity, the transmitted load, W_t, could be determined using the power input, P, of 12 horsepower: $W_t = 33000 \times \frac{P}{V} = 33000 \times \frac{12}{487.34} = 812.57lbf$

The transmitted load is the force that acts perpendicular to the centreline of the gear tooth, which causes bending. Figure *3* shows where W_t is applied on the gear:

Figure 3: Transmitted load, Wt, for a gear and pinion [14]

The transmitted load is the primary input to determining the bending stress, σ_t, and contact stress, σ_c. The equations to calculate the bending stress and contact stress are as follows:

$$\sigma_t = \frac{W_t K_o K_v K_s P_d K_m K_B}{FJ} \tag{6}$$

$$\sigma_{t,all} = \frac{S_t Y_N}{S_F K_T K_R} \tag{7}$$

$$\sigma_c = C_p \sqrt{\frac{W_t K_o K_v K_s K_m C_f}{d_P F I}} \tag{8}$$

$$\sigma_{c,all} = \frac{S_c Z_N}{S_H K_T K_R} \tag{9}$$

Each of the variables in equations (6) through (9) are explained in the sections to follow. Each variable was found for the stage 1 pinion and gear. The following sections of this appendix demonstrate how each variable was calculated.

4.2.1 Overload Factor, K_o

The overload factor was determined using TABLE *III*:

TABLE III: POSSIBLE OVERLOAD FACTORS (ADAPTED FROM [15])

	Driven Machine		
Power Source	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light Shock	1.25	1.50	2.00
Medium Shock	1.50	1.75	2.25

The power source is uniform since it is an electric motor. The driven machine falls under the category of "Heavy Shock" [15]. Therefore, the overload factor for all reduction stages, including stage 1, is 1.75.

4.2.2 Dynamic Factor, K_v

The dynamic factor for stage 1 was calculated using the following formulas:

$$K_v = \left(\frac{A + \sqrt{V}}{A}\right)^B \tag{10}$$

$$A = 50 + 56(1 - B) \tag{11}$$

$$B = 0.25(12 - Q_v)^{\frac{2}{3}} \tag{12}$$

The value for the AGMA quality number, Q_v, was set at 5, which is an average value of commercially produced gears. Using the pitch-line velocity, *V*, of 487.34fpm, the value of the dynamic factor was determined to be 1.363 for stage 1.

4.2.3 Size Factor, K_s

The size factor for stage 1 was determined using the following formula:

$$K_s = 1.192\left(\frac{F\sqrt{Y}}{P_d}\right)^{0.0535} \tag{13}$$

Typically, face widths, denoted as *F* in equation (13), are between three and five times greater than the circular pitch of the gear mesh, therefore, an estimate for the face width can be calculated with the following formula:

$$F = \frac{4\pi}{P_d} \tag{14}$$

An estimate for the face width of the stage 1 gears is therefore 2.513, or rounded to the nearest half-inch, 2.5in. The Lewis form factor, *Y*, is determined by the number of teeth on the pinion or gear using TABLE IV:

TABLE IV: VALUES OF THE LEWIS FORM FACTOR, Y, FOR PRESSURE ANGLE OF 20° AND FULL-DEPTH TEETH (ADAPTED FROM [14])

Number of Teeth	Y	Number of Teeth	Y
12	.245	28	.353
13	.261	30	.359
14	.277	34	.371
15	.290	38	.384
16	.296	43	.397
17	.303	50	.409
18	.309	60	.422
19	.314	75	.435
20	.322	100	.447
21	.328	150	.460
22	.331	300	.472
24	.337	400	.480
26	.346	Rack	.485

Therefore, with 16 and 98 teeth on the pinion and gear, respectively, the Lewis form factors were determined to be 0.296 for the pinion and 0.446 for the gear.

Given the Lewis form factors, diametral pitch, and face width, the size factors for the pinion and gear were determined to be 1.112 and 1.124, respectively.

4.2.4 Load Distribution Factor, K_m

The load distribution factor is dependent on several other factors. The equation for the load distribution factor is as follows:

$$K_m = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right) \tag{15}$$

Assuming uncrowned teeth, the C_{mc} factor is one. The C_{pf} factor is as follows, given that the face width of the gear mesh is between one and 17 inches:

$$C_{pf} = \frac{F}{10d_P} - 0.0375 + 0.0125F \tag{16}$$

The C_{pm} factor depends on the placement of the gear mesh along the shaft. Assuming the gear mesh is straddle-mounted (i.e. mounted between shaft bearings), the value for C_{pm} is one, given that $S_1/S < 0.175$ in Figure 4:

Figure 4: Gear placement relative to bearings for determining the value of C_{pm} [14]

The equation for C_{ma} for commercial, enclosed gearing is as follows:

$$C_{ma} = 0.127 + 0.0158F - 0.930 \times 10^{-4}F^2 \tag{17}$$

Finally, the factor C_e is 0.8, assuming that the gear-and-pinion compatibility is improved by lapping. Given each factor determined above, the value of the load distribution factor for stage 1 was determined to be 1.205.

4.2.5 Rim-Thickness Factor, K_B

The rim-thickness factor for all gears in the gearbox can be determined using Figure 5:

Figure 5: Chart for determining the rim-thickness factor [14]

The rim-thickness factor for this design was assumed to be one. Normally, large gears are made from rimmed blanks to reduce the weight of the gearbox. In rimmed gears, if the rim thickness is not sufficient enough to provide enough support to the tooth, the gear can fail through the rim as opposed to at the base of the tooth [14]. Assuming that the gear manufacturer takes the rim thickness into account, there was no need to apply a rim-thickness factor to this design.

4.2.6 Geometry Factor for Bending Strength, J

To determine the geometry factor for bending strength, given a pressure angle of 20° and full-depth teeth, Figure 6 can be used:

Figure 6: Chart for determining the geometry factor for bending stress [14]

For a pinion of 16 teeth mating with a gear of 98 teeth, such as in stage 1 of the gearbox, the geometry factor for bending strength was determined to be 0.271. The bending strength geometry factor for the gear was determined to be 0.423.

4.2.7 Bending Stress, σ_t

The bending stress for the pinion and gear differ due to differing geometry factors and size factors. Generally, the pinion has the higher bending stress due to having a lower geometry factor and a lower size factor. Using equation (6), the bending stress estimate for the gear and pinion are:

$$\sigma_{t,pinion} = \frac{W_t K_o K_v K_s P_d K_m K_B}{FJ} = \frac{(812.57)(1.75)(1.363)(1.112)(5)(1.205)(1)}{(2.5)(0.271)} = 19159psi$$

$$\sigma_{t,gear} = \frac{W_t K_o K_v K_s P_d K_m K_B}{FJ} = \frac{(812.57)(1.75)(1.363)(1.124)(5)(1.205)(1)}{(2.5)(0.423)} = 12410psi$$

4.2.8 Elastic Coefficient, C_p, and Surface Condition Factor, C_f

The elastic coefficient takes into account the elastic properties of the gear and pinion material. TABLE *V* shows several possible elastic coefficient values:

TABLE V: POSSIBLE ELASTIC COEFFICIENTS [14]

[redacted]

Assuming that the gears will be manufactured from steel, the elastic coefficient value is 2300$psi^{0.5}$.

The surface condition factor accounts for any known surface condition alterations, such as surface finish, residual stress, or work hardening [14]. Since nothing is known about the selected gear vendor's manufacturing process, the surface condition factor is assumed to be one.

4.2.9 Geometry Factor for Pitting Resistance, I

To determine the geometry factor for pitting resistance, equation (18) was used:

$$I = \frac{cos\phi sin\phi m_G}{2m_N(m_G + 1)} \tag{18}$$

For spur gears, m_N is one. Given a pressure angle of 20° and a speed ratio, m_G, of 6.125, the value of 0.149 was used for the geometry factor for pitting resistance.

4.2.10 Contact Stress, σ_c

The contact stress for the pinion and gear is always the same, since all factors in equation (8) are the same for the pinion and gear. Using equation (8), the contact stress estimate for the gear and pinion is:

$$\sigma_c = C_p\sqrt{\frac{W_tK_oK_vK_sK_mC_f}{d_PFI}} = 2300\sqrt{\frac{(812.57)(1.75)(1.363)(1.112)(1.205)(1)}{(3.2)(2.5)(0.149)}} = 107493psi$$

4.2.11 Expected Number of Cycles, N_c

Using the AGMA standard, the selection of a material uses charts that assume the number of cycles of the gear and pinion is 10^7 cycles. Therefore, modifications to the bending stress and contact stress are necessary if the gears are not designed to last exactly 10^7 cycles.

Cycles for the pinion and gear differ due to the difference in rotational speeds. The number of cycles, N_c, can be calculated as follows, where n is the rotational velocity of the gear in question and q is the number of load applications per revolution:

$$N_c = 60Lnq \tag{19}$$

To determine the design life in hours, L, the following assumptions were made:

- 20-year design life for the gearbox
- Four 95-foot passes per year (i.e. the wind tunnel is moved the full 95-foot rail span four times per calendar year)
- Each 95-foot pass takes 15 minutes (based on the wheel speed of 0.736rpm and 10.080 wheel revolutions per pass)

For the pinion, using equation (19), the number of expected cycles is approximately $6.89*10^5$ and the expected number of cycles for the gear is approximately $1.13*10^5$.

4.2.12 Stress-Cycle Factor for Bending, Y_N, and Stress-Cycle Factor for Pitting, Z_N

The stress-cycle factors modify the bending and contact strengths determined by equations (6) and (8) for design lives other than 10^7 cycles. The stress-cycle factor for bending is determined using Figure 7:

Figure 7: Chart for determining stress-cycle factor for bending [14]

Assuming a mid-range hardness value, the bending stress-cycle factor for the pinion is approximately 1.3. For the gear, the value is approximately 1.5. For the pitting resistance stress-cycle factor, Figure 8 is used:

Figure 8: Chart for determining stress-cycle factor for pitting resistance [14]

For the pinion, the pitting resistance stress-cycle factor was calculated as 1.162 and for the gear as 1.286.

4.2.13 Temperature Factor, K_T, and Reliability Factor, K_R

The temperature factor for the allowable stresses accounts for when operating temperatures are larger than 250°F. Since this gearbox will be outdoors and not operating continuously for long periods of time, a temperature factor of one was assumed. The reliability factor was assumed to be one, which accounts for a 99% reliability, according to TABLE *VI*:

TABLE VI: POSSIBLE RELIABILITY FACTORS (ADAPTED FROM [14])

Reliability	K_R
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

4.2.14 Allowable Stress, $\sigma_{c,all}$ and $\sigma_{t,all}$

The allowable bending stress and the allowable contact stress are determined using equations (7) and (9), respectively. All the factors in equations (6) and (8) typically account for a large enough safety factor that the value of S_F and S_H can be assumed as one. However, to ensure an extra safe design, the S_F and S_H values were assumed to be 1.2:

$$S_{t,pinion} = \frac{\sigma_{t,pinion} S_F K_T K_R}{Y_{N,pinion}} = \frac{(19159)(1.2)(1)(1)}{(1.3)} = 17686psi$$

$$S_{t,gear} = \frac{\sigma_{t,gear} S_F K_T K_R}{Y_{N,gear}} = \frac{(12410)(1.2)(1)(1)}{(1.5)} = 9928psi$$

$$S_{c,pinion} = \frac{\sigma_c S_F K_T K_R}{Z_{N,pinion}} = \frac{(107493)(1.2)(1)(1)}{(1.162)} = 111050psi$$

$$S_{c,gear} = \frac{\sigma_c S_F K_T K_R}{Z_{N,gear}} = \frac{(107493)(1.2)(1)(1)}{(1.286)} = 100332psi$$

The contact stress in the pinion was the largest value at 111050psi, so the material was required to withstand this stress at the very least. Figure 9 shows the allowable contact stress number, S_c, as a function of Brinell hardness for Grade 1 and Grade 2 through-hardened steel:

Figure 9: Chart for determining grades 1 and 2 through-hardened steel allowable contact stress numbers [14]

By selecting Grade 1 through-hardened steel with a Brinell hardness of 225, the allowable contact stress number is:

$$S_{c,all} = 322H_B + 29100psi = 322(225) + 29100 = 101550psi$$

This value can then be re-substituted into equation (9) to calculate the overall contact stress factor of safety for the first stage of the gearbox:

$$S_H = \frac{S_{c,all}Z_{N,pinion}}{S_{c,pinion}} = \frac{(101550)(1.162)}{111050} = 1.06$$

A similar calculation can be done for the bending stress in the pinion. Figure 10 shows the allowable bending stress number, S_c, as a function of Brinell hardness for Grade 1 and Grade 2 through-hardened steel:

Figure 10: Chart for determining grades 1 and 2 through-hardened steel allowable bending stress numbers [14]

By selecting the same material as determined by the contact stress process (Grade 1 steel with a Brinell hardness of 225), the allowable bending stress number is:

$$S_{t,all} = 77.3H_B + 12800psi = 77.3(225) + 12800 = 30193psi$$

This value can then be re-substituted into equation (7) to calculate the overall bending stress factor of safety for the first stage of the gearbox:

$$S_F = \frac{S_{t,all} Y_{N,pinion}}{S_{t,pinion}} = \frac{(30193)(1.3)}{17686} = 2.22$$

The values for S_F and S_H are both greater than one, which indicates that the selected material is a safe design selection.

4.3 Gear Specifications – Stage 2

Determining the planetary gear arrangement for stage 2 has numerous similarities to the design of stage 1. Many of the design factors are calculated or selected in the same manner. This section of the report outlines the differences that arise when determining the design of a planetary gear reduction.

4.3.1 Number of Planets and Transmitted Load, W_t

A planetary gear arrangement requires two inputs and one element to be fixed. In the case of a speed reducer, an effective solution is to fix the ring gear, drive the sun gear, and allow the planet carrier to be the output. Essentially, the ring gear and the sun gear are the inputs, but one input is equal to zero. With a fixed ring gear, the planets are free to rotate as driven by the sun gear, and the planet carrier rotates as the axes of the planets revolve around the central axis of the sun gear.

One of the many benefits of a planetary gear arrangement is that the power transferred through the reduction is split between the planets. The power split also applies to the transmitted loads. Therefore, the greater number of planets that are used, the lower the force applied to each gear. However, since the planetary gearbox in this case is a speed reducer and the ring gear is fixed, the sun gear must be smaller than planets to reduce the speed. If planets are too large and too numerous, they will not fit within the ring gear. Therefore, the ideal selection so as not to overcrowd the design space is four planets. Since four planets are available to transfer the power through the system, the design assumes that a load-balancing device transfers the power equally between the four planets, resulting in one-quarter of the transmitted load being transferred through each planet.

4.3.2 Gear Ratio

The first gear-and-pinion style reduction was designed with a 6.125:1 gear ratio. The OAGR of the reduction was 789.95:1. Therefore, for the remaining three stages, the gear ratio was required to be approximately 5.05:1 to obtain the OAGR.

A planetary gear system that has the ring gear fixed has the following gear ratio where n_{sun} and $n_{carrier}$ are the rotational velocities of the sun gear and the carrier, respectively:

$$GR = \frac{n_{sun}}{n_{carrier}} \tag{20}$$

4.3.3 Rotational Speeds and Number of Teeth

To determine the gear and carrier speeds, as well as the number of teeth on each gear, the following equations are required:

$$n_{sun} = x + y \tag{21}$$

$$n_{carrier} = y \tag{22}$$

$$n_{planet} = -\frac{N_1}{N_3}x + y \tag{23}$$

$$n_{ring} = -\frac{N_1}{N_4}x + y \tag{24}$$

To solve this system of equations, the rotational velocity of the ring gear is set to zero. The value of the rotational velocity of the sun gear is known based on the output from the previous stage. Also, the required carrier rotational velocity (output) is known due to the desired gear ratio of 5.05:1.

$$n_{ring} = -\frac{N_1}{N_4}x + y = 0$$

$$n_{sun} = x + y = 94.98rpm$$

$$n_{carrier} = y = \frac{n_{sun}}{5.05} = 18.80rpm$$

The rotational velocity of the carrier, represented as *y*, is known, therefore, the value of *x* can be determined using equation (21):

$$x = n_{sun} - y = 94.98 - 18.80 = 76.18$$

To solve for the number of teeth on each gear, one gear must be arbitrarily specified. Allow the sun gear teeth, N_1, to be equal to 35. Then the number of ring gear teeth, N_4, is:

$$N_4 = \frac{-N_1 x}{n_{ring} - y} = -\frac{35(76.18)}{0 - 18.80} = 141.82$$

The basic formula that governs the number of gear teeth is as follows:

$$N_{ring} = 2N_{planet} + N_{sun} \tag{25}$$

Therefore, the number of teeth on the planets is:

$$N_{planet} = \frac{N_{ring} - N_{sun}}{2} = \frac{141.82 - 35}{2} = 53.41$$

However, gears must have an integer number of teeth, rounding up to the nearest suitable values for ring gear teeth and planet gear teeth yield 143 teeth for the ring gear and 54 teeth for the planet gear. Finally, the rotational velocity of a planet gear around its own axis is:

$$n_{planet} = -\frac{N_1}{N_3}x + y = -\frac{35}{54}(76.18) + 18.80 = -30.58rpm$$

The negative rotational velocity of the planets indicates that they are spinning the opposite direction from the sun and planet carrier.

The number of gear teeth in a planetary gear system is a calculation that requires iterations. In this project, a baseline number of gear teeth was selected and the stress calculations were completed based on the number of teeth. If the stresses were found to be too large to select a material, the number of teeth on the gears was increased and the stress calculations were performed again. Ultimately, the number of teeth that yielded the ability to select a material was 35 teeth for the sun gear, 54 teeth for the planet gears, and 143 teeth for the ring gear.

4.4 Gear Specification Summary

The four-stage reduction system incorporates one gear-and-pinion style reduction as the first stage and three planetary gear stages as the final three stages. Figure 11 illustrates the layout of the final gearing design. Note that the shaft and bearing design is not incorporated into the schematic:

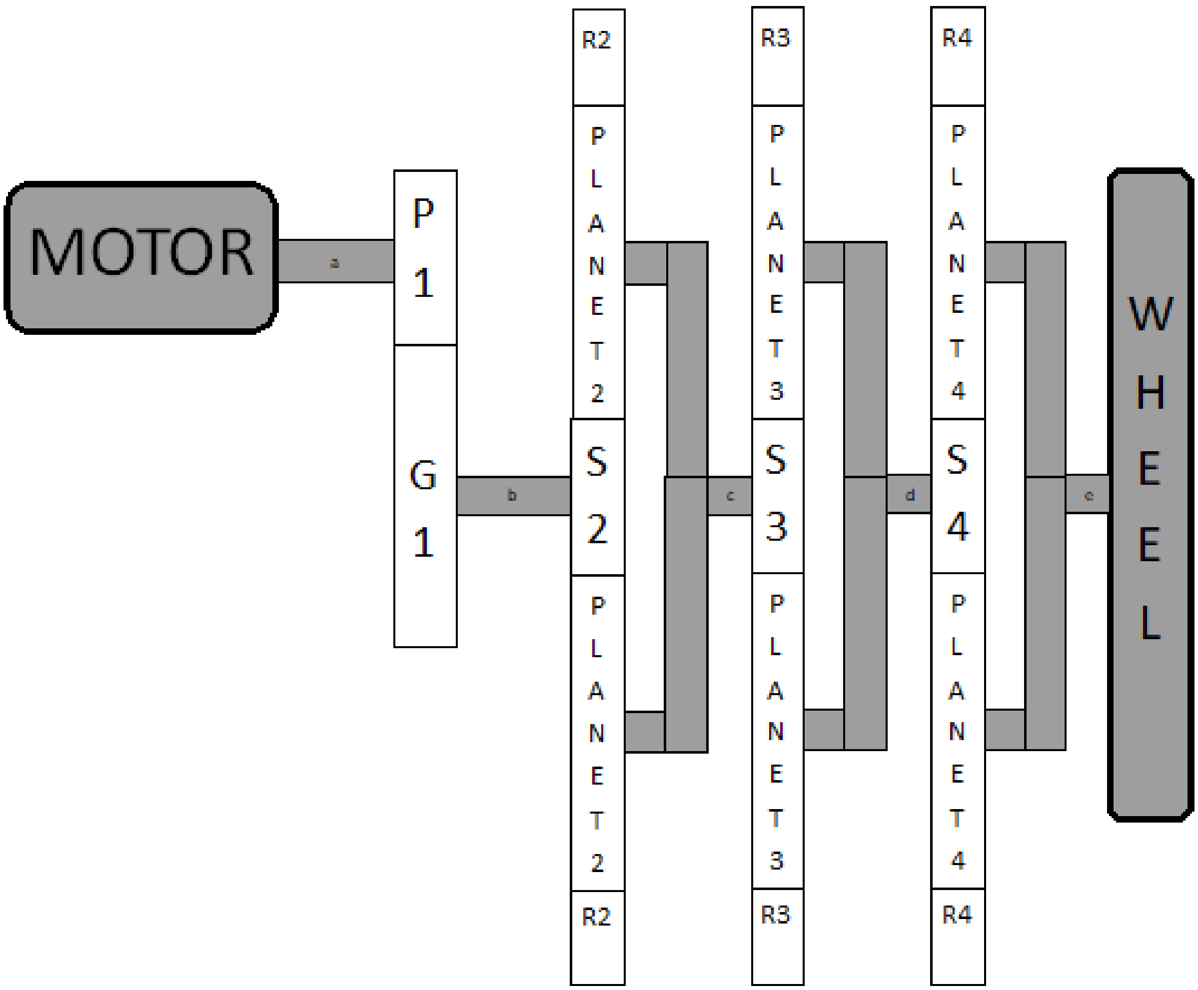

Figure 11: Top view of final gear layout

The calculations of only two of the four stages were shown in this appendix. Due to the similarities between stages 2, 3, and 4, the inclusion of stage 3 and 4 calculations was not necessary. TABLE VII and TABLE VIII summarize the results of the calculations for all four stages, including bending and contact stresses, as well as selected material and hardness values where applicable:

TABLE VII: FINAL DESIGN INPUT VARIABLES

Design Item	Value	Unit
Assumed Input Power	12	HP
Diametral Pitch	5	teeth/in
Pressure Angle	20	degrees

TABLE VIII: FINAL DESIGN ANALYSIS RESULTS

	Stage 1	Stage 2	Stage 3	Stage 4	Note
Reduction Type	Gear and pinion	Planetary	Planetary	Planetary	
Number of Planets	N/A	4	4	4	
Input Speed (Sun Gear) [RPM]	581.72	94.98	18.80	3.72	
Output Speed (Carrier) [RPM]	94.98	18.80	3.72	0.74	
Gear Ratio	6.125:1	5.052:1	5.052:1	5.052:1	
Pinion Diameter [in]	3.2	N/A	N/A	N/A	
Gear Dameter [in]	19.6	N/A	N/A	N/A	
Pinion Teeth	16	N/A	N/A	N/A	
Gear Teeth	98	N/A	N/A	N/A	
Sun Diameter [in]	N/A	7	7	7	
Planet Diameter [in]	N/A	10.8	10.8	10.8	
Ring Diameter [in]	N/A	28.6	28.6	28.6	
Sun Teeth	N/A	35	35	35	
Planet Teeth	N/A	54	54	54	
Ring Teeth	N/A	143	143	143	
Pitch-line Velocity [ft/min]	487.34	174.05	34.45	6.82	Sun-planet for stages 2, 3,, 4
Transmitted Load [lbf]	812.6	568.8	2873.8	14519.5	Sun to one planet for stages 2, 3,, 4
Face Width [in]	2.5	2.5	2.5	4	
Pinion/Sun Tooth Contact Stress [psi]	111050	63164	123075	196103	
Pinion/Sun Tooth Bending Stress [psi]	17686	7953	33061	94722	
Material Selection	Grade 1 through-hardened steel	Grade 1 through-hardened steel	Grade 2 through-hardened steel	Grade 3 carburized and hardened steel	
Allowable Contact Stress	101550	101550	156450	275000	
Allowable Bending Stress	30193	30193	52100	75000	
Hardness Value [Brinell]	225	225	350	See note	See Table 9 of ANSI/AGMA 2001-D04
Factor of Safety for Wear	1.06	1.91	1.66	2.06	
Factor of Safety for Bending	2.22	4.94	2.21	1.35	

5 Shaft Calculations

The shaft diameters were determined using the following data gathered from the gear reduction:

TABLE IX: GEAR SPECIFICATION SUMMARY USED FOR SHAFT CALCULATIONS

Torques		
	# planetary gears/stage	4
Stage 1	Pinion diameter [in]	3.2
	Gear diameter [in]	19.6
	Transmitted load [lbf]	812.5675
Stage 2	Sun gear diameter [in]	7
	sun gear teeth	35
	Planet gear diameter [in]	10.8
	Ring gear diameter [in]	28.6
	ring gear teeth	143
	Transmitted load [lbf]	568.7973
Stage 3	Sun gear diameter [in]	7
	sun gear teeth	35
	Planet gear diameter [in]	10.8
	Ring gear diameter [in]	28.6
	ring gear teeth	143
	Transmitted load [lbf]	2873.788
Stage 4	Sun gear diameter [in]	7
	sun gear teeth	35
	Planet gear diameter [in]	10.8
	Ring gear diameter [in]	28.6
	ring gear teeth	143
	Transmitted load [lbf]	14519.51

From the values in TABLE *IX* and knowing that the pressure angle was 20 degrees throughout the reduction, the torques on the shafts could be determined.

A free body diagram of the shaft system was generated to see how the forces interacted:

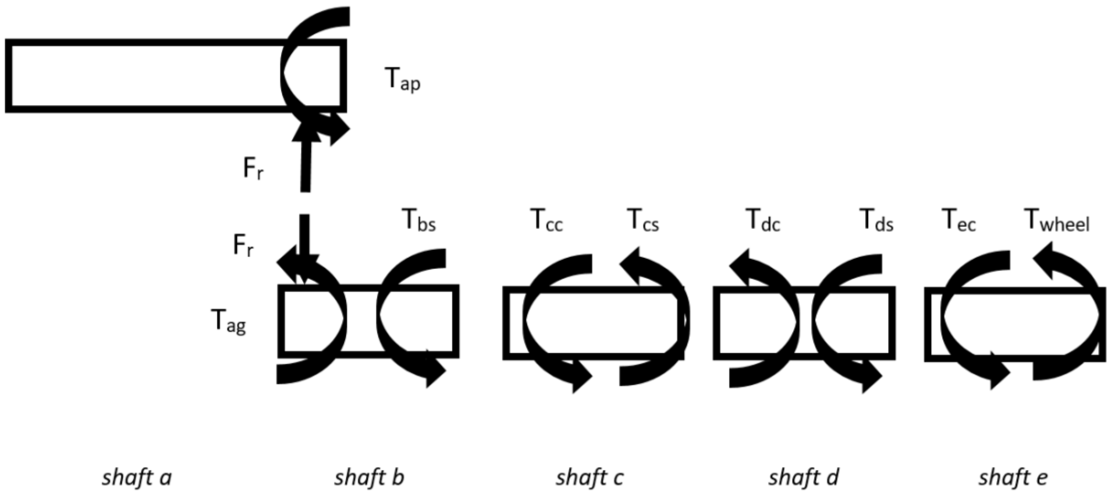

Figure 12: Free-body diagram of the forces and torques on the shafts

5.1 Shaft *a*

The material selected for shaft *a* was AISI 4130, forged and Normalized to 1600°F (~871°C). This material has excellent strength and is quite common for shafts. AISI 4130 has the following mechanical properties [16]:

TABLE X: ULTIMATE AND YIELD STRENGTH OF AISI 4130 NORMALIZED TO 1600°F

AISI 4130 (Normalized to 1600 °F)	
Ultimate tensile strength (psi)	95000
Yield tensile strength (psi)	60200

The minimum diameters of the critical areas of the shafts were determined using the DE-ASME elliptic equation in the following form:

$$d = \left\{\frac{16n}{\pi}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\frac{1}{2}}\right\}^{\frac{1}{3}} \tag{26}$$

Where:

n is the factor of safety.

K_f is the fatigue stress concentration factor.

K_{fs} is the fatigue concentration factor due to shear.

M_a is the alternating bending stress.

M_m is the midrange bending stress.

T_a is the alternating torque.

T_m is the midrange torque.

S_e is the fatigue stress.

S_y is the yield tensile stress.

Shaft *a* was subjected to a torque and a bending force from the stage 1 gear-and-pinion reduction. The bending force was calculated based on the transmitted load (W_t):

$$F_r = (W_t)\tan(Pressure\ angle) \tag{27}$$

$$M_a = F_r = 813\tan(20) = 296\ lbf$$

The torque can be determined by:

$$T = W_t\left(\frac{d}{2}\right) \tag{28}$$

Where T is the torque

W_t is the transmitted load

d is the diameter of the gear

Thus, for shaft *a,* the torque at the pinion is:

$$T_m = T_{ap} = 813\left(\frac{3.2}{2}\right) = 1300.8\ [lbf \cdot in]$$

The midrange bending stress (M_m) and alternating torque (T_a) were found to be zero.

The equations used in determining the fatigue stress concentration factors were:

$$K_f = 1 + q(K_t - 1) \tag{29}$$

$$K_{fs} = 1 + q(K_{ts} - 1) \tag{30}$$

Where K_t and K_{ts} are the stress concentration factors.

In calculating the stress concentrations, a small diameter ratio (D/d = 1.1) and a small fillet radius (r/d = 0.025) were assumed to keep the estimates conservative. The values for K_t and K_{ts} were taken from Figure *13* and Figure *14* as 2.6 and 1.6, respectively.

Figure 13: Round shaft with shoulder fillet in torsion.

Figure 14: Round shaft with shoulder fillet in bending.

The notch sensitivity factors q and q_s were determined from Figure *15* and Figure *16*, assuming a notch radius of 0.02 inches to be 0.7 for both.

Figure 15: Notch sensitivity factor for steels under reversed axial loading.

Figure 16:Notch sensitivity factor for steels under reversed torsion.

As a result:

$$K_f = 1 + 0.7(1.6 - 1) = 1.42$$

$$K_{fs} = 1 + 0.7(2.6 - 1) = 2.12$$

The remaining value to be found was the endurance limit, since no convincing values were found, the approximation $S_e \cong 0.5S_{ut}$ was used, as this is lower than what research has shown. The value used for S_e' was 47.5 kpsi, based off the value of AISI 4130's ultimate tensile strength in the Machinery's handbook [16]. AISI 4130 has a tensile yield strength of 95 kpsi.

Several modifying factors were applied to the endurance limit of the metal, accounting for the surface finish, size, operating temperature, loading condition and desired reliability.

The Surface factor (k_a) can be determined by substituting a and b from TABLE *XI* into Equation (31) below. Since the anticipated surface finish is machined, the surface factor becomes:

TABLE XI: PARAMETERS FOR THE SURFACE FACTOR (K_A)

[redacted]

$$k_a = aS_{ut}^b \tag{31}$$

The Size factor (k_b) was determined by the following Equation (32). Assuming a diameter of 0.75 in for shaft a, k_b = 0.9.

$$\begin{aligned} &0.879d^{-0.107}\ for\ 0.11 \le d \le 2\ in \\ &0.91d^{-0.157}\ for\ 2 \le d \le 10\ in \end{aligned} \tag{32}$$

The Temperature factor (k_d) starts to drop once the operating temperature exceeds 70 °F, thus it is taken to be 1.

The Loading factor (k_c) is also 1 because this is a case of combined loading, and the loading is accounted for in the DE-ASME equation.

The Reliability factor (k_e) is taken from TABLE *XII*, and at a desired reliability of 99%, is 0.814.

TABLE XII: RELIABILITY FACTORS FOR VARIOUS LEVELS OF RELIABILITY.

[redacted]

The factors and their associated values are summarized in TABLE *XIII*:

TABLE XIII: ENDURANCE LIMIT MODIFYING FACTORS FOR SHAFT A.

Factor	Condition	Value
Surface factor (k_a)	As machined	0.7
Size factor (k_b)	Assuming d ~ 0.75 in	0.9
Temperature factor (k_d)	Operating a temp < 70 °F	1
Loading factor (k_c)	Assuming torsion	1
Reliability factor (k_e)	99 % reliability	0.814

The adjusted endurance limit can now be determined my multiplying the nominal endurance limit by the modifying factors.

$$S_e = k_a k_b k_c k_d k_e S'_e$$

$$S_e = (0.7)(0.9)(1)(1)(0.814)(47.5) = 24.36\ Kpsi$$

The safety factor (n) was taken to be 1 as the stresses in the gears had already been calculated with a safety factor. A summary of the values determined for the DE-ASME elliptic failure criterion is given in TABLE *XIV*:

TABLE XIV: VALUES USED TO FIND THE CRITICAL DIAMETER OF SHAFT A.

Variable	Value
M_a	296 in-lbf
T_a	0
M_m	0
T_m	1300.8 in-lbf
K_f	2.12
K_{fs}	1.42
S_e	24360 psi
S_y	60200 psi
n	1

With these values, a minimum diameter could be established.

$$d = \left\{ \frac{16n}{\pi} \left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = \left\{ \frac{16(1)}{\pi} \left[4\left(\frac{2.12(296)}{24360}\right)^2 + 3\left(\frac{1.42(1300.8)}{60200}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = 0.73\ inches$$

5.2 Shaft b

The material selected for shaft be was also AISI 4130. The relevant mechanical properties are listed in TABLE *X*.

The same process as Shaft a was followed. For the fatigue concentration factors, the ratios D/d and r/D were assumed to be 1.2 and 0.025 respectively, and the notch radius was assumed to be 0.03 (2% of the assumed diameter of 1.5 inches).

A summary of the modifying factors and DE_ASME Elliptic equation values are listed in TABLE *XV* and TABLE *XVI*:

TABLE XV: MODIFYING FACTORS FOR ENDURANCE LIMIT OF SHAFT B.

Factor	Condition	Value
Surface factor (k_a)	As machined	0.7
Size factor (k_b)	Assuming d ~ 1.5 in	0.84
Temperature factor (k_d)	Operating a temp < 70 °F	1
Loading factor (k_c)	Assuming torsion	1
Reliability factor (k_e)	99 % reliability	0.814

TABLE XVI: VALUES USED TO FIND THE CRITICAL DIAMETER OF SHAFT B.

Variable	Value
M_a	296 in-lbf
T_a	0
M_m	0
T_m	7963.2 in-lbf
K_f	2.2
K_{fs}	1.462
S_e	24360 psi
S_y	60200 psi
n	1

$$d = \left\{ \frac{16(1)}{\pi} \left[4\left(\frac{2.2(296)}{24360}\right)^2 + 3\left(\frac{1.462(7963.2)}{60200}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = 1.2\ inches$$

5.3 Shaft c

The material selected for shaft be was also AISI 4130. The relevant mechanical properties are listed in TABLE *X*.

The same process as Shaft a and b was followed. For the fatigue concentration factors, the ratios D/d and r/D were assumed to be 1.2 and 0.025 respectively, and the notch radius was assumed to be 0.05 (10% of the assumed diameter of 2.5 inches).

TABLE XVII: MODIFYING FACTORS FOR ENDURANCE LIMIT OF SHAFT C.

Factor	Condition	Value
Surface factor (k_a)	As machined	0.7
Size factor (k_b)	Assuming d ~ 2.5 in	0.79
Temperature factor (k_d)	Operating a temp < 70 °F	1
Loading factor (k_c)	Assuming torsion	1
Reliability factor (k_e)	99 % reliability	0.814

TABLE XVIII: VALUES USED TO FIND THE CRITICAL DIAMETER OF SHAFT C.

Variable	Value
M_a	0
T_a	0
M_m	0
T_m	40499 in-lbf
K_f	2.264
K_{fs}	1.492
S_e	21652 psi
S_y	60200 psi
n	1

$$d = \left\{ \frac{16(1)}{\pi} \left[\left(\frac{1.492(7963.2)}{60200}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = 2.07\ inches$$

5.4 Shaft d

The material selected for shaft be was also AISI 4130. The relevant mechanical properties are listed in TABLE *X*.

The same process as Shaft a, b and c was followed. For the fatigue concentration factors, the ratios D/d and r/D were assumed to be 1.2 and 0.025 respectively, and the notch radius was assumed to be 0.08 (10% of the assumed diameter of 4 inches).

TABLE XIX: MODIFYING FACTORS FOR ENDURANCE LIMIT OF SHAFT D.

Factor	Condition	Value
Surface factor (k_a)	As machined	0.7
Size factor (k_b)	Assuming d ~ 2.5 in	0.72
Temperature factor (k_d)	Operating a temp < 70 °F	1
Loading factor (k_c)	Assuming torsion	1
Reliability factor (k_e)	99 % reliability	0.814

TABLE XX: VALUES USED TO FIND THE CRITICAL DIAMETER OF SHAFT D.

Variable	Value
M_a	0
T_a	0
M_m	0
T_m	205964 in-lbf
K_f	2.296
K_{fs}	1.51
S_e	21652 psi
S_y	60200 psi
n	1

$$d = \left\{ \frac{16(1)}{\pi} \left[\left(\frac{1.51(205946)}{60200} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = 3.58\ inches$$

5.5 Shaft e

The material selected for shaft be was AISI 4150. The relevant mechanical properties are listed in TABLE *XXIII*. This material is a bit stronger than AISI 4130 and was selected for this shaft so that the shaft diameter would stay relatively small, as this shaft has a large amount of torque being transferred through it.

The same process as Shaft a, b, c and d was followed. For the fatigue concentration factors, the ratios D/d and r/D were assumed to be 1.2 and 0.025 respectively, and the notch radius was assumed to be 0.11 (10% of the assumed diameter of 5.5 inches). Qs= 0.85 Qst=0.9

TABLE XXI: MODIFYING FACTORS FOR ENDURANCE LIMIT OF SHAFT E.

Factor	Condition	Value
Surface factor (k_a)	As machined	0.7
Size factor (k_b)	Assuming d ~ 2.5 in	0.7
Temperature factor (k_d)	Operating a temp < 70 °F	1
Loading factor (k_c)	Assuming torsion	1
Reliability factor (k_e)	99 % reliability	0.814

TABLE XXII: VALUES USED TO FIND THE CRITICAL DIAMETER OF SHAFT E.

Variable	Value
M_a	0
T_a	0
M_m	0
T_m	1047470 in-lbf
K_f	2.296
K_{fs}	1.51
S_e	33404 psi
S_y	106500 psi
n	1

TABLE XXIII: ULTIMATE AND YIELD STRENGTH OF AISI 4150 NORMALIZED TO 1600 °F [16]

AISI 4150 (Normalized to 1600 °F)	
Ultimate tensile strength (psi)	167500
Yield tensile strength (psi)	106500

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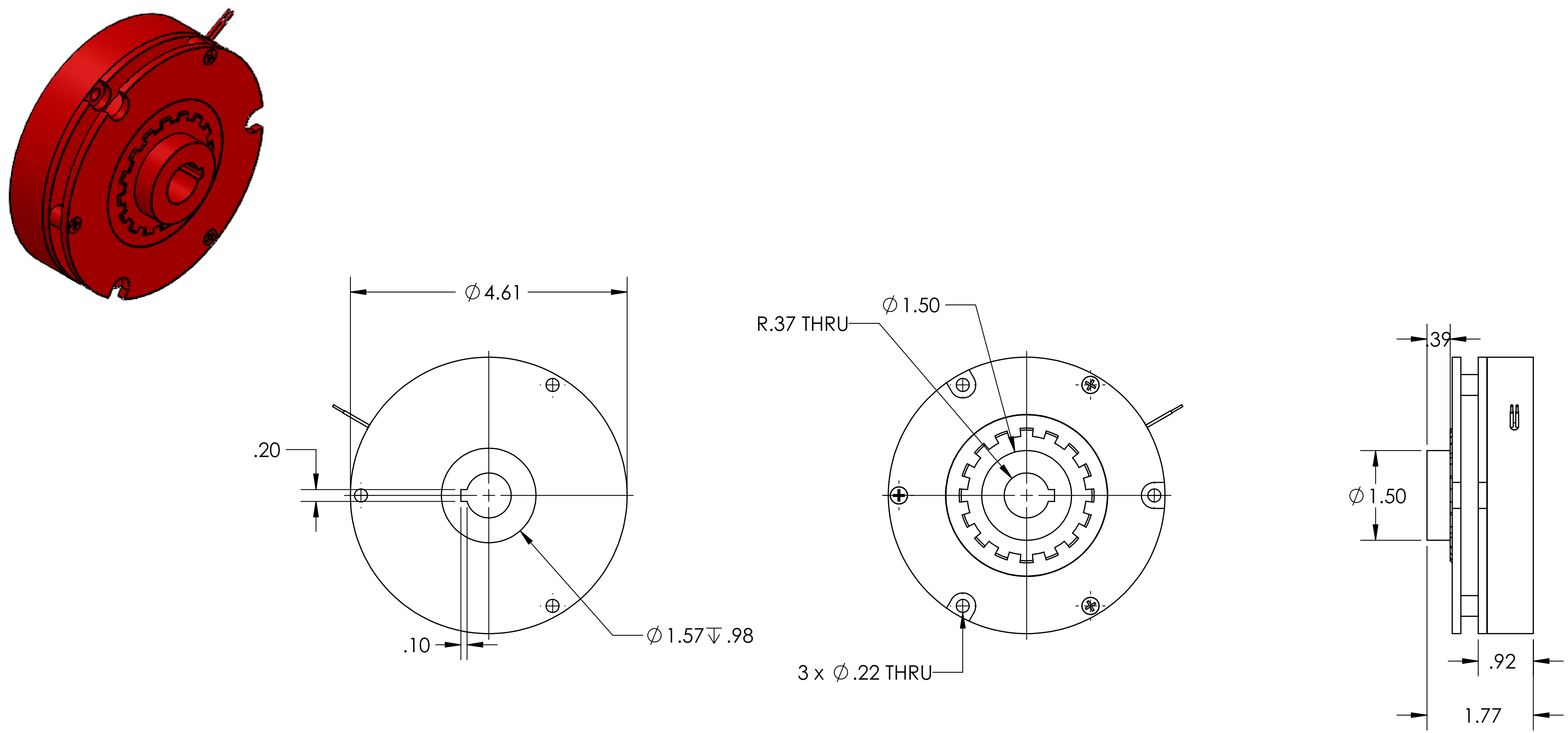
Appendix D – Drawings and Bill of Materials

Table of Contents

1 Drawings

2 Bill of Materials



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TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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ENG APPR.		
MFG APPR.		
Q.A.		

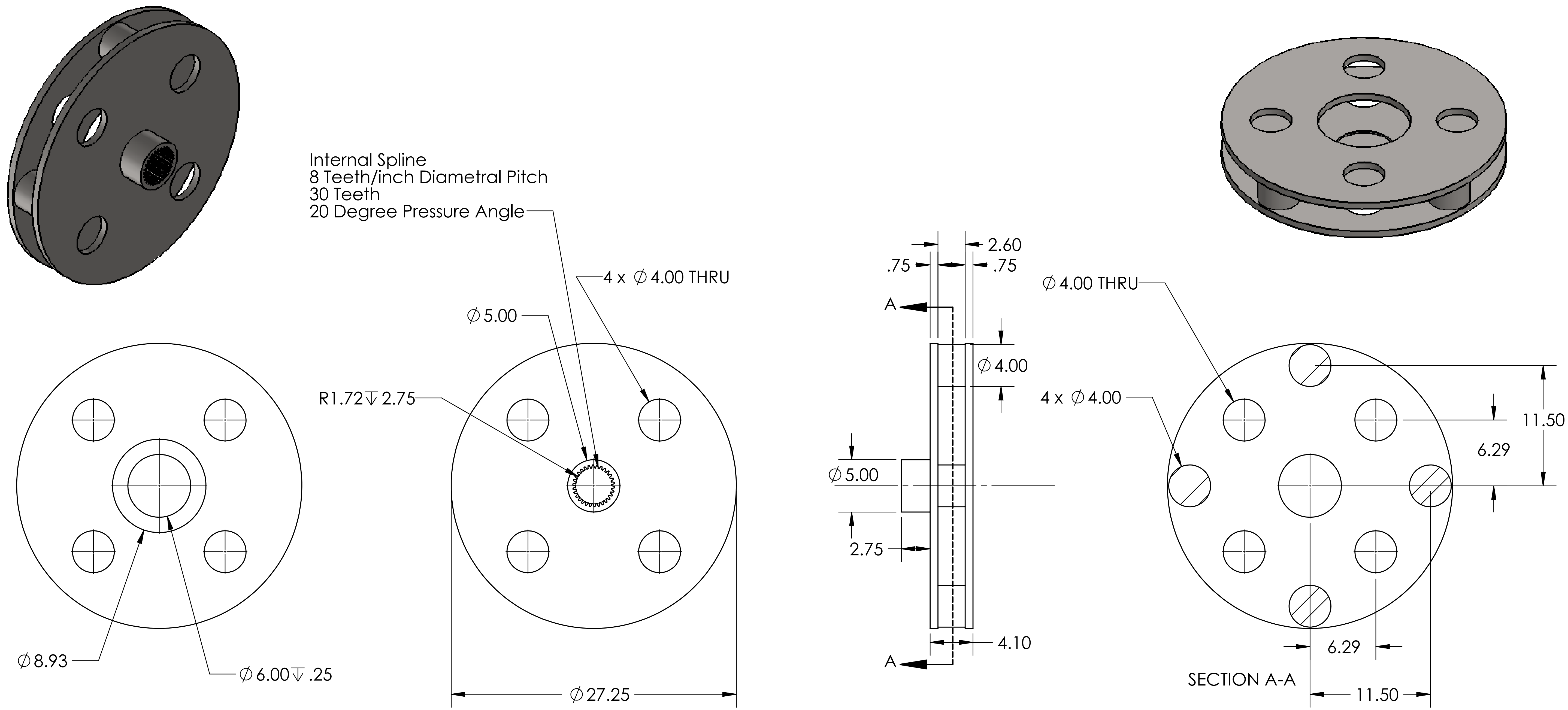
COMMENTS:

TITLE:

BRake

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



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APPLICATION		DO NOT SCALE DRAWING			

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TITLE: Carrier stage 2 and 3

SIZE	DWG. NO.	REV
B		
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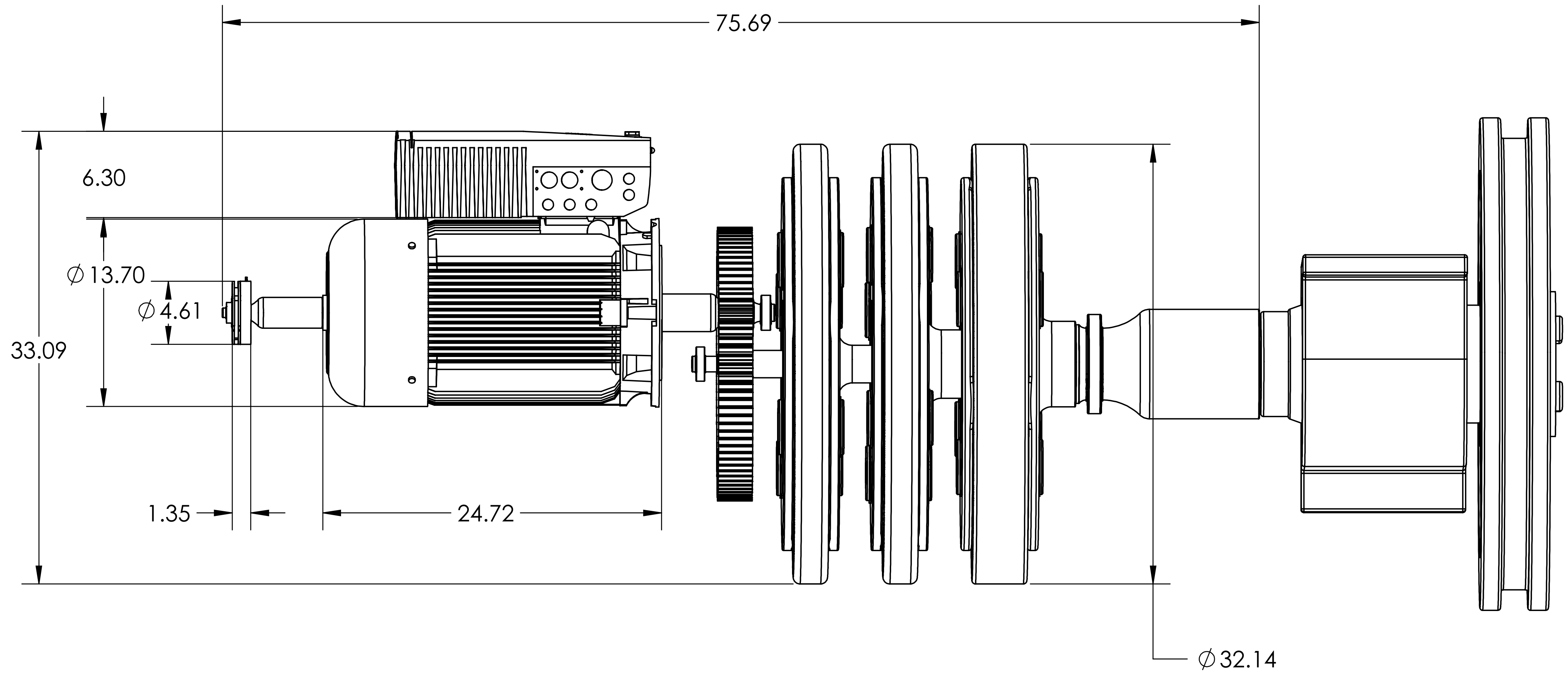
4 3 2 1

B



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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

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	NAME	DATE
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ENG APPR.		
MFG APPR.		
Q.A.		

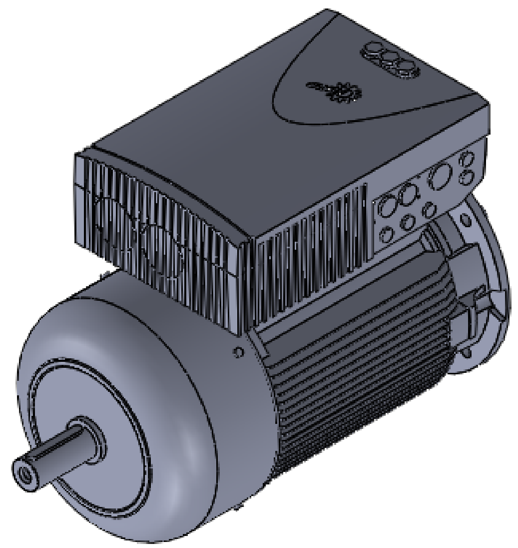
COMMENTS:

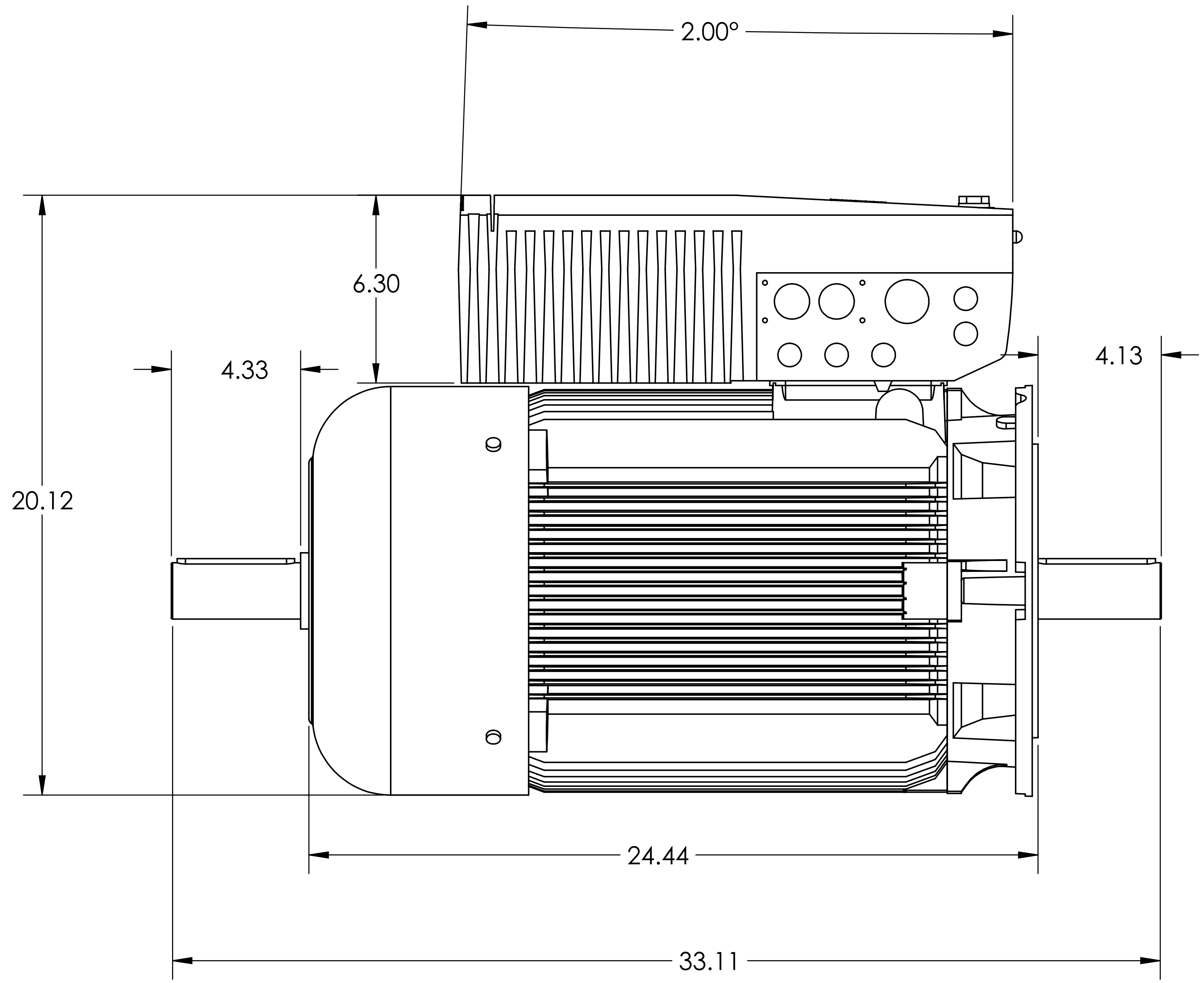
TITLE: Drive System Assembly

SIZE	DWG. NO.	REV
B		

SCALE: 1:50 WEIGHT: SHEET 1 OF 1

4 3 2 1





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MFG APPR.		
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COMMENTS:

TITLE:

AC Motor

SIZE B | DWG. NO. | REV

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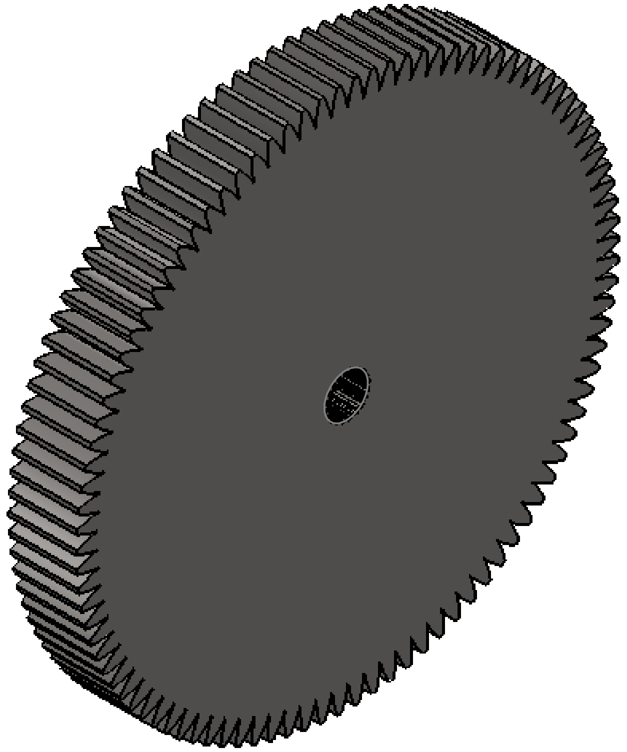
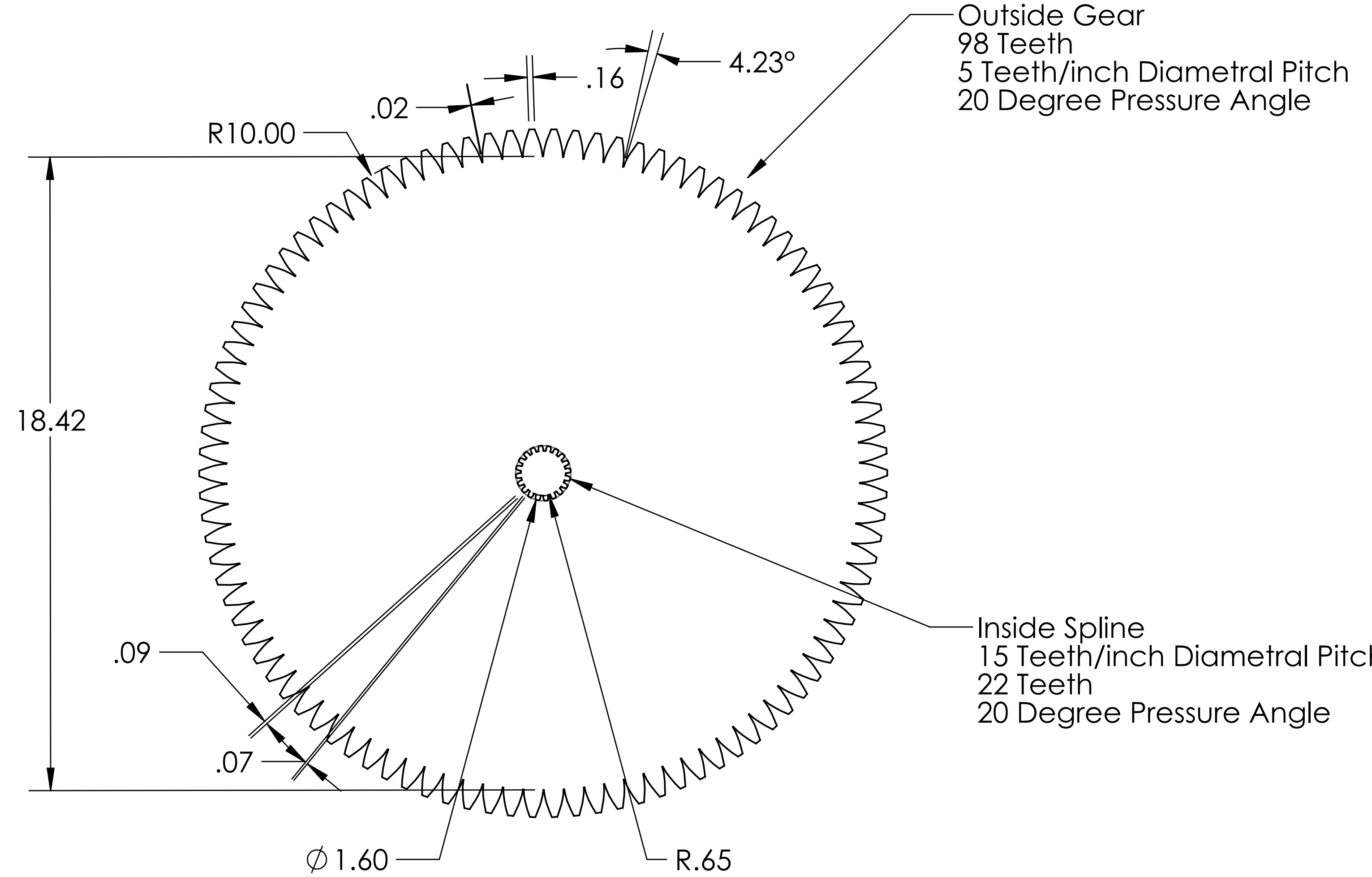
4 3 2 1

B





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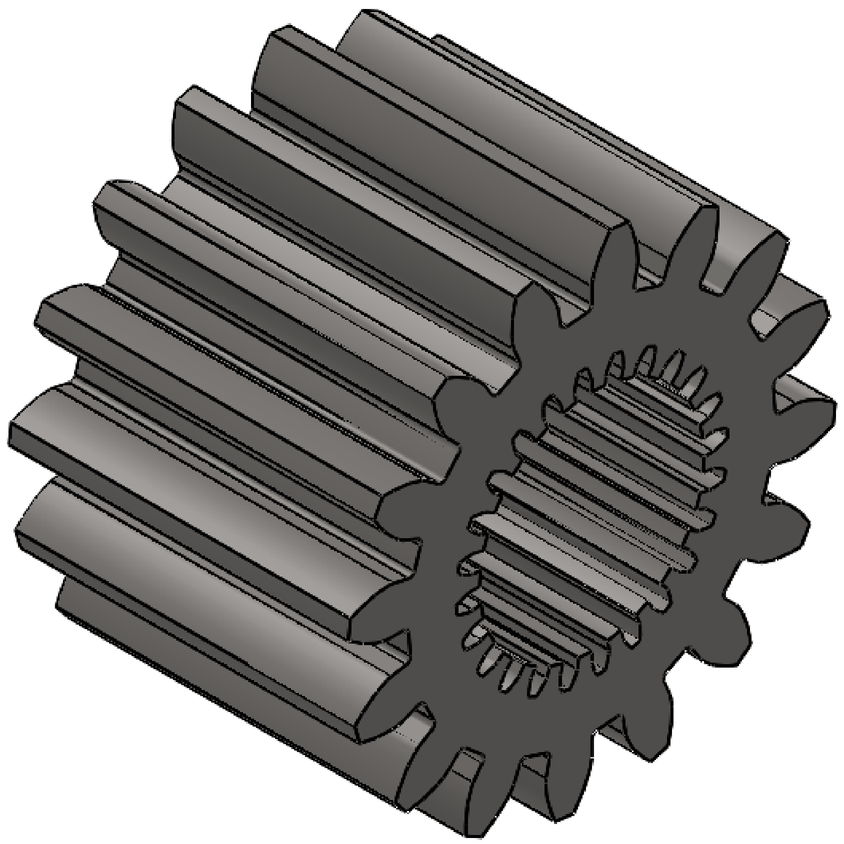
COMMENTS:

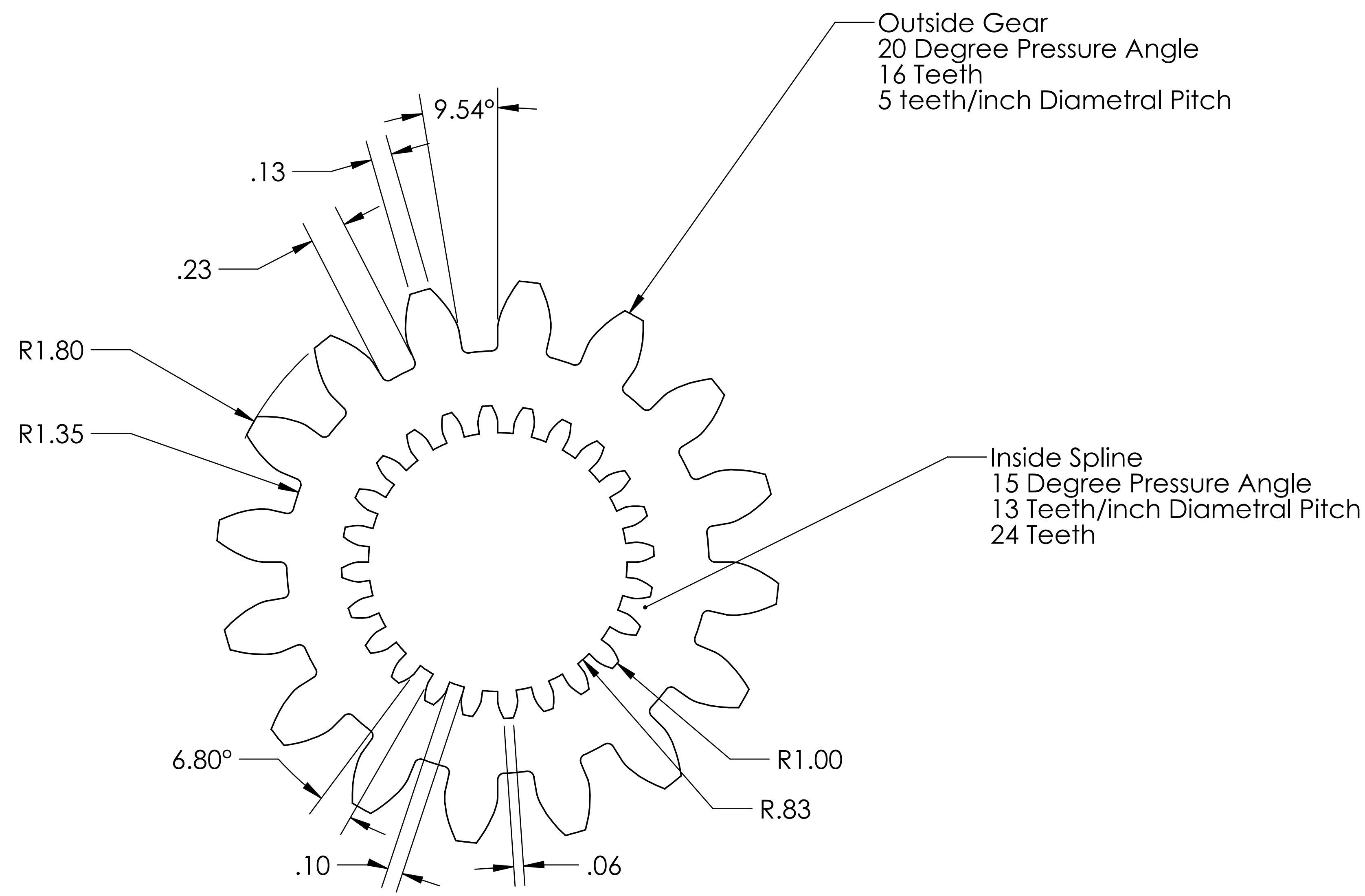
TITLE:

SIZE B | DWG. NO. Gear | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

4 3 2 1





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INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
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NEXT ASSY	USED ON
APPLICATION	

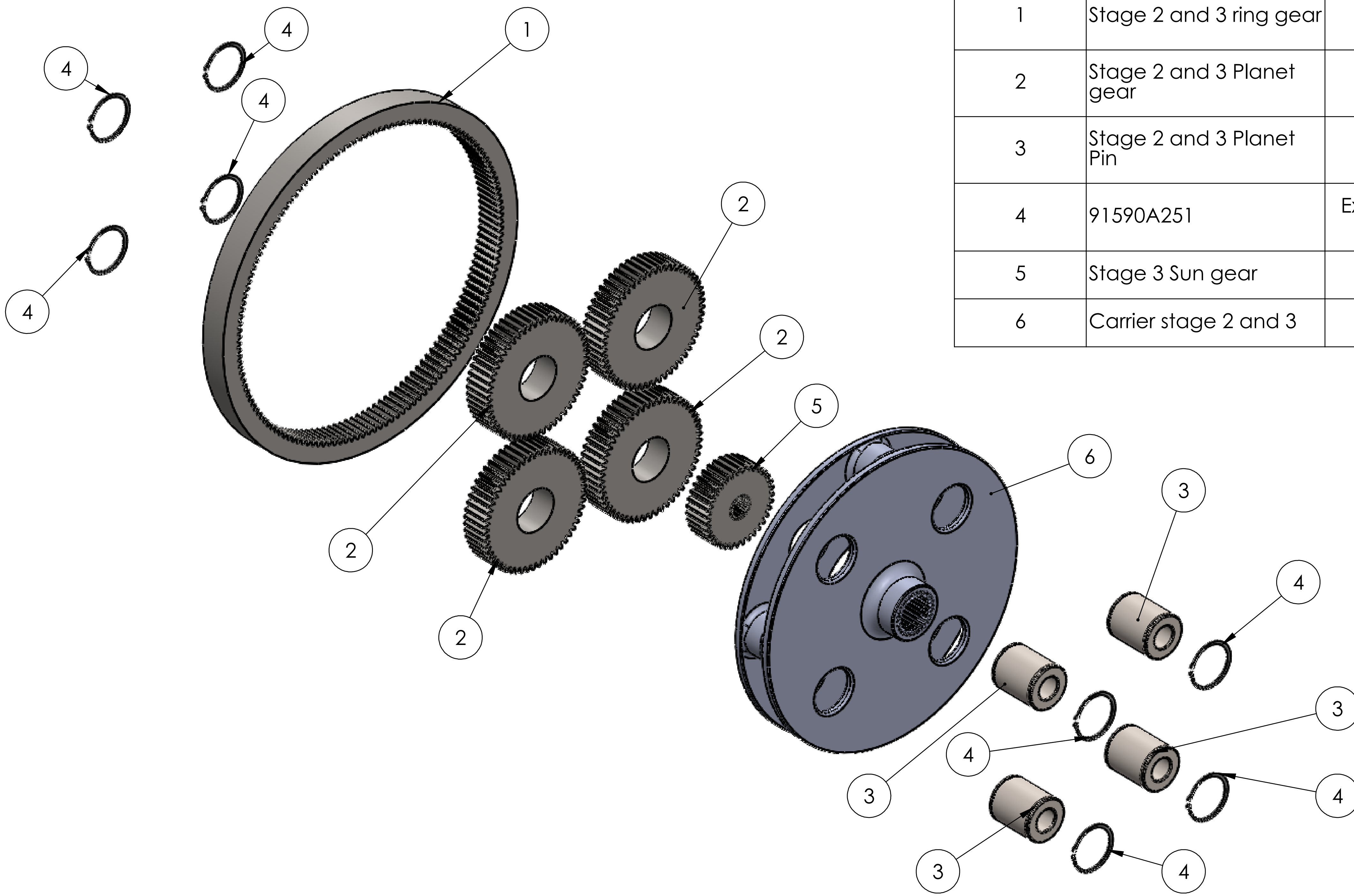
TITLE:

Pinion

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	planetary bom 1/QTY.
1	Stage 2 and 3 ring gear		1
2	Stage 2 and 3 Planet gear		4
3	Stage 2 and 3 Planet Pin		4
4	91590A251	External Retaining Ring, Shaft size 4" OD	8
5	Stage 3 Sun gear		1
6	Carrier stage 2 and 3		1



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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

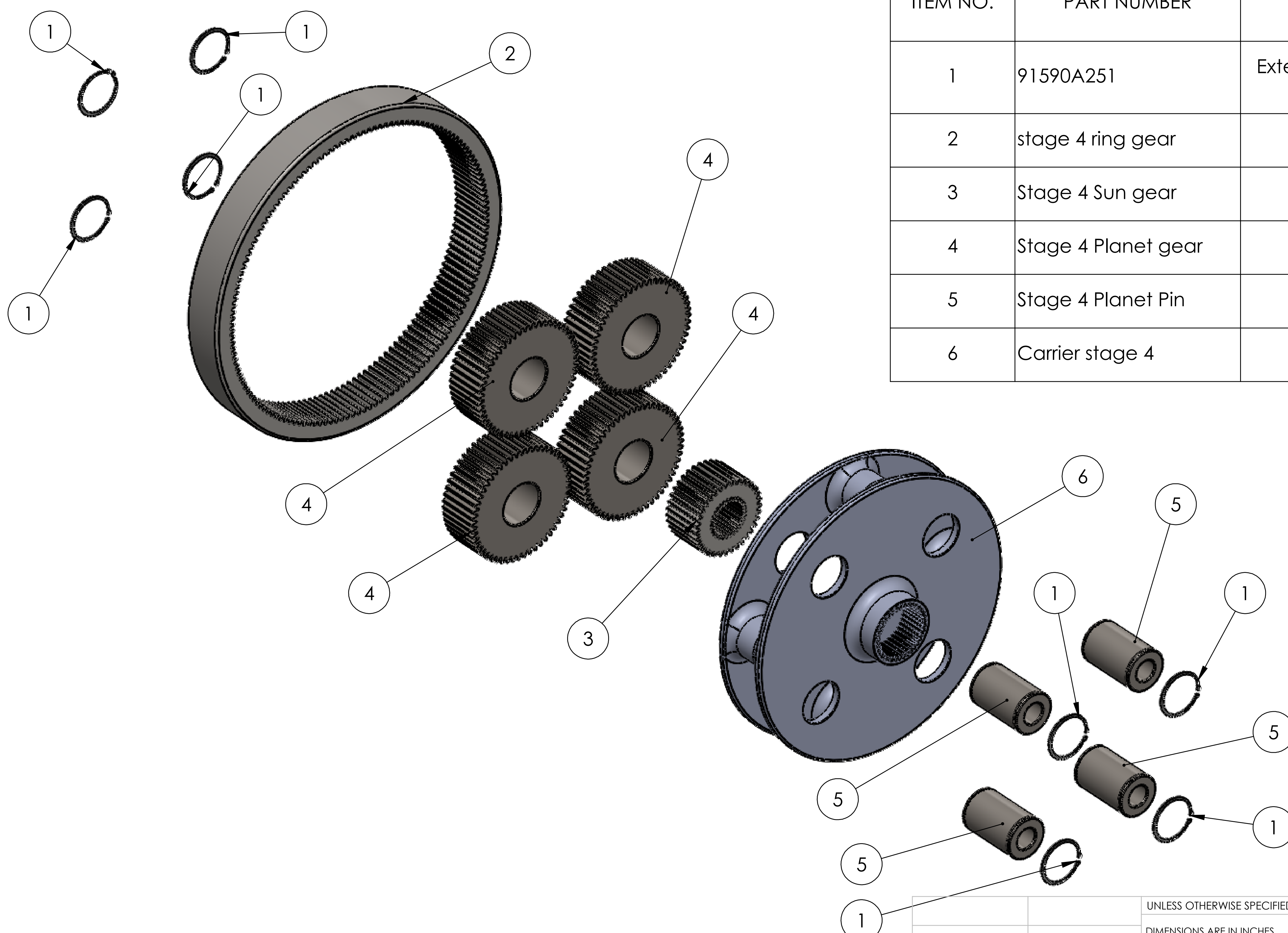
TITLE:

Stage 3 Planetary Set

SIZE B | DWG. NO. | REV

SCALE: 1:10 | WEIGHT: | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	planetary bom 2/QTY.
1	91590A251	External Retaining Ring, shaft size 4" OD	8
2	stage 4 ring gear		1
3	Stage 4 Sun gear		1
4	Stage 4 Planet gear		4
5	Stage 4 Planet Pin		4
6	Carrier stage 4		1



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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
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ENG APPR.		
MFG APPR.		
Q.A.		

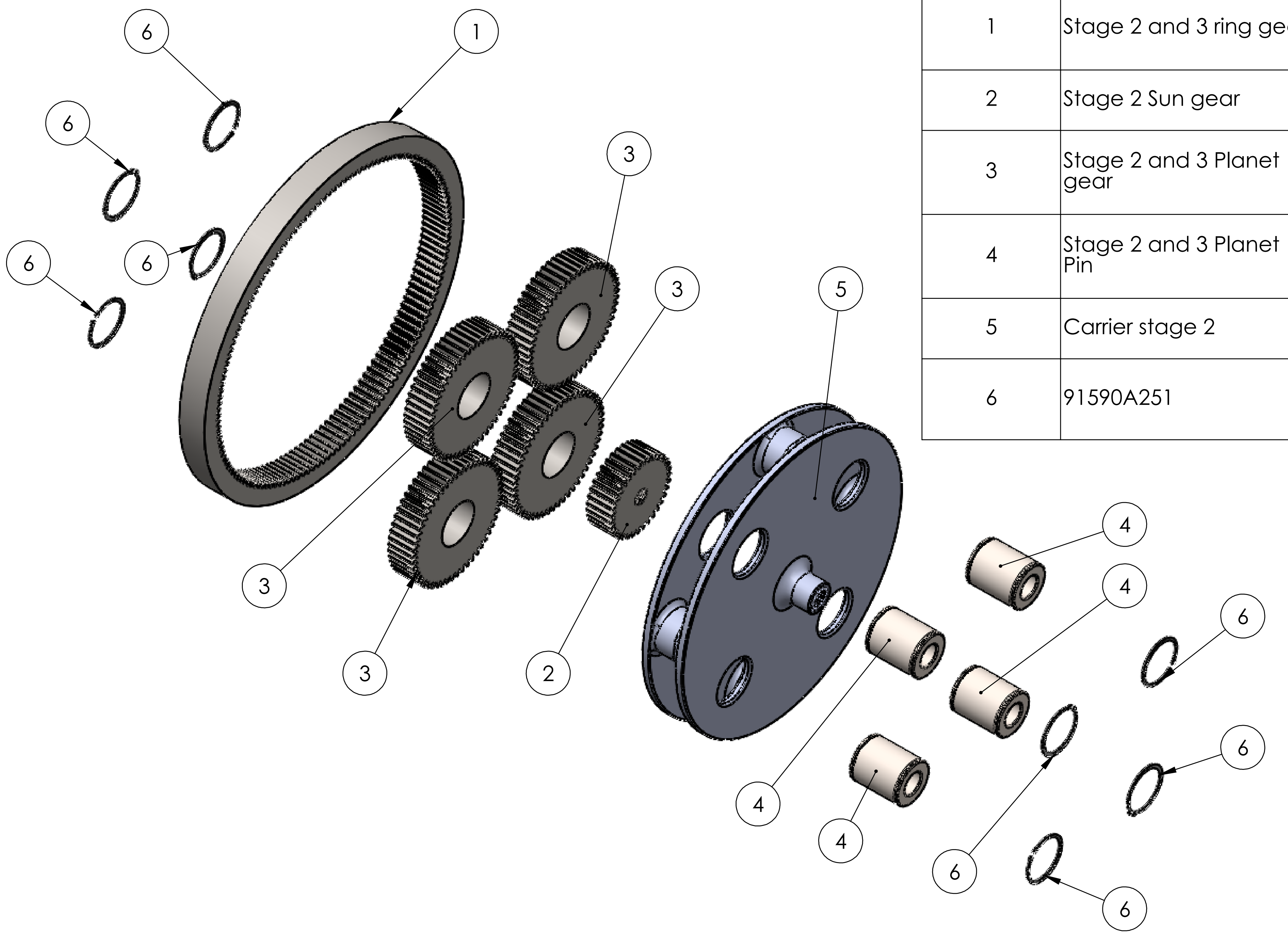
COMMENTS:

TITLE:
Stage 4 Planetary Set

SIZE B | DWG. NO. | REV

SCALE: 1:10 | WEIGHT: | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	Planetary/QTY.
1	Stage 2 and 3 ring gear		1
2	Stage 2 Sun gear		1
3	Stage 2 and 3 Planet gear		4
4	Stage 2 and 3 Planet Pin		4
5	Carrier stage 2		1
6	91590A251	External Retaining Ring shaf t size 4" OD	8



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			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE B DWG. NO. REV
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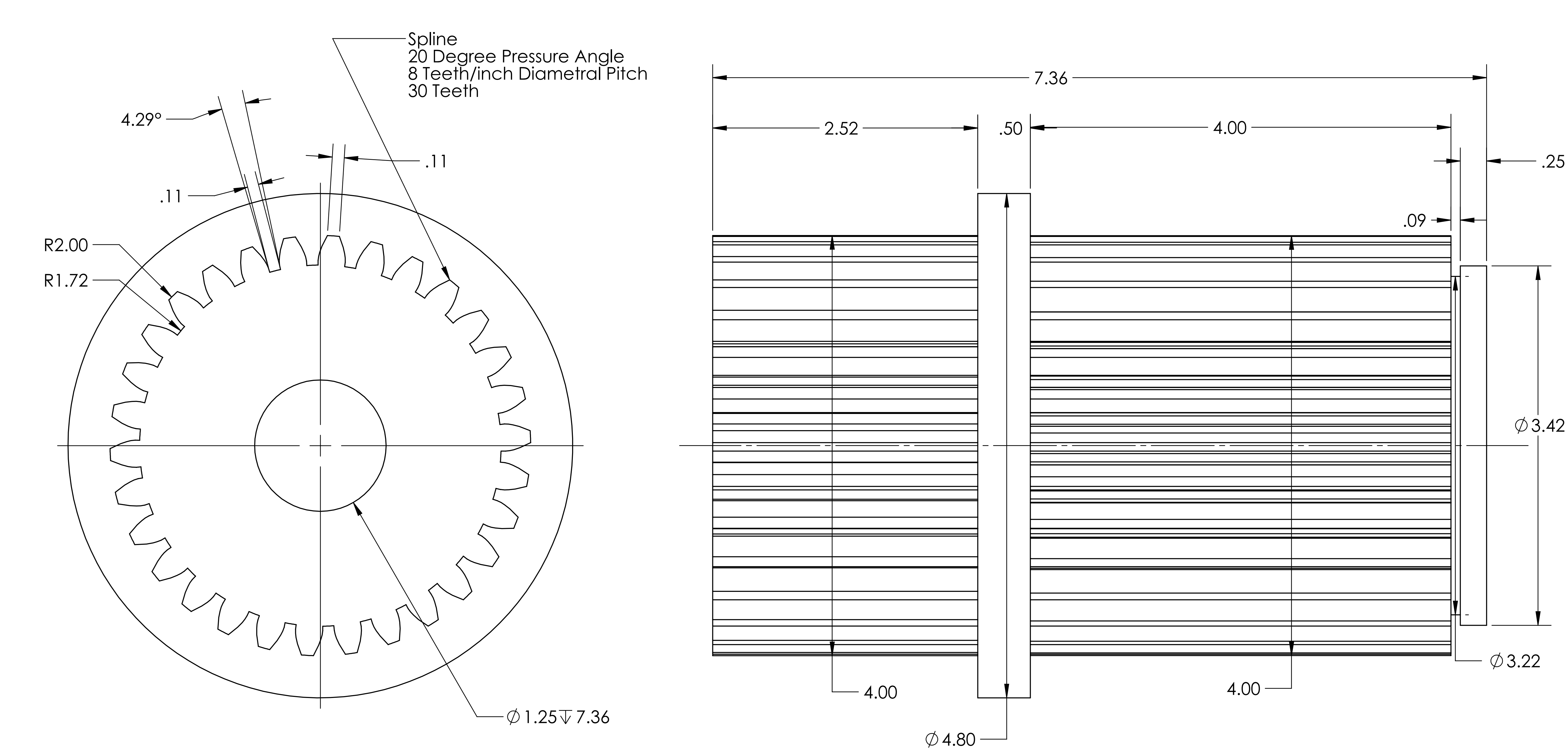

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ANGULAR: MACH ± BEND ±
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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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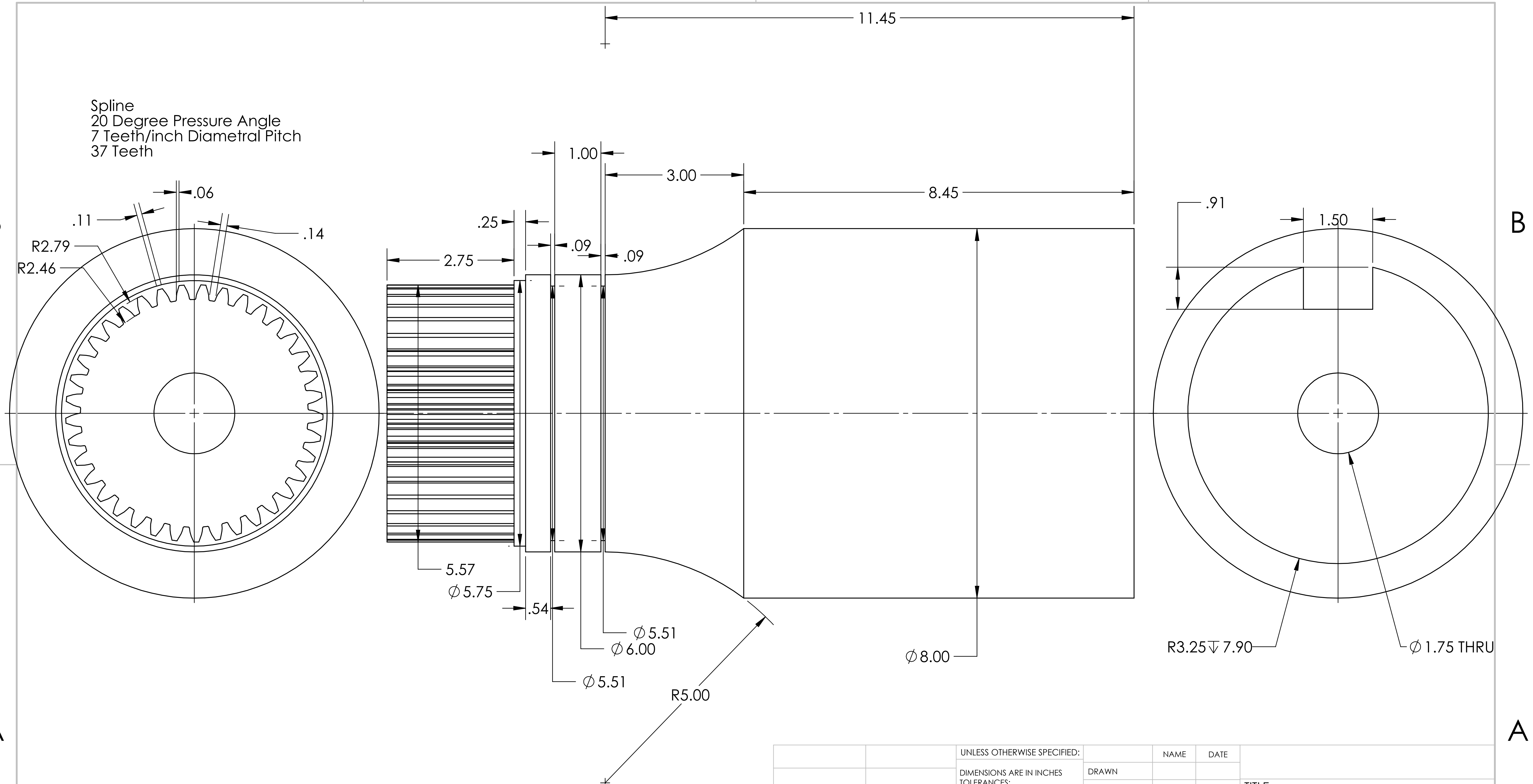
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:

Shaft D

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1



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TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

	NAME	DATE
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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

Shaft E

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION

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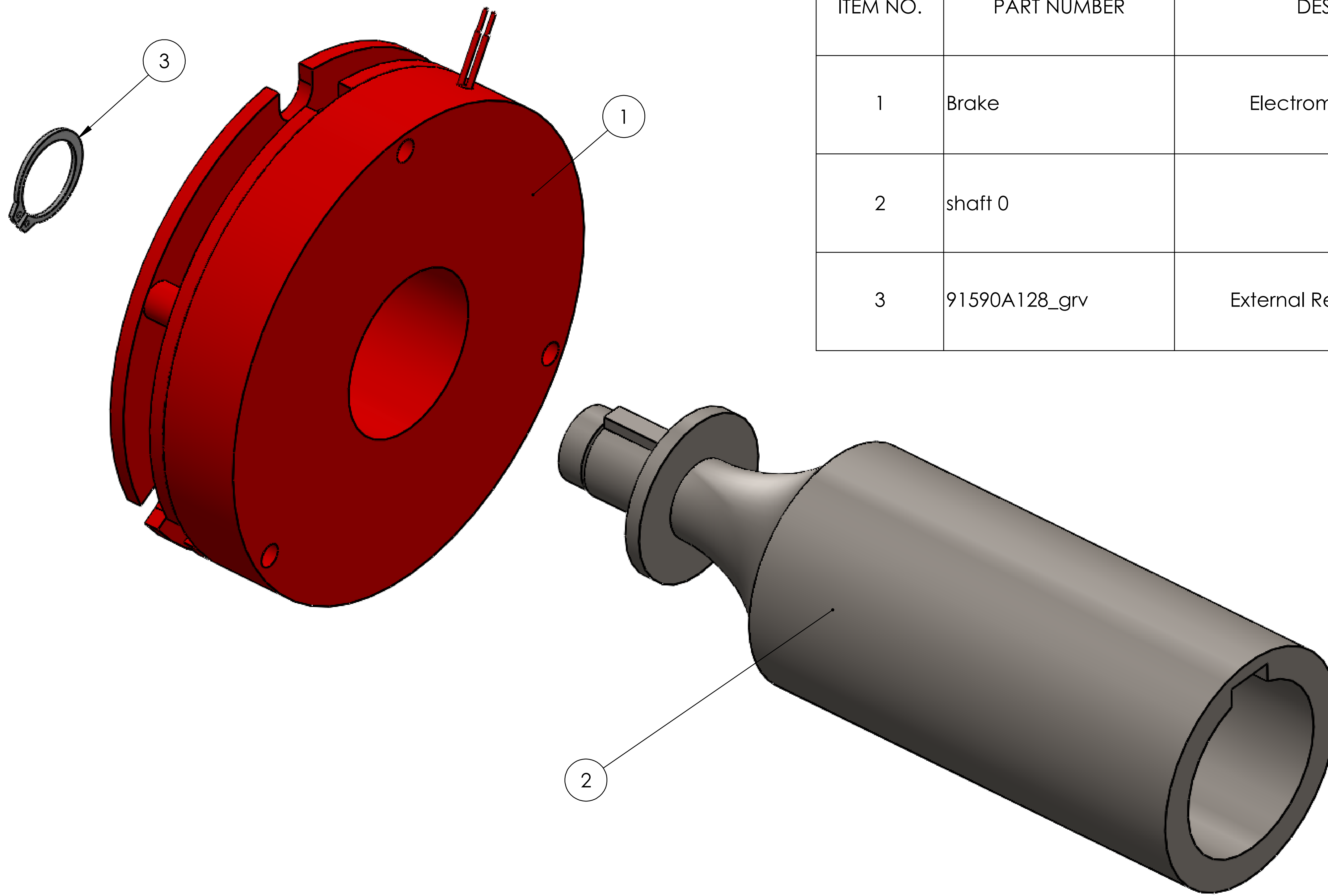
ITEM NO.	PART NUMBER	DESCRIPTION	Shaft 0 BOM/QTY.
1	Brake	Electromagnetic Brake	1
2	shaft 0		1
3	91590A128_grv	External Retainer Ring 0.75"	1



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ANGULAR: MACH ± BEND ±
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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
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FINISH
DO NOT SCALE DRAWING

	NAME	DATE
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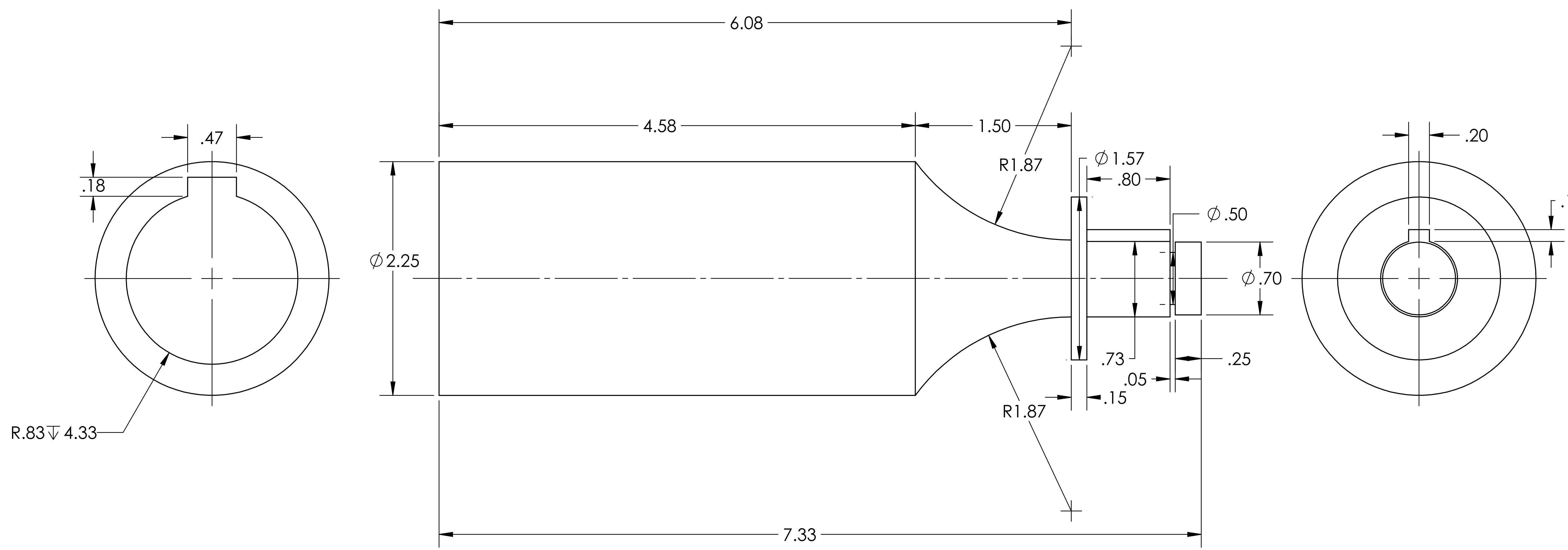
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
Shaft 0 BOM

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1



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INTERPRET GEOMETRIC TOLERANCING PER:

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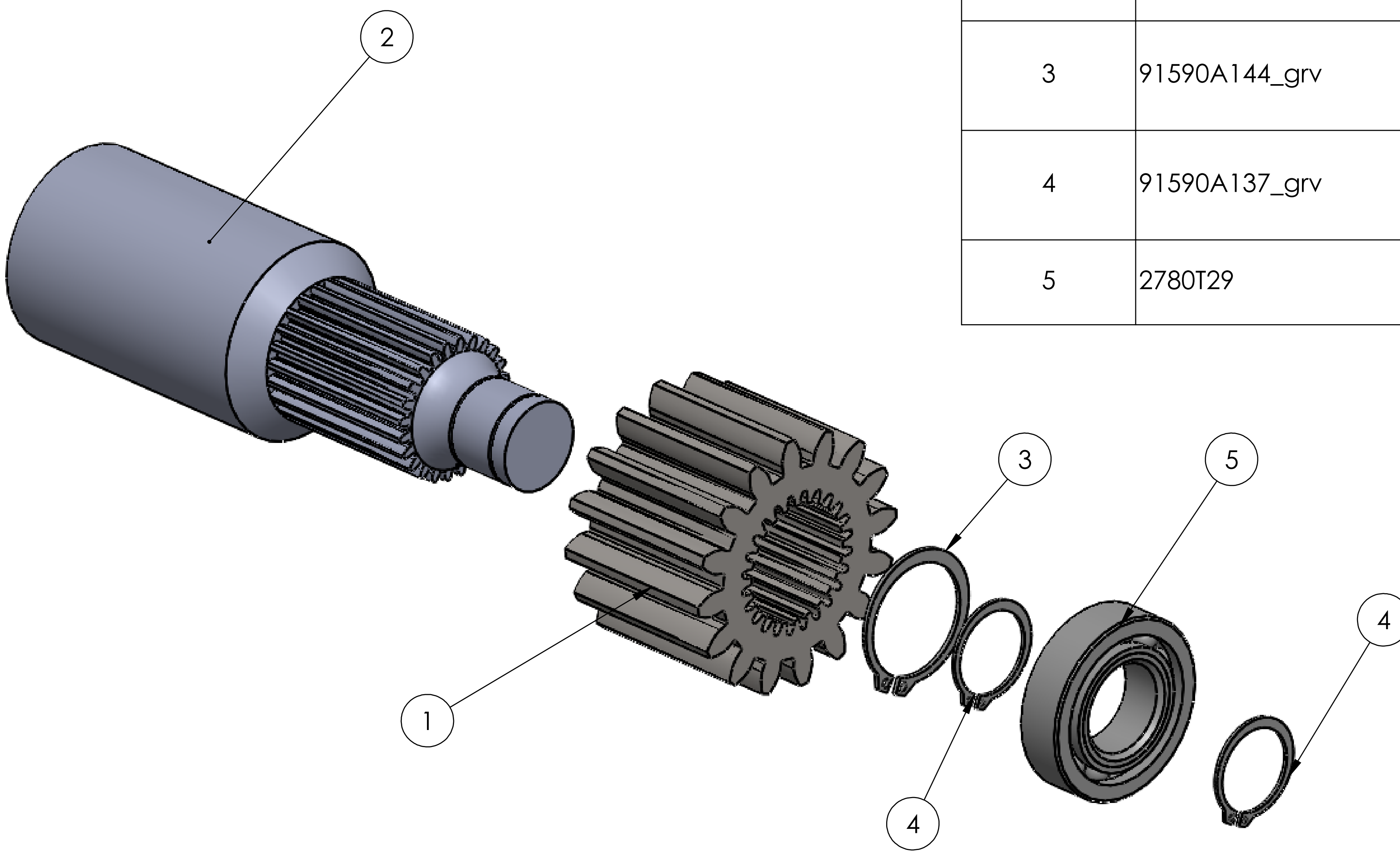
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

SIZE	DWG. NO.	REV
B	shaft 0	
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	Shaft A BOM/QTY.
1	Pinion		1
2	Shaft A		1
3	91590A144_grv	External Retaining Ring, shaft size 1.65" OD	1
4	91590A137_grv	External Retaining Ring, shaft size 1.25" OD	2
5	2780T29	High Load Ball Bearing 1.25"	1



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FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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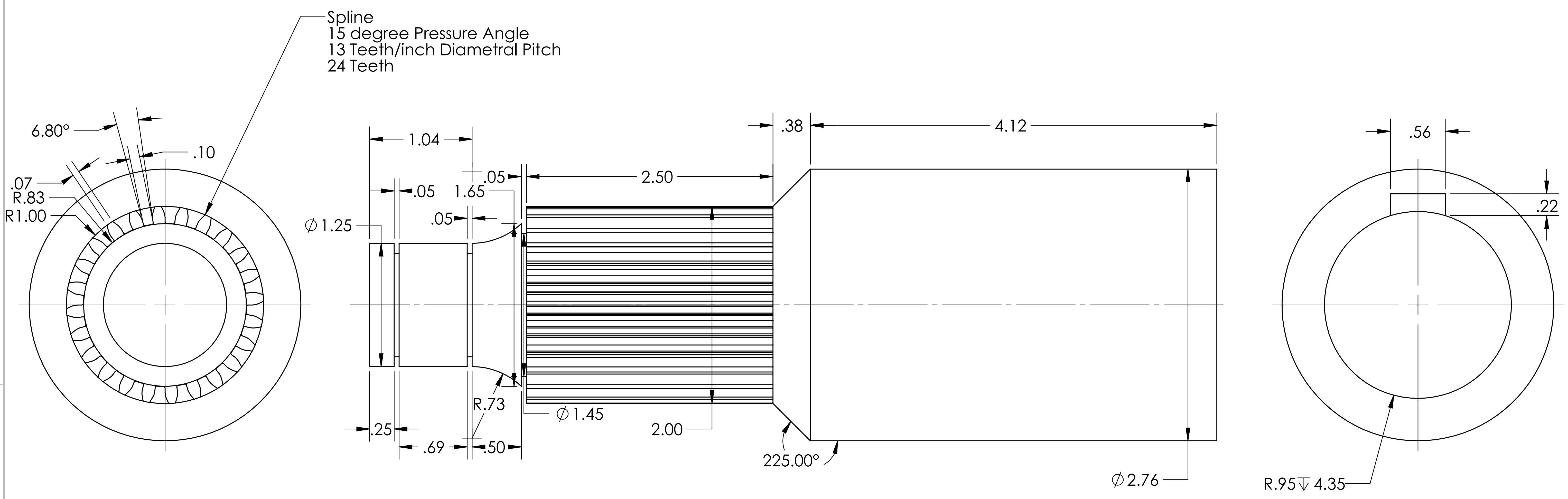
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE: Shaft A Assembly

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



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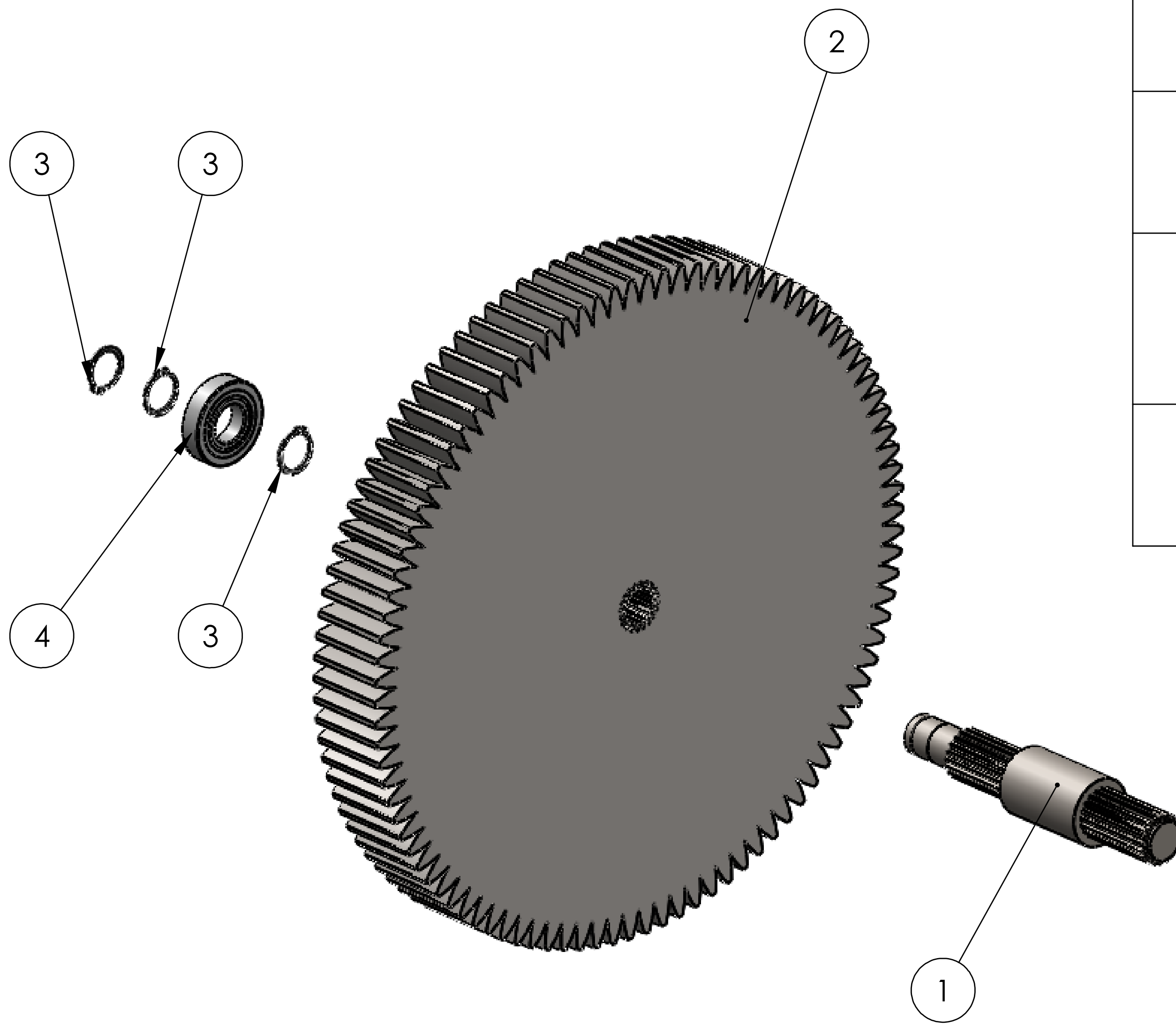
COMMENTS:		

TITLE:

SIZE B DWG. NO. Shaft A REV

SCALE: 1:1 WEIGHT: SHEET 1 OF 1



ITEM NO.	PART NUMBER	DESCRIPTION	shaft B/QTY.
1	Shaft B		1
2	Gear		1
3	91590A135_grv	External Retaining Ring, shaft size 1.25" OD	3
4	2780T67	High Load Ball Bearing 1.25"	1

		NAME	DATE
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ENG APPR.			
MFG APPR.			
Q.A.			

COMMENTS:

UNLESS OTHERWISE SPECIFIED:
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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
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INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

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NEXT ASSY	USED ON
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TITLE:

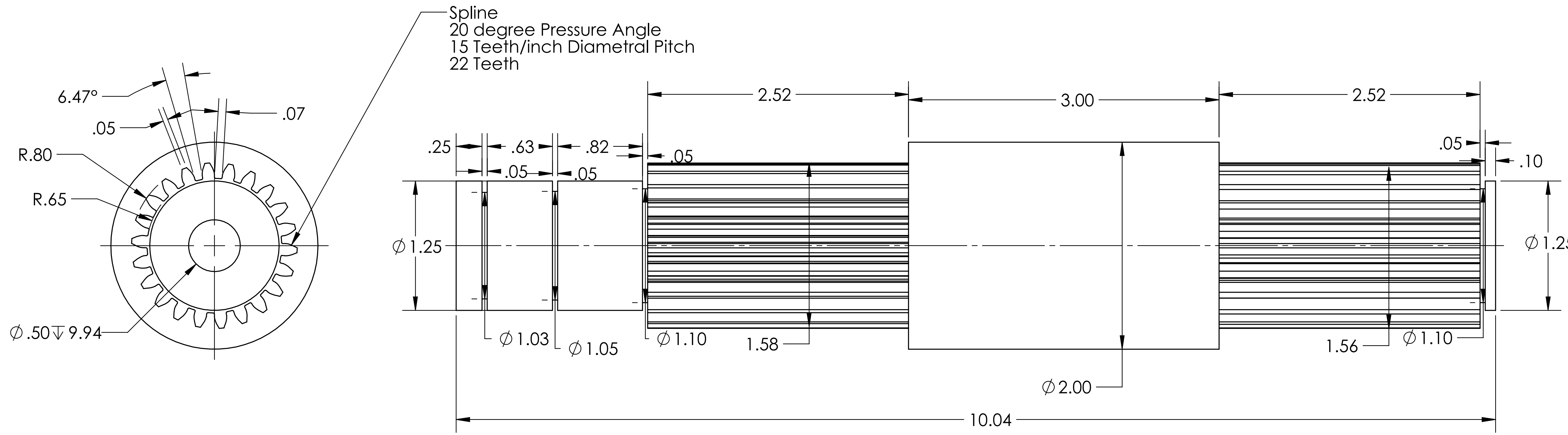
Shaft B Assembly

SIZE B | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

4 3 2 1

B



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MFG APPR.		
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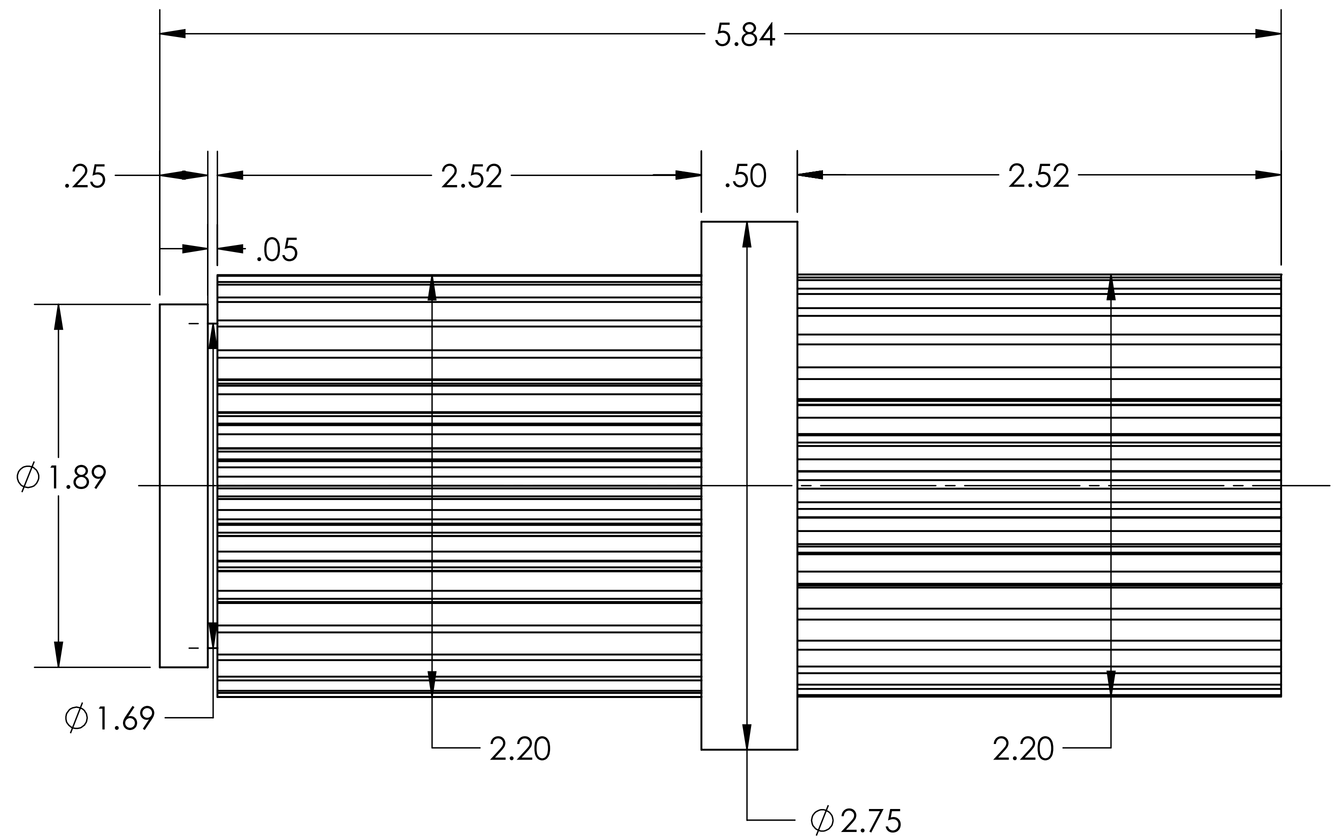
COMMENTS:

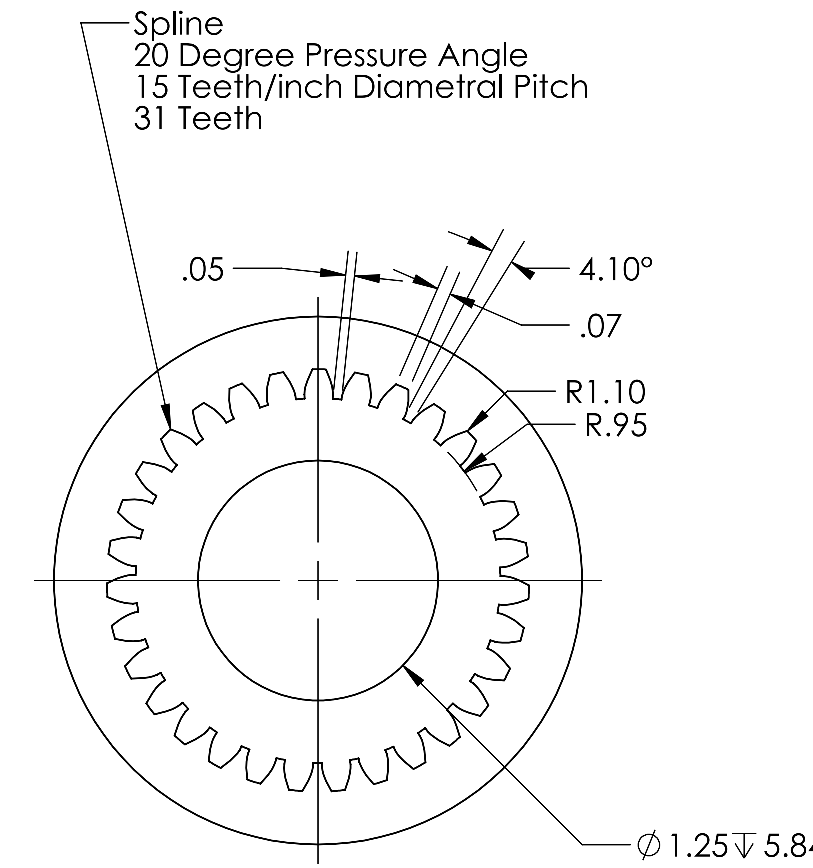
TITLE:

SIZE	DWG. NO.	REV
B	Shaft B	

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

4 3 2 1





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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

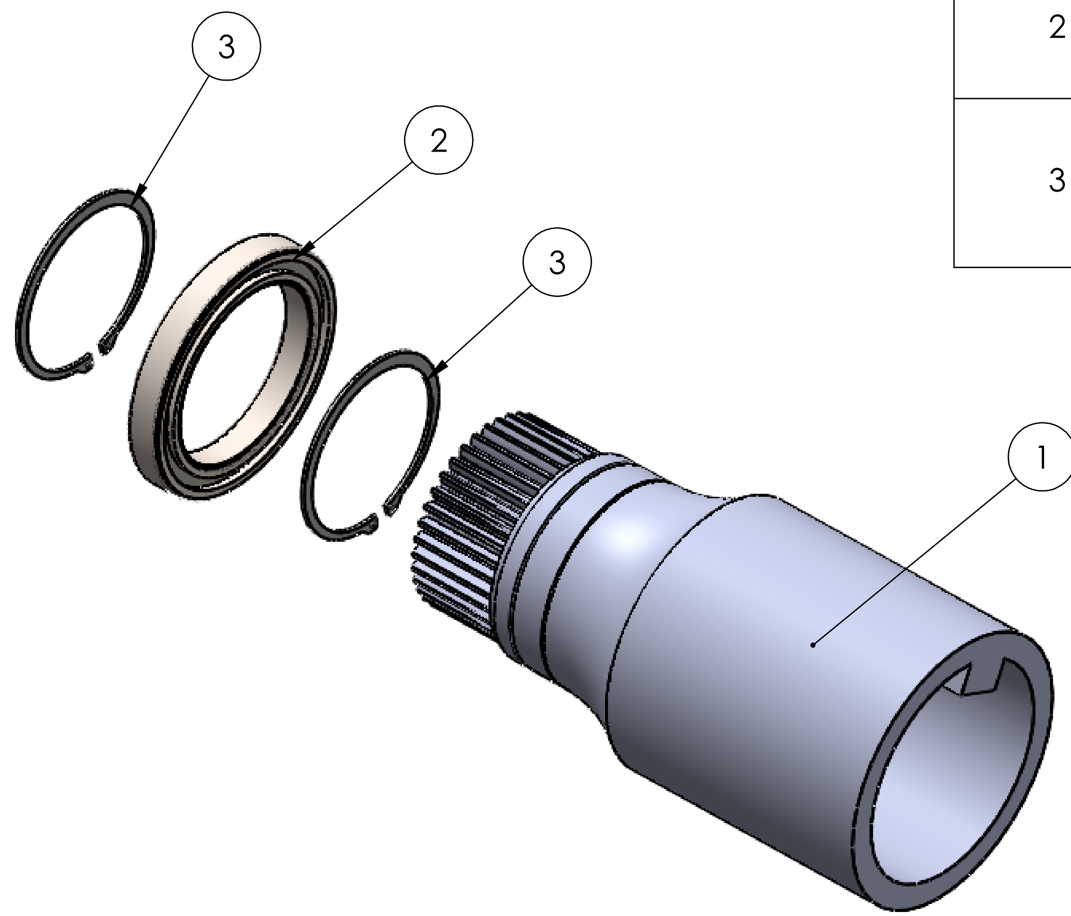
TITLE:

Shaft C

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	SHaft E BOM/QTY.
1	Shaft E		1
2	600 bearing	NSK Roller Bearing 6"	1
3	91590A280_grv	External Retaining Ring, shaft size 6" OD	2



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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

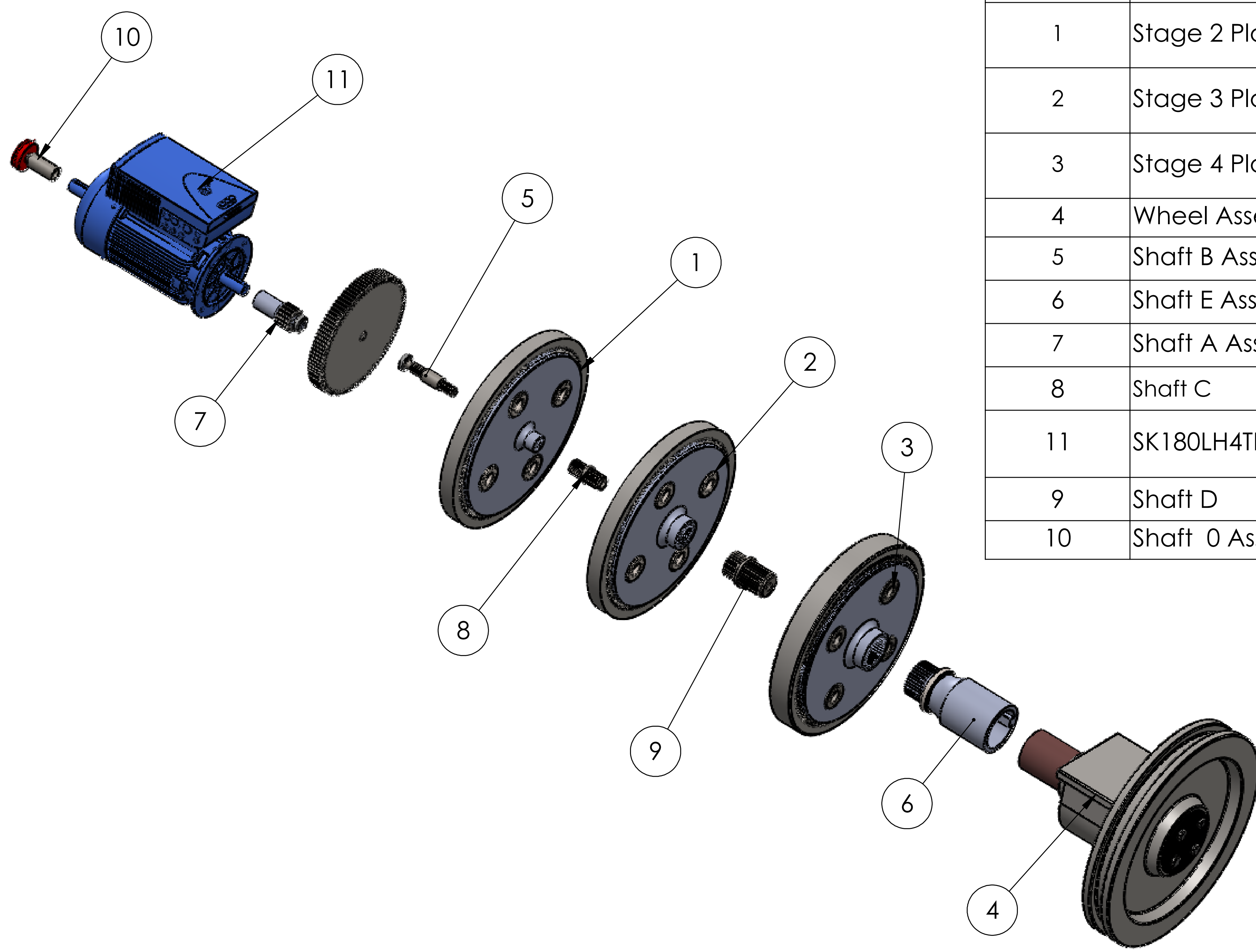
TITLE:

Shaft E Assembly

SIZE B | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only.



ITEM NO.	PART NUMBER	DESCRIPTION	Small Exploded view/QTY.
1	Stage 2 Planetary Set		1
2	Stage 3 Planetary Set		1
3	Stage 4 Planetary Set		1
4	Wheel Assembly		1
5	Shaft B Assembly		1
6	Shaft E Assembly		1
7	Shaft A Assembly		1
8	Shaft C		1
11	SK180LH4TFWE	AC induction squirrel cage motor (30Hp, 144Nm)	1
9	Shaft D		1
10	Shaft 0 Assembly		1

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

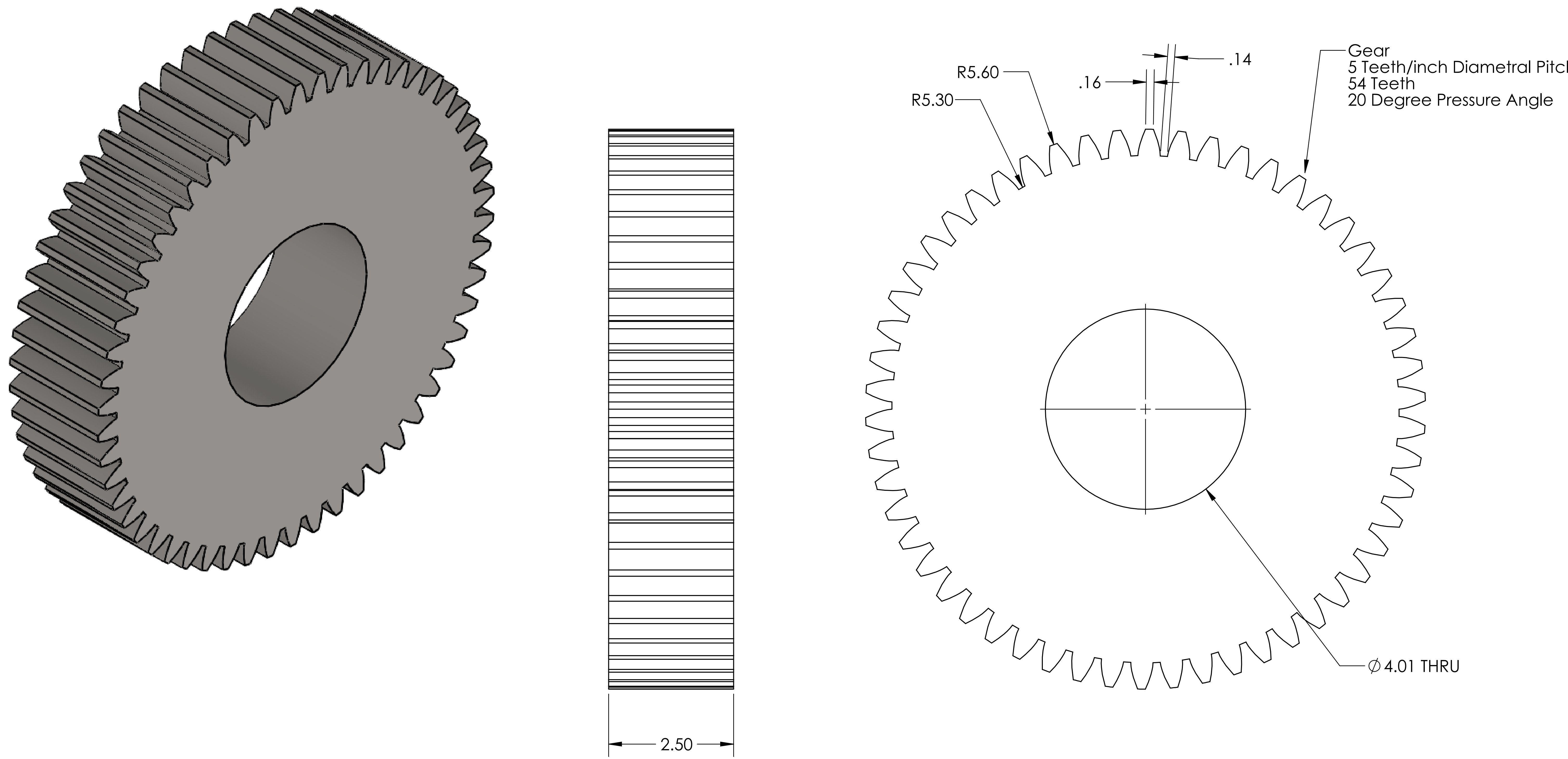
	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE: Drive System Assembly

SIZE B | DWG. NO. | REV

SCALE: 1:20 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

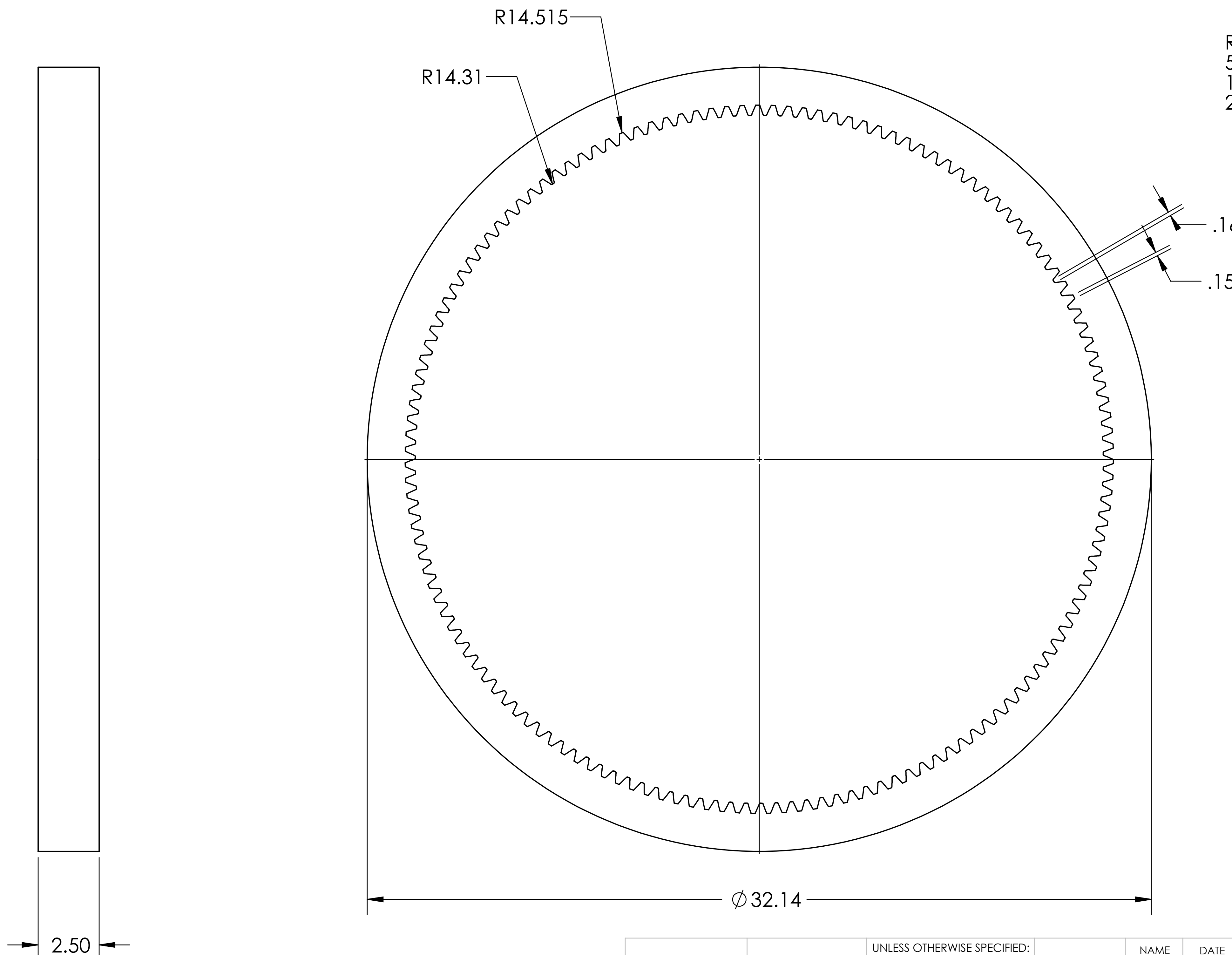
NEXT ASSY	USED ON
APPLICATION	

TITLE:

Stage 2 and 3 Planet gear

SIZE	DWG. NO.	REV
B		
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

Ring Gear
5 Teeth/inch Diametral Pitch
143 Teeth
20 Degree Pressure Angle



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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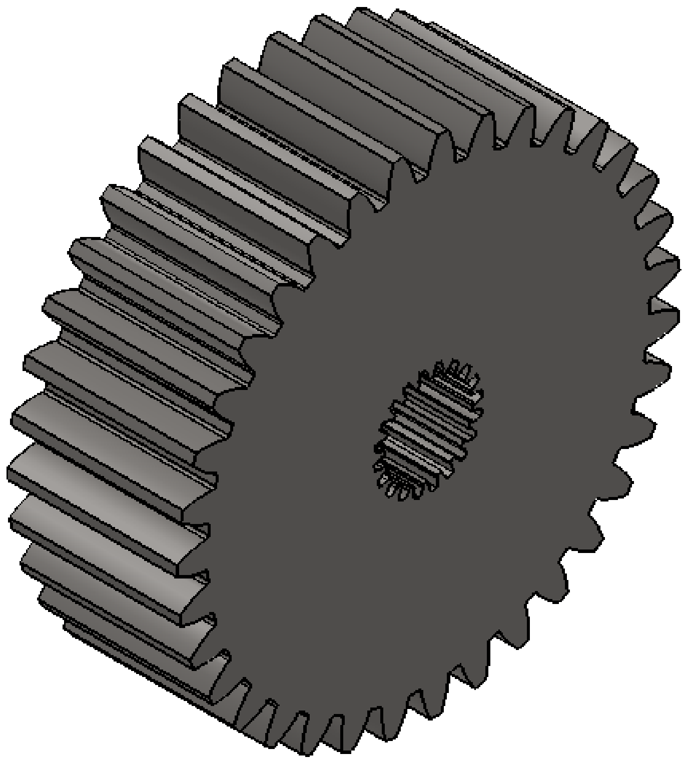
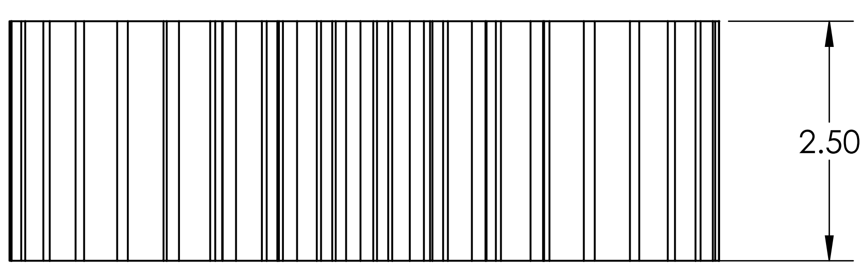
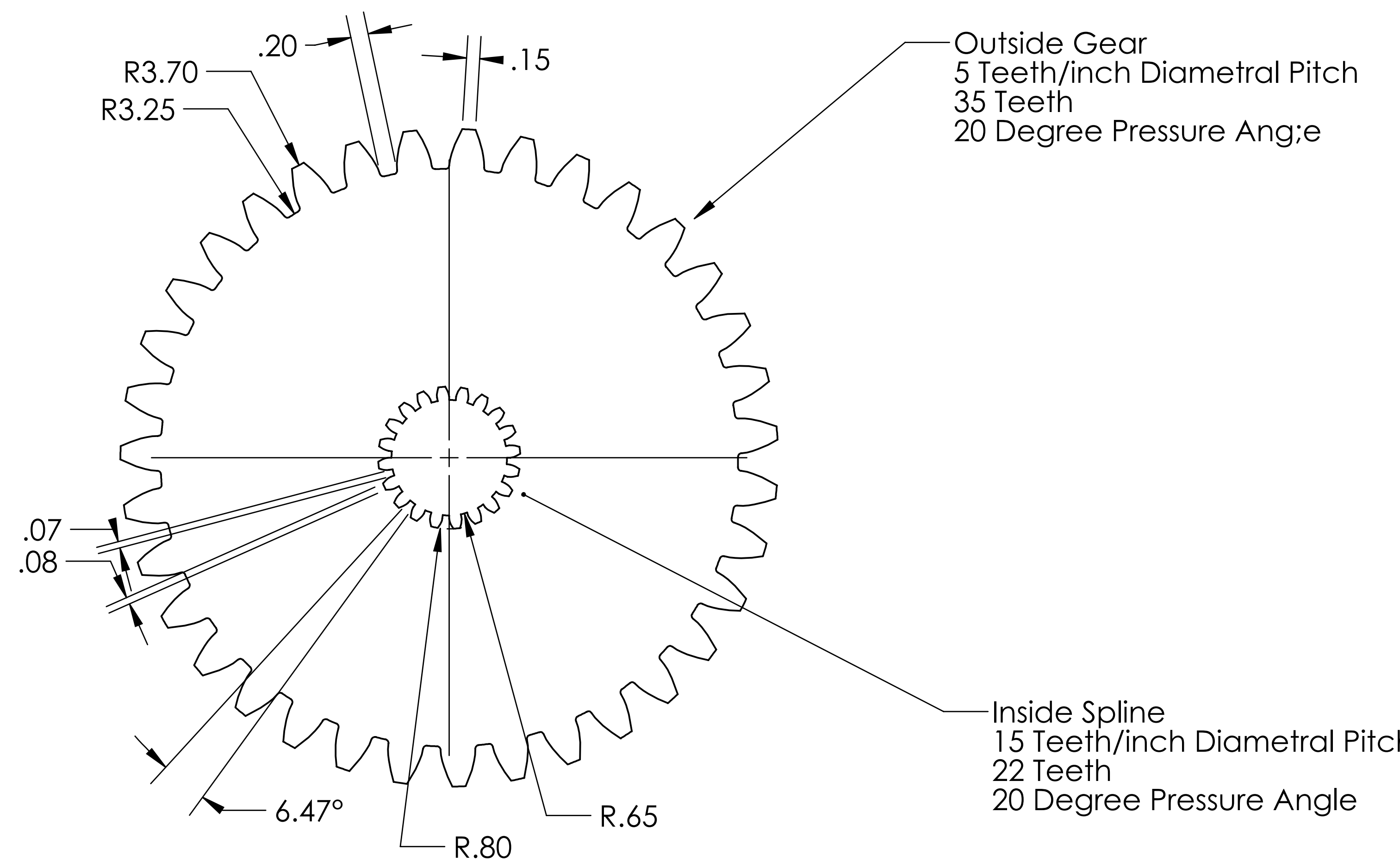
TITLE:
Stage 2 and 3 ring gear

SIZE B | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1







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NEXT ASSY	USED ON
APPLICATION	

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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

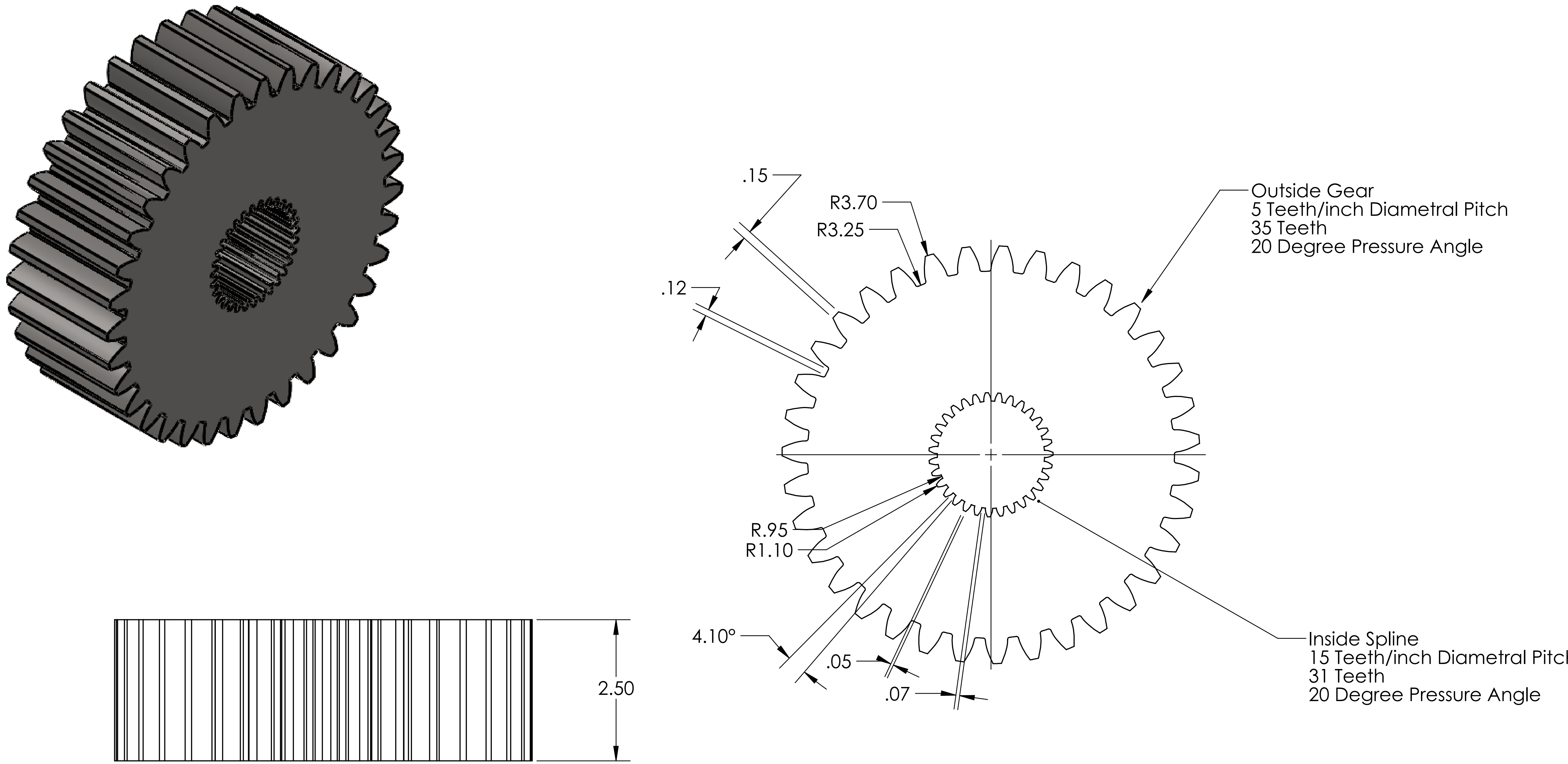
COMMENTS:

TITLE:

Stage 2 Sun gear

SIZE	DWG. NO.	REV
B		

SCALE: 1:2 WEIGHT: SHEET 1 OF 1



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NEXT ASSY | USED ON

APPLICATION

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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

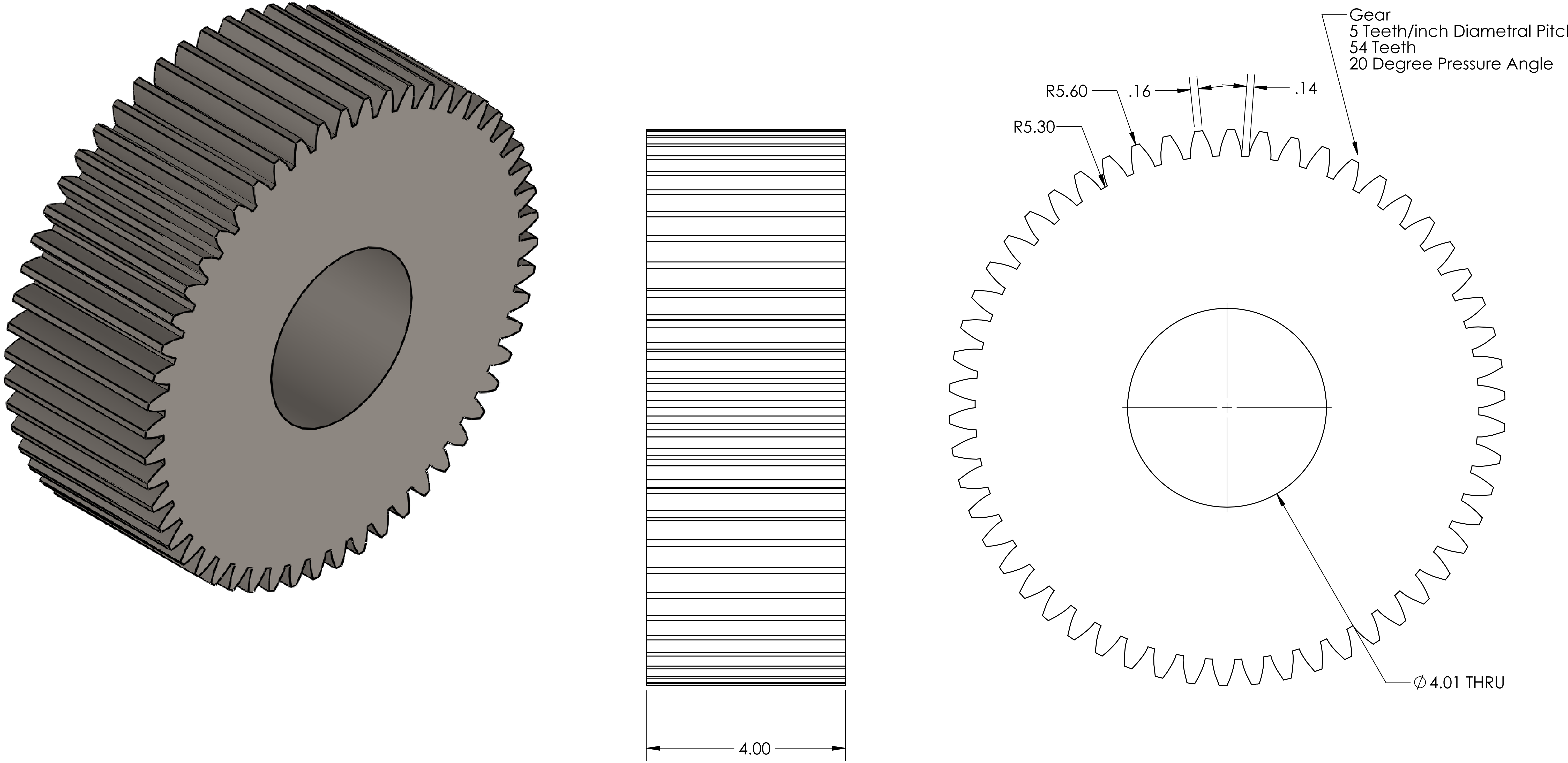
COMMENTS:

TITLE:

Stage 3 Sun gear

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



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NEXT ASSY	USED ON
APPLICATION	

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TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

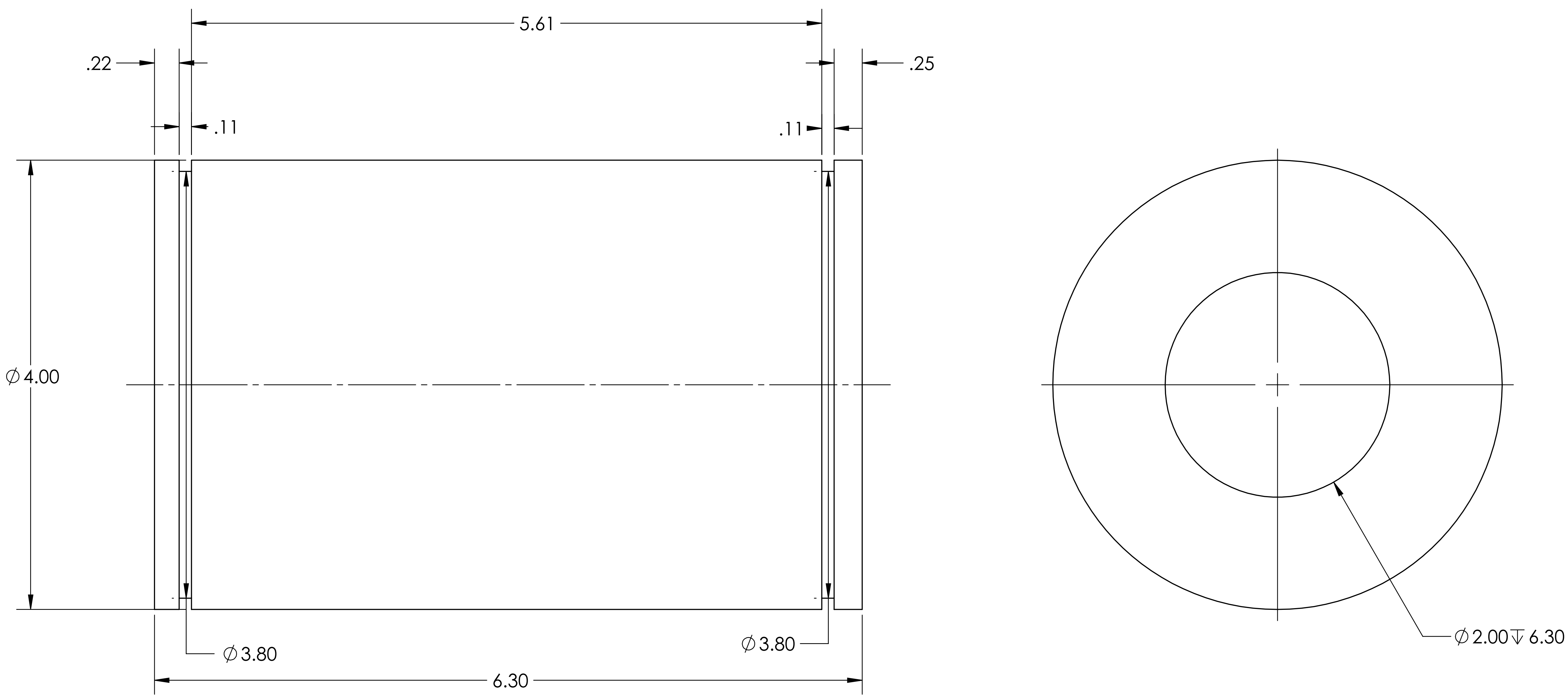
	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE: Stage 4 Planet gear

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

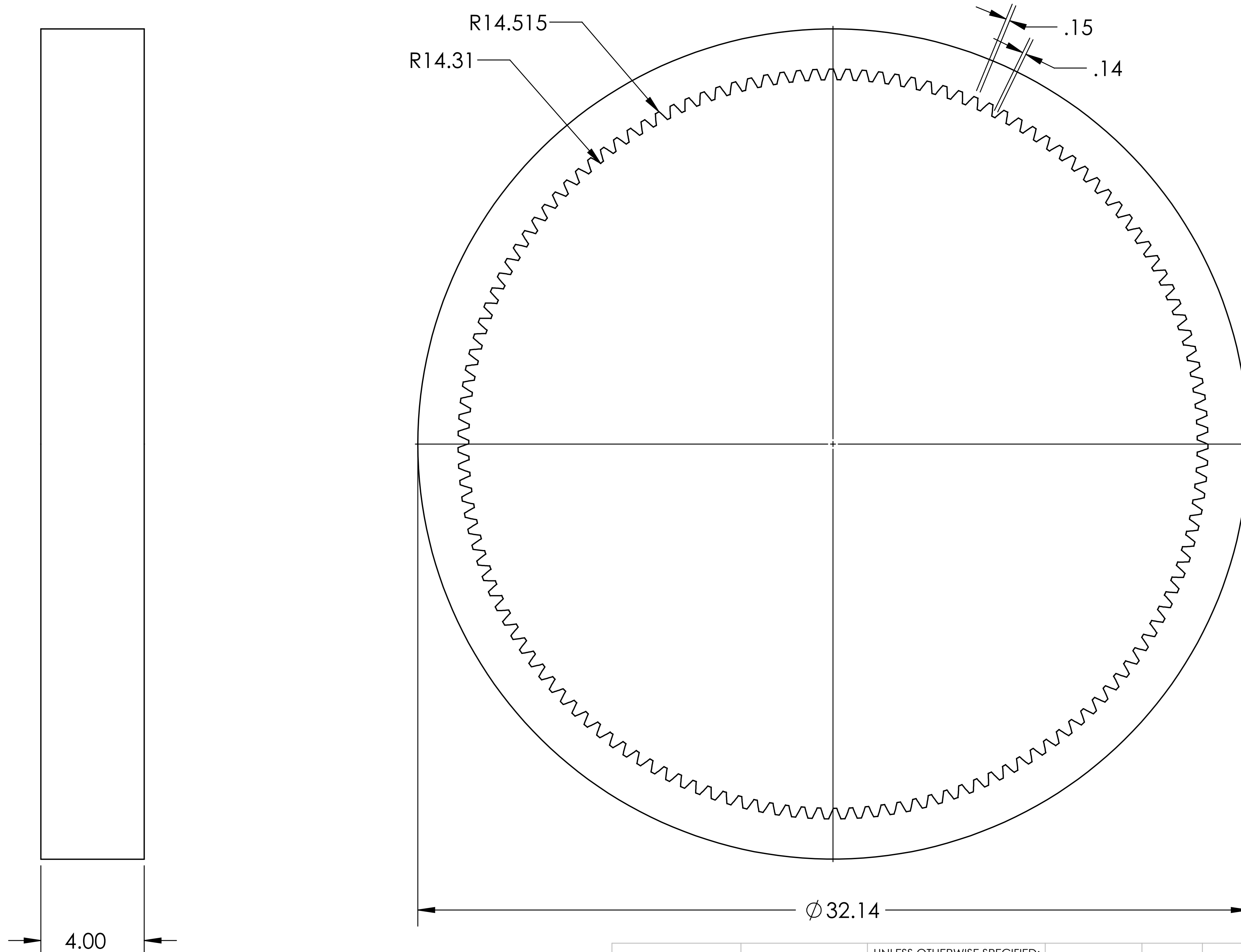
TITLE: Stage 4 Planet Pin

SIZE	DWG. NO.	REV
B		

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

4 3 2 1

B



Ring Gear
5 Teeth/inch Diametral Pitch
143 Teeth
20 Degree Pressure Angle

A

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

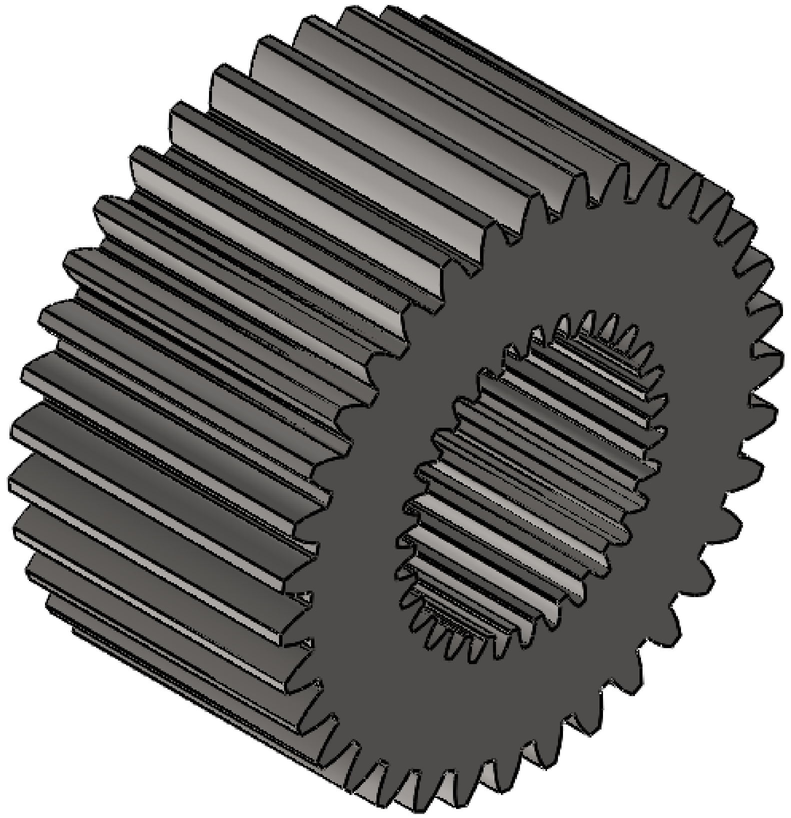
NEXT ASSY	USED ON
APPLICATION	

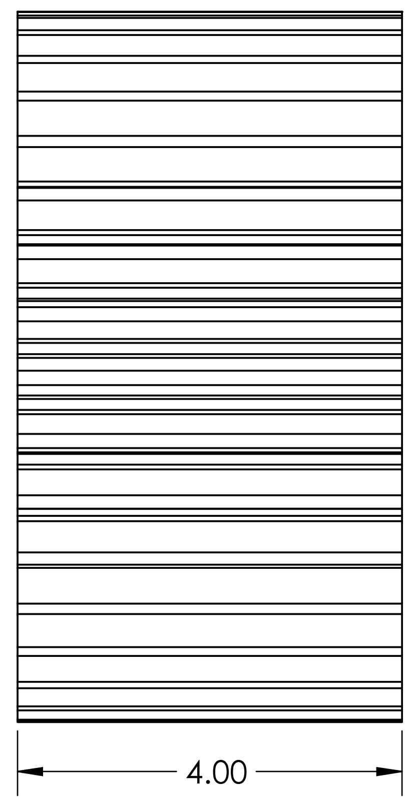
TITLE:
stage 4 ring gear

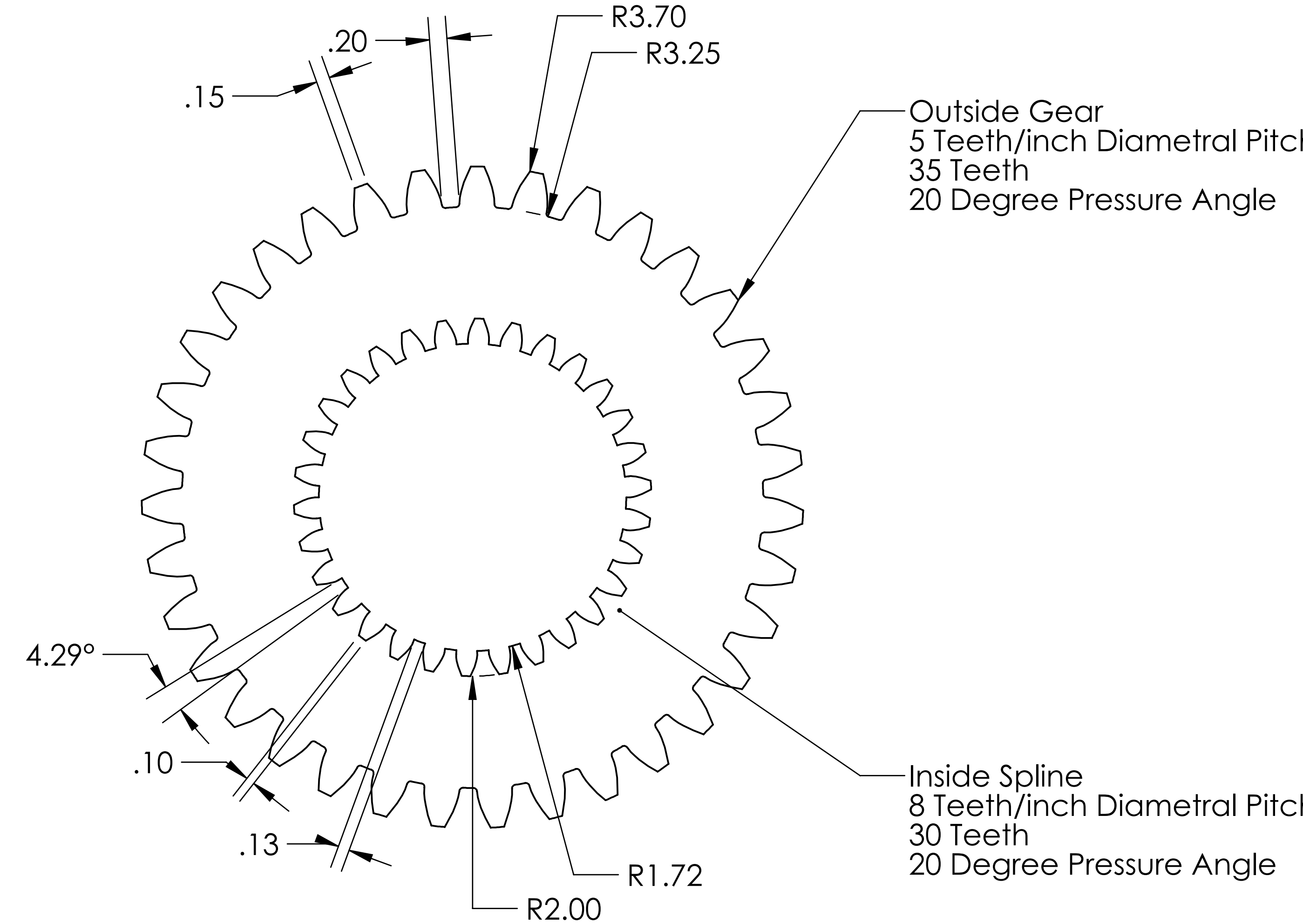
SIZE B | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

4 3 2 1







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MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

Stage 4 Sun gear

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1