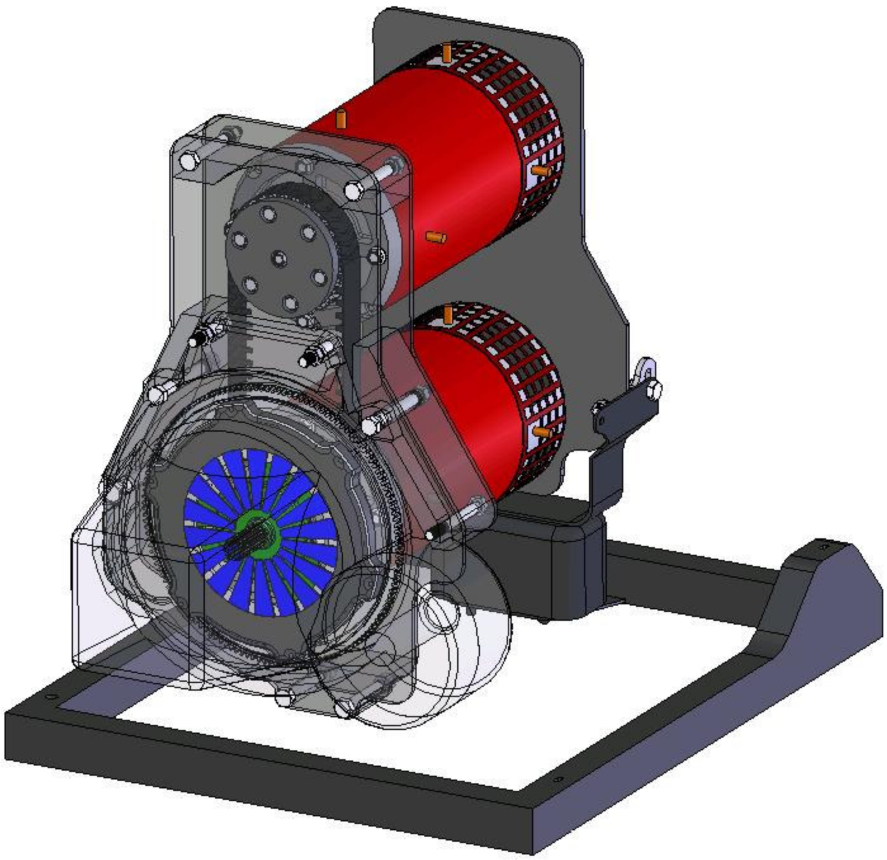
ME 495 EV

Electric Fiero Drive Train Design

Presented by Team BJ CANE

March 17, 2005



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Alex Fabbiano
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Mentored by Professor A. Emery

Problem Description

In January 2005, the BJ CANE Design Team was contracted by the University of Washington Student Electric Vehicle Project to design an electric drive train for project leader Stephen Johnsen's 1988 Pontiac Fiero. During the first week of the design process, team member responsibilities (Appendix A.1) were assigned, an initial project timeline (Appendix A.2) was drafted, and the following Problem Statement was composed: "Design a safe and reliable electric drive train for a 1988 Pontiac Fiero that best meets the needs of the customer."

BJ CANE met with Mr. Johnsen to generate and rank a set of customer requirements (Appendix A.4) and to establish project parameters. The team agreed to provide a complete Solidworks assembly of a fully functional electric drive train within a ten week design period. A budget of $5,000 was assigned for the procurement or manufacture of all drive train components including the motor, transmission, adapter plate, clutch, flywheel, pressure plate, and mounting systems.

Acceleration (to a velocity of 60 mph in less than 5 seconds) and system safety received the highest possible customer ranking. The requirements were reevaluated and modified by BJ CANE then used to define significant engineering characteristics through Quality Functional Deployment (Appendix A.5). System current received the highest relative QFD importance of 0.15 with complexity (number of parts) ranking second at 0.13.

Motor Selection

The significant engineering characteristics were entered into a Pugh Selection Matrix (Appendix B.1) to evaluate design concepts for the system motor. The concepts (Appendix B.2) included 8", 9", and 11" DC wound motors, combinations of two 8" or 9" motors in both series and parallel, and a system of multiple 6" Etek permanent magnet DC motors. The parallel combination of two 8" motors was the only design concept to rank higher than the baseline concept of one 9" motor. Although it received negative Pugh values in several important categories including cost, weight, volume, efficiency, and complexity, the dual 8" concept was selected for its superior power characteristics. One 9" motor can not achieve the acceleration required by the customer, as confirmed by the BJ CANE Vehicle Dynamics Simulation (Appendix F.3). Two Warp, 8" advanced timing, dual shaft, DC wound motors were ordered from Cloud Electric Vehicles on February 3rd.

Motor Coupling

Several design concepts were considered for transmitting power between the two motors, including a chain, a positive drive (Pd) belt, a friction belt, and a system of gears. The gear concept was eliminated for its complexity, the friction belt concept for its unattainable pretension requirements, and the chain concept for its noise production and service requirements. A spreadsheet (Appendix C.1) was created to optimize the Pd belt size. A

B-994 positive drive belt and two B-30S-MPB sprockets were ordered from Goodyear on March 16th. The parts were ordered three weeks behind schedule (Appendix A.3) due to an unexpected setback in the coupling system design.

Appendix B.3 shows the manufacturer's specifications for the Warp 8" motor. The coupling system was initially designed to connect the smaller (0.75 inch diameter) motor shafts. Design calculations and Cosmos finite element analysis later showed that the shafts would fail in torsion (Appendix C.2). The system had to be redesigned to connect the larger (1.25 inch diameter) motor shafts on the transmission side of the drive train. Several problems arose from this reconfiguration including the need for complex adapter plate geometry and for custom belt bushings (Appendix C.3).

Adapter Plate System

The adapter plate system connects the motor coupling system to the transmission, providing the primary structural support for the motors and insuring proper alignment between the drive shaft and flywheel. The system consists of four lengths of 2 x 2.5 inch bar stock bolted between two 0.25 inch plates. All components will be made of Aluminum 6061-T6 and will be manufactured at the UW Engineering Annex.

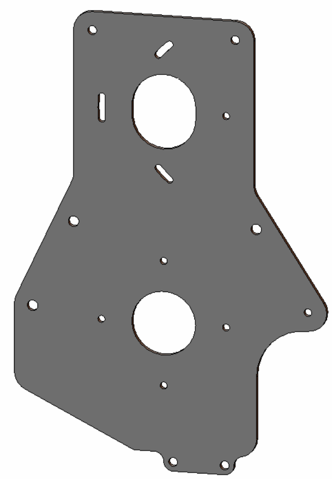


Figure 1. Adapter Plate P2

The adapter plate system geometry was designed to maximize the number of through transmission bolts and to provide housing for the belt system and flywheel. It also allows the system to obtain the PD belt pretension requirements. Figure 1 shows the P2 plate of the adapter plate system. The bolt hole geometry allows vertical translation of the upper motor for belt tension adjustment. Cosmos finite element analysis of the adapter plate system is shown in appendix C.4.

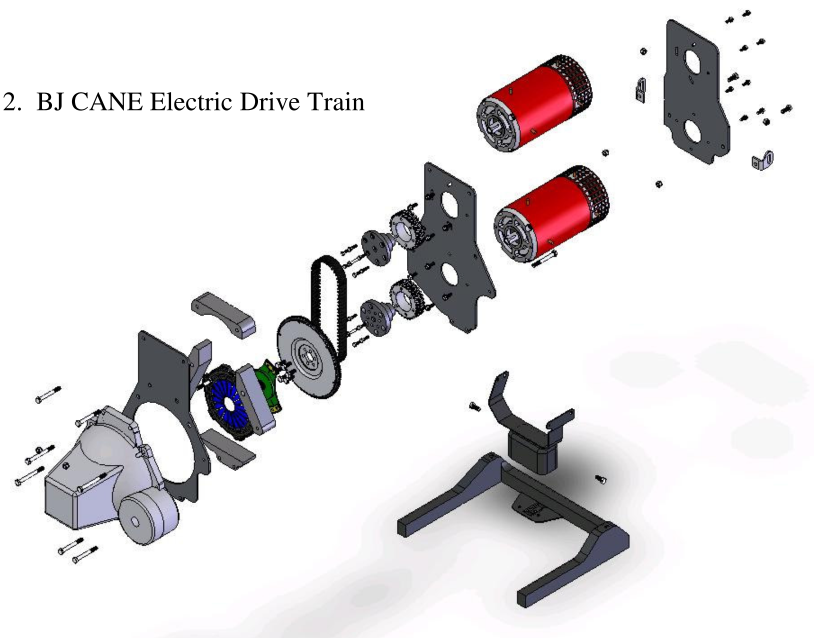
Clutch, Pressure Plate, and Flywheel

A detailed description of the clutches and flywheels considered for the system can be seen in Appendix D.1. An EZ-Lock Pro 8.5", three button, sprung hub clutch and corresponding pressure plate were ordered from Clutchnet on March 4th. Aluminum or composite racing flywheels were considered for their decreased mass and moment of inertia, but a compatible racing flywheel was not immediately available and the expense of manufacturing a custom flywheel was unjustifiable. It was decided that the flywheel salvaged from the original Fiero drive train would be reused. The future use of an aluminum flywheel is feasible for a performance upgrade if one can be procured, but is not necessary to achieve design requirements. The gearing will be removed from the current flywheel to decrease its mass and moment of inertia. Cosmos finite element analysis of the flywheel can be seen in Appendix D.2

Project Deliverables

The BJ CANE Design Team met all project requirements within the allotted design period of 10 weeks. Appendix E.1 gives a detailed parts list and budget assessment. The total system cost was $4145.86, less than 85% of the allotted budget. An exploded view of the final BJ CANE Electric Drive Train assembly is shown below in Figure 2. It should be noted that the team was unable to meet the overly optimistic goals of assembling and installing the drive train as stated in the initial project timeline (Appendix A.2a). Appendix G contains detailed drawings of each drive train component that requires manufacturing. BJ CANE composed an instruction manual for Mr. Johnsen which details the assembly and installation processes.

Figure 2. BJ CANE Electric Drive Train



The manual also contains driving instructions and directions on managing the BJ CANE Vehicle Dynamics Simulation (Appendix E.2). Simulation results (Appendix E.3) suggest the electric drive train has sufficient power to accelerate the vehicle to a velocity of 60 miles per hour from rest in less than five seconds. Figure 3 shows the first ten seconds of a projected velocity vs. time curve for a model that included drag force, tire slippage, system efficiencies and a shift time of 0.2 seconds.

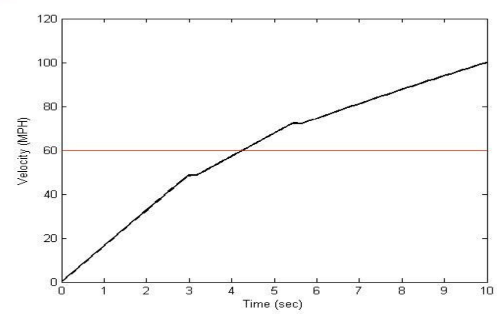


Figure 3. Velocity vs. Time Curve

The team was forced to respond to many unforeseen problems throughout the design of the BJ CANE Electric Drive Train. The inevitability of accepting design tradeoffs in order to meet customer requirements and the challenges of working for a customer without an engineering background provided a genuine introduction to real world design. A detailed log of the team's past and future progress can be seen at the project website: <http://courses.washington.edu/me495ev>.

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- **F.10 – Alignment Pin**

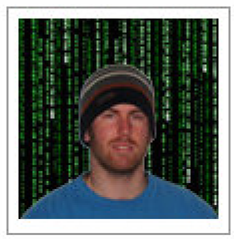
Appendix G – References

Appendix A.1 – Team Member Responsibilities

Brett Joseph
Team Management
Report Composition

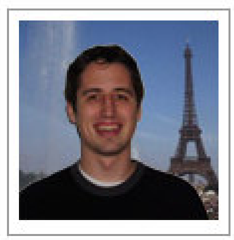
John Jolly
Reference Coordination
Drafting Coordination

Alex Fabbiano
Design Calculation
FEA Coordinator



Colin Bates
Website Coordination
Instruction Manual Composition

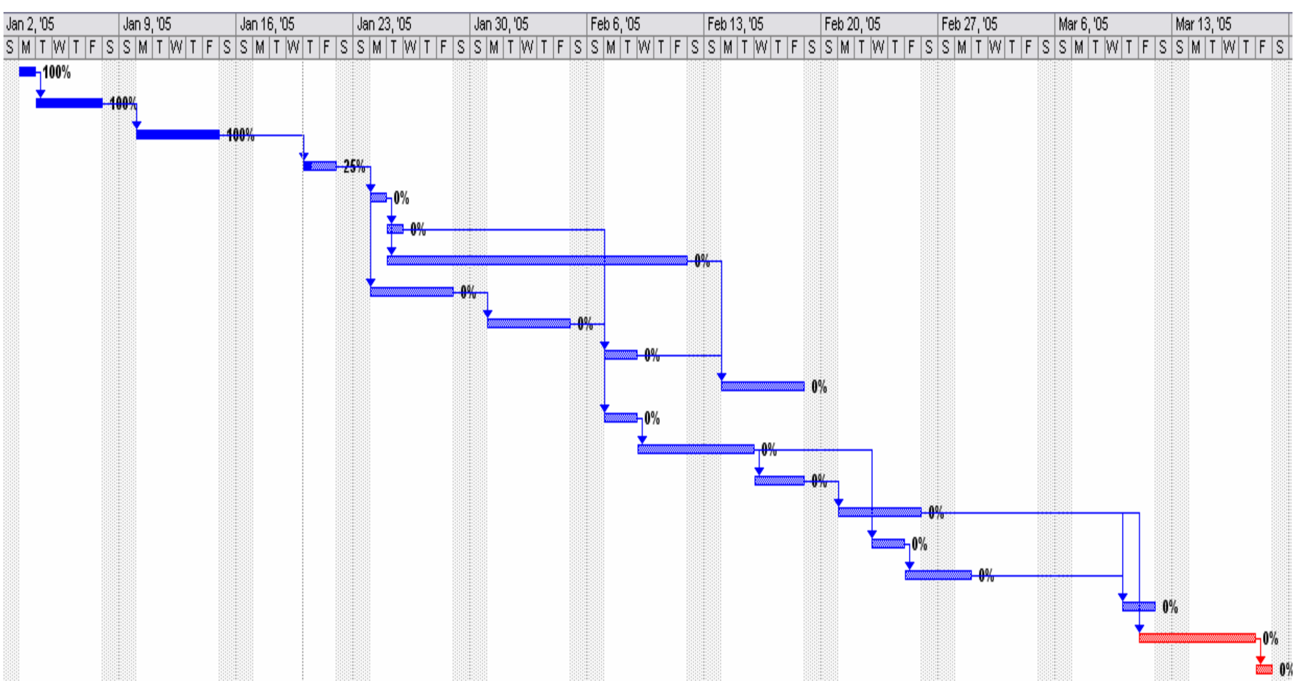
Nick Smith
Shop Coordination
CAD Assembly



Eric Yost
Simulation Design
PD Belt Optimization

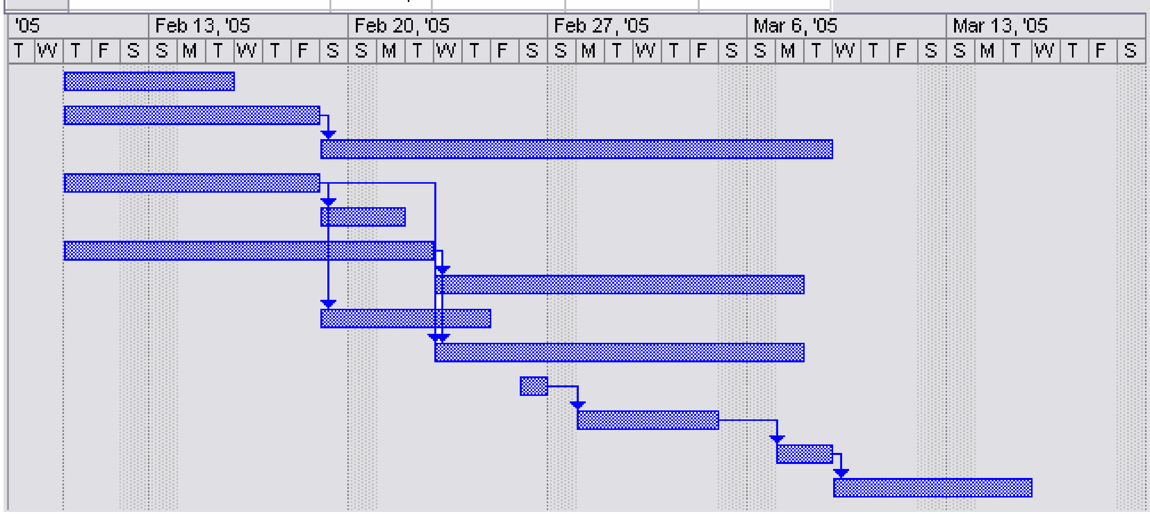
Appendix A.2a – Initial Project Timeline

Team Member Responsibilities	1 day	1/3/2005 8:00	1/3/2005 17:00	
Problem Statement	4 days	1/4/2005 8:00	1/7/2005 17:00	1
Requirements	5 days	1/10/2005 8:00	1/14/2005 17:00	2
Motor Selection	2 days	1/20/2005 8:00	1/21/2005 17:00	3
Order Motor	1 day	1/24/2005 8:00	1/24/2005 17:00	4
Obtain Physical Motor Dimensions	1 day	1/25/2005 8:00	1/25/2005 17:00	5
Motor Shipping - Receive Motor	14 days	1/25/2005 8:00	2/11/2005 17:00	5
Transmission & Safety Shield Selection	5 days	1/24/2005 8:00	1/28/2005 17:00	4
Acquire Transmission	5 days	1/31/2005 8:00	2/4/2005 17:00	8
Select Clutch and Flywheel	2 days	2/7/2005 8:00	2/8/2005 17:00	9,6
Acquire / Manufacture / Assemble Clutch-Flywheel	5 days	2/14/2005 8:00	2/18/2005 17:00	10,7
Design Adaptor Plate	2 days	2/7/2005 8:00	2/8/2005 17:00	6,9
Manufacture Adaptor Plate	5 days	2/9/2005 8:00	2/15/2005 17:00	12
Design Mounting Basketry / Misc.	3 days	2/16/2005 8:00	2/18/2005 17:00	13
Manufacture Mounting Basketry / Misc.	5 days	2/21/2005 8:00	2/25/2005 17:00	14
Assemble Drive train	2 days	2/23/2005 8:00	2/24/2005 17:00	13
Test Drive train	2 days	2/25/2005 8:00	2/28/2005 17:00	16
Install Drive train	2 days	3/10/2005 8:00	3/11/2005 17:00	17,15
Compose Report	5 days	3/11/2005 8:00	3/17/2005 17:00	
Present	1 day	3/18/2005 8:00	3/18/2005 17:00	19



Appendix A.2b – Final Project Timeline

	Task Name	Duration	Start	Finish	Predecessors
1	Order Clutch	4 days	Thu 2/10/05	Tue 2/15/05	
2	Design Flywheel	7 days	Thu 2/10/05	Fri 2/18/05	
3	Manufacture/Purchase Fly	14 days	Sat 2/19/05	Tue 3/8/05	2
4	Finish Coupling system	7 days	Thu 2/10/05	Fri 2/18/05	
5	Order Goodyear parts	2 days	Sat 2/19/05	Mon 2/21/05	4
6	Design Adapter Plate	10 days	Thu 2/10/05	Tue 2/22/05	
7	Manufacture Adapter Plate	10 days	Wed 2/23/05	Mon 3/7/05	6
8	Design Shielding	5 days	Sat 2/19/05	Thu 2/24/05	4
9	Design Engine Mount syste	10 days	Wed 2/23/05	Mon 3/7/05	6,4
10	Run Dynamometer	1 day	Sat 2/26/05	Sat 2/26/05	
11	MATLAB simulation	5 days	Mon 2/28/05	Fri 3/4/05	10
12	Calculate Shift Points	2 days	Mon 3/7/05	Tue 3/8/05	11
13	Write Instruction Manual	5 days	Wed 3/9/05	Tue 3/15/05	12



Appendix A.3 – Customer Requirements

Customer/ Stakeholder Requirements	Customer Ranking	Engineering Ranking
0-60mph in less than 5 seconds (the lower the better)	5	5
"Good Range" (~30 miles or more)	3	3
Luggage (cargo space)	3	2
Safe	5	5
Street Legal	5	5
Easy Maintenance	4	4
Limited Structural Modification	3	5
Compact	3	3
Waterproof	4	4
Lightweight	3	1
Reliable	4	5
Simple Design	3	4
Low Cost	5	5
Appearance	1	1

Appendix A.4 – Quality Functional Deployment

Customer and Stakeholder Requirements	Engineering Characteristics												
	Cost ($)	*Voltage (V)*	*Current (Amps)*	*Number of Parts (#)*	*Weight (lbs)*	*Lifespan*	*Power (HP)*	*Volume (in^3)*	*Efficiency*	*Power Drop Off (RPM)*	*Torque/RPM*	*Max Torque (ft-lb)*	**Engineering Ranking**
1. *0-60 in < 5 seconds (fast)*	3	3	9		9		9		1	3	1	9	5
2. *Good Range (≈30 miles)*	9	9	9		9		1		9				3
3. *Cargo Room*				1				9					2
4. *Safe*		9	9										5
5. *Street Legal*													5
6. *Easy maintenance*				9	3	9		1					4
7. *Limited Structural Mods*	1							9					5
8. *Compact*				3				9					3
9. *Water Proof*	3			1									4
10. *Light Weight*	3			3	9								2
11. *Simple*	1			9					3				4
12. *Low cost*	9			3									5
13. *Looks Good / Clean*	1												2
14. *Reliable*				9		9							5
Absolute Importance	59	87	117	96	84	81	48	94	32	15	5	45	
Relative Importance	0.08	**0.11**	**0.15**	**0.13**	**0.11**	0.11	0.06	**0.12**	0.04	0.02	0.01	0.06	

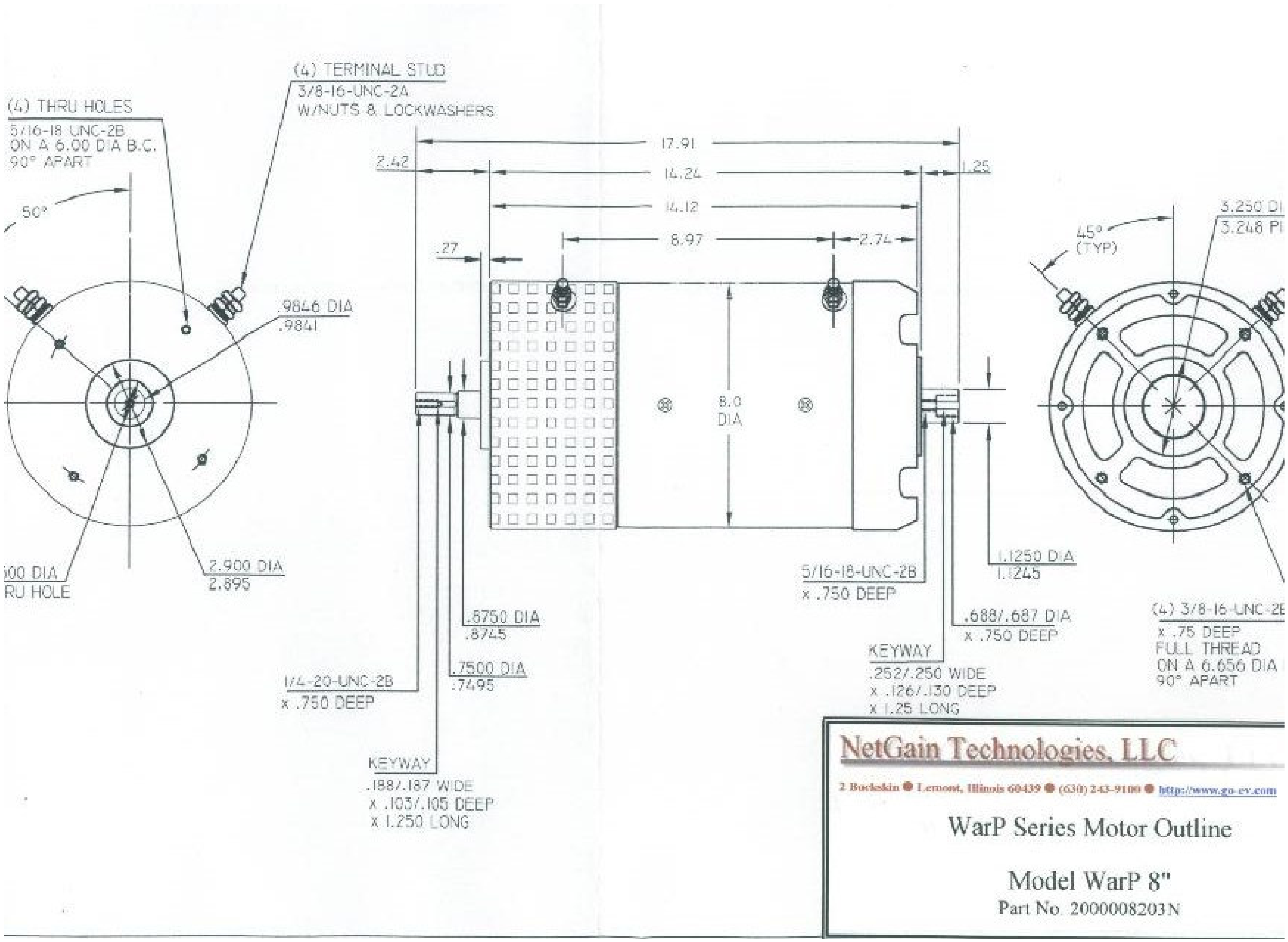
Appendix B.1 – Pugh Selection Matrix

Engineering Characteristics	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6
Cost ($)	-		-		-	-
Voltage (V)	-		+	-		
Current (amps)	+		+			
Complexity			-	-	-	-
Weight (lbs)	-		-	+	-	-
Lifespan (yrs)	+		+		+	
Power (hp)	-		+		+	+
Volume (in^3)	-		-	-	-	-
Efficiency (%)	-		-	-	-	-
Torque drop off	-		+			
Max torque (ft*lb)	+		+			
Total	**-4**	**0**	**+1**	**-3**	**-3**	**-4**

Appendix B.2 – Motor Design Concepts

Design Concepts		Cost	Weight
1	11" DC wound motor	$2,800	200 lbs
2	9" DC wound motor	$1,495	185 lbs
3	Duel 8" DC wound motor	$2,600	224 lbs
4	6 Etek Permanent Magnet Motors	$2,100	124.8 lbs
5	Duel 9" DC wound motor	$2,990	370 lbs
6	8" and 9" DC wound motors	$2,795	297 lbs

Appendix B.3 – Warp Motor Specifications



Appendix C.1 – Positive Drive Belt Size

Power Requirements		
Max Torque (ft-lbs)	230	
Max RPM	7000	
Geometric Requirements		
Min Motor Spacing (in)	10	11.2992126
Max Motor Spacing (in)	14	
Min Gear Radius (ft)	0	0.219307992
Max Gear Radius (ft)	0.833333333	
Safety Requirements		
Factor of Safety	1.8	1.894165059
Cost Requirements		
Max Cost ($)	500	381.4753567

PD BELT

Eagle PD Belt	**Sprocket**	**pitch (mm)**	14	
Name	Blue	Green	~~Orange~~	~~Red~~
Radius (ft)	0.219307992	0.20468746	~~0.409374919~~	~~0.409374919~~
Radius (in)	2.631695909	2.456249515	~~4.912499031~~	~~4.912499031~~
Circumfrence (ft)	1.377952756	1.286089239	~~2.572178478~~	~~2.572178478~~
Number of teeth	30	28	~~56~~	~~56~~
Width (mm)	37	54.5	~~72~~	~~107~~
Cost ($ / sprocket)	137.896	128.75	~~300.79268~~	~~388.789984~~

	Belt	**pitch (mm)**	~~14~~	
Width (mm)	35	52.5	~~70~~	~~105~~
Tensile Strength (lbf)	3050	13575	~~18100~~	~~27150~~
Modulus (lbf for 100% elong)	257000	384000	~~514000~~	~~771000~~
Weight (lbf / foot)	0.14	0.21	~~0.28~~	~~0.42~~
Length (in)	39.13385827	39.13385827	~~0~~	~~0~~
Number of teeth	71	71	~~0~~	~~0~~
Initial Tension (N)	2494	3745	~~4989~~	~~7562~~
Cost ($ / belt)	105.6833567	178.24	~~97.63725~~	~~147.7145~~

Initial Tension (lbf)	560.6737077	841.9097976	~~1121.572224~~	~~1700.005845~~
Max Hoop Tension (lbf)	0.781046189	1.02056702	~~5.443024105~~	~~8.164536158~~
Max Force 1 on Belt (lbf)	1610.208141	1966.594708	~~1688.84742~~	~~2270.002553~~
Max Force 2 on Belt (lbf)	-487.2986337	-280.7339792	~~565.1830766~~	~~1146.338209~~
Max Force on Belt (lbf)	1610.208141	1966.594708	~~1688.84742~~	~~2270.002553~~

Appendix C.2 – Motor Shaft Analysis

Analysis of motor shaft to was done to determine location of the coupling system, if the bushing could be positioned on the large 1.125in diameter shaft or the thin 0.75in diameter shaft.

Thin Shaft Test: Diameter of 0.75in.

A torque of 250 ft-lbs was applied to the center section to simulate the motor torque. By fixing the keyway it was possible to simulate the max forces on the motor shaft. The Shaft material (type obtained from Netgain) was 1144 Stressproof steel. Stressproof is a classification of steel made by General Steel Warehouse INC, and has a yield strength of 100,000 psi.
A max von Mises stress of 1.607e5 psi was seen on the thin shaft pictured below. This corresponds to a safety factor of 0.62, well into the failure mode of the material.

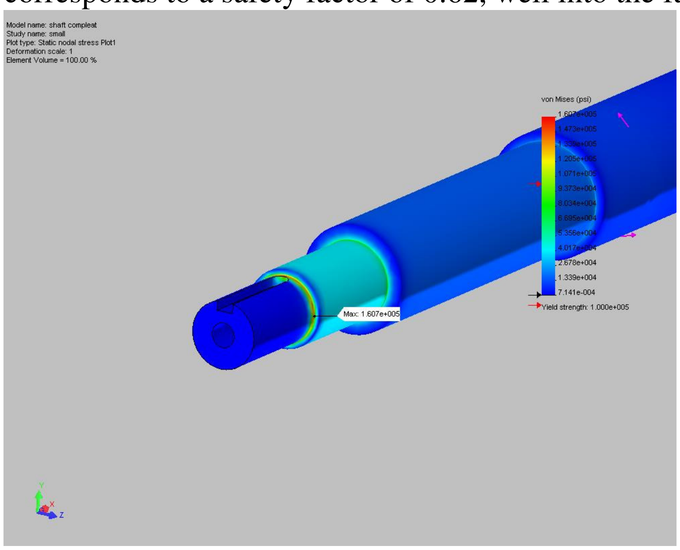


Figure 1: Max Von Mises Stress of 1.607e5 psi.

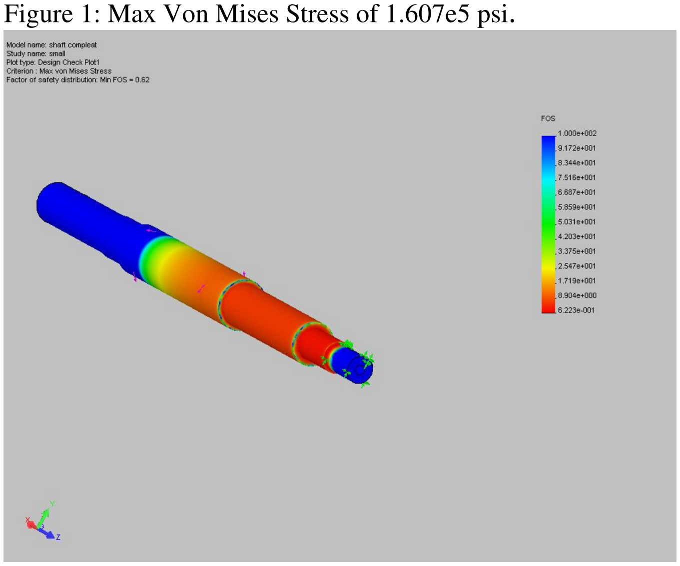


Figure 2: Max Factor of Safety 2.

Appendix C.2 – Motor Shaft Analysis

Large Shaft Test: Diameter of 1.125in

Using the same setup as the thin shaft, except the large side was fixed instead.

A max von Mises stress of 3.312e4 psi was seen on the large shaft in figure 3 below. This corresponds to a safety factor of 2.5 well into the failure mode for the shaft

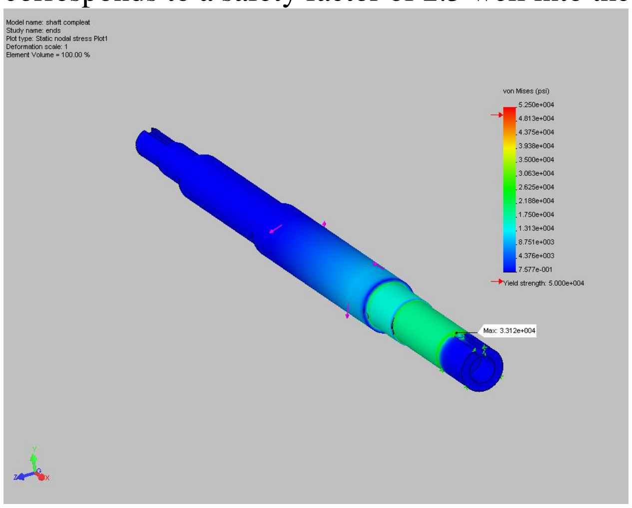


Figure 3: Max Von Mises Stress of 3.312e4 psi.

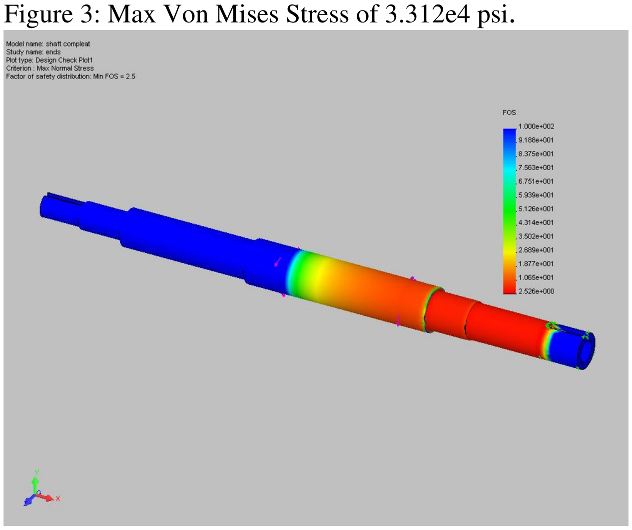


Figure 4: Max Factor of Safety 2.5.

Appendix C.2 – Motor Shaft Analysis

Hand Calculations

T = 250 ft-lb of torque from motors
Bending = 500 lbs from pretension on belt. Located 0.625 inches from the end of the shaft this is the center of the sprocket.
Sy = 100,000 psi 1144 stress proof steel

D1 = 0.75 in thin shaft
D2 = 1.125in large shaft
Shear = $\tau = T*r/J$

Polar Moment of Inertia = $\frac{J}{r} = \frac{\pi * d^3}{16} =$

For D1 J/r = 0.0818
For D2 J/r = 0.13978

Bending = $\sigma_x = \frac{Mc}{I}$

Moment of inertia = $\frac{I}{c} = \frac{\pi * d^3}{32}$

D1 I/c = 0.041417
D2 I/c = 0.2796

Principle Stress = $\sigma_1, \sigma_2 = \sigma_x/2 \pm [(\sigma_x/2)^2 + (\tau)^2]^{0.5}$

Von Mises Stress = $\sigma' = [\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2]^{0.5}$

Safety factor = $n = \frac{Sy}{4\sigma'}$

For 0.75 inch shaft n = 0.68
For 1.125 inch shaft n = 2.28

This confirms our FEA from Cosmos.

Appendix C.3 – Bushing Analysis

Cosmos Analysis was done on the bushings to determine the maximum stresses that they would be taking based on the maximum possible loads. Initially 6061 T-6 aluminum was chosen for the bushing construction. This has a yield strength of 40,000 psi. based on cosmos this material had a safety factor of 1.1. In an effort to increase the safety factor, 7075 T-6 was selected and the safety factor was increased to 2.5.

The bushing was fixed at the bolt holes connecting to the flywheel to simulated power transmission from the motors to the transmission this is seen in Figure 1. A 250 ft-lb was applied to the bolt holes connecting to the sprocket simulating the power from the second motor. Another 250 ft-lb was applied to the keyway to simulate the bottom motors power.

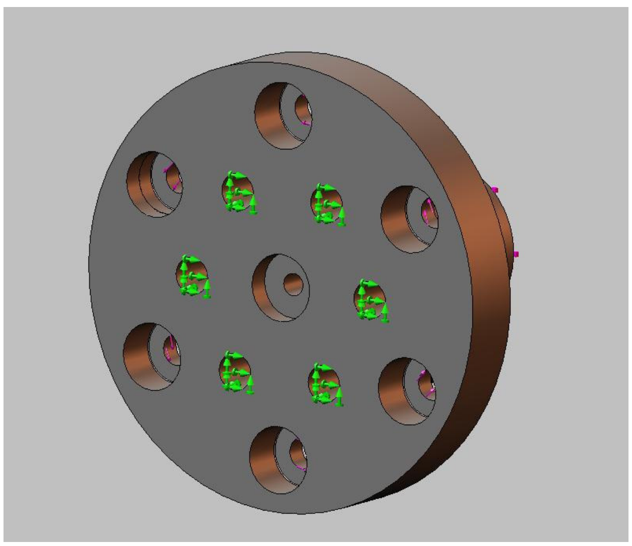


Figure 1: Loading and Constraints

Results

The calculated max Von Mises Stress was found to be 3.62e4 psi seen in Figure 2. The max stress was in located in the keyway. In figure 3 and 4 a factor of safety of 2 was found for 7075 T6 Al and 1.1 for 6061T6 Al. 7075 T6 was selected for the bushing material.

Appendix C.3 – Bushing Analysis

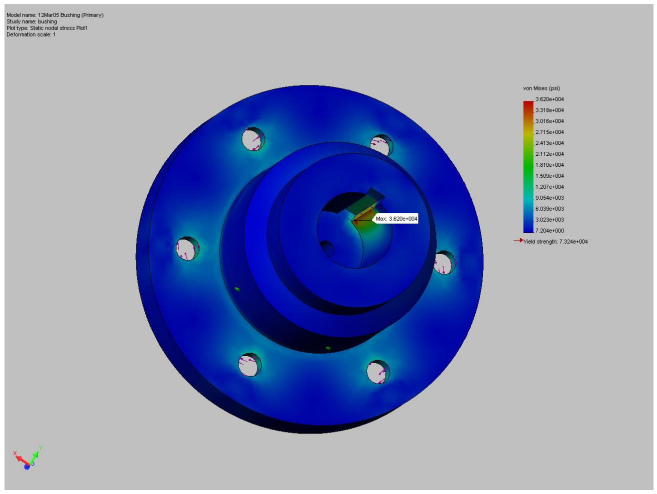


Figure 2: Max Von Mises Stress of 3.62e4 psi

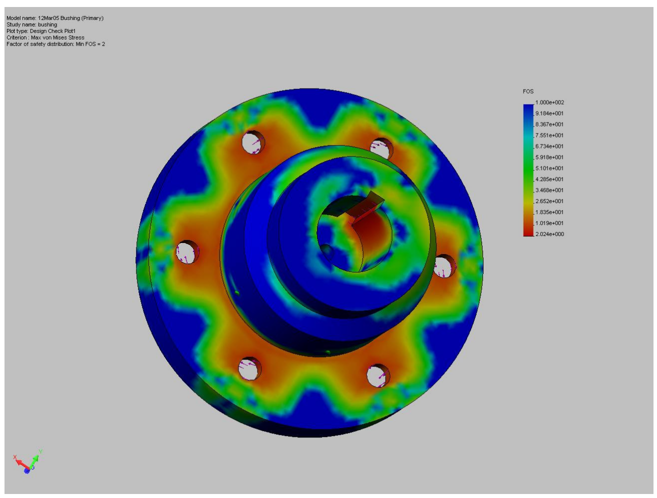


Figure 3: Al 7075 T6 Factor of Safety of 2.

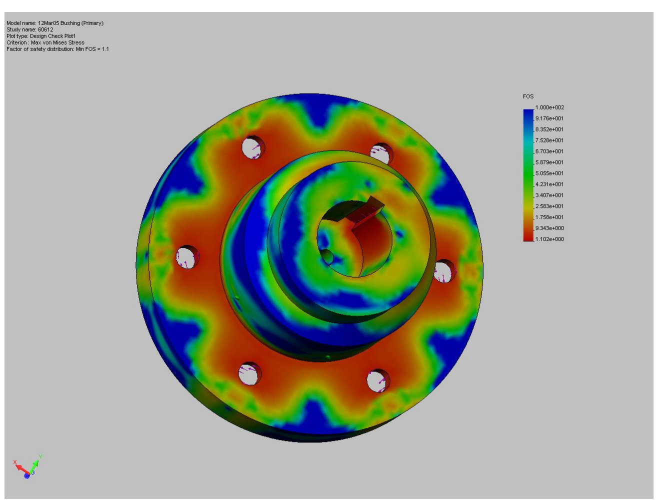


Figure 4: Al 6061 T6 Factor of Safety of 1.1.

Appendix C.4 – Adapter Plate Analysis

Stress analysis of the Adapter Plate System in Torsion

Authors: Alex Fabbiano and Nick Smith

Company: BJ CANE Design Team

Date: 3/17/05

Appendix C.4 – Adapter Plate Analysis

1. Introduction

Summarize the FEM analysis on the belt box with the levels of torsion which will be experienced when each motor is reviving max power.

2. File Information

Model name:	Solid Box
Model location:	C:\Documents and Settings\fabbs\Desktop\16mar05 Assy\FEA Parts\Solid Box.SLDPRT
Results location:	c:\temp
Study name:	new (-Default-)

3. Materials

No.	Part Name	Material	Mass	Volume
1	Solid Box	6061-T6*	11.0999 kg	0.00411107 m^3

Appendix C.4 – Adapter Plate Analysis

4. Load & Restraint Information

Restraint		
Restraint-1 <Solid Box>	on **6 Face(s)** fixed.	
Description:	6 transmission bolts (assuming transmission is rigid)	
Load		
Force-1 <Solid Box>	on **4 Face(s)** apply torque **3000 lb** with respect to selected reference **Face< 1 >** using uniform distribution	Sequential Loading
Description:	Primary Motor Torsion	
Force-2 <Solid Box>	on **4 Face(s)** apply torque **3000 lb** with respect to selected reference **Face< 1 >** using uniform distribution	Sequential Loading
Description:	Secondary Moror Torsion	

Appendix C.4 – Adapter Plate Analysis

5. Study Property

Mesh Information	
Mesh Type:	Solid mesh
Mesher Used:	Standard
Automatic Transition:	On
Smooth Surface:	On
Jacobian Check:	4 Points
Element Size:	0.48099 in
Tolerance:	0.02405 in
Quality:	High
Number of elements:	38263
Number of nodes:	65882

Solver Information	
Quality:	High
Solver Type:	FFE
Option:	Include Thermal Effects
Thermal Option:	Input Temperature
Thermal Option:	Reference Temperature at zero strain: 77 Fahrenheit

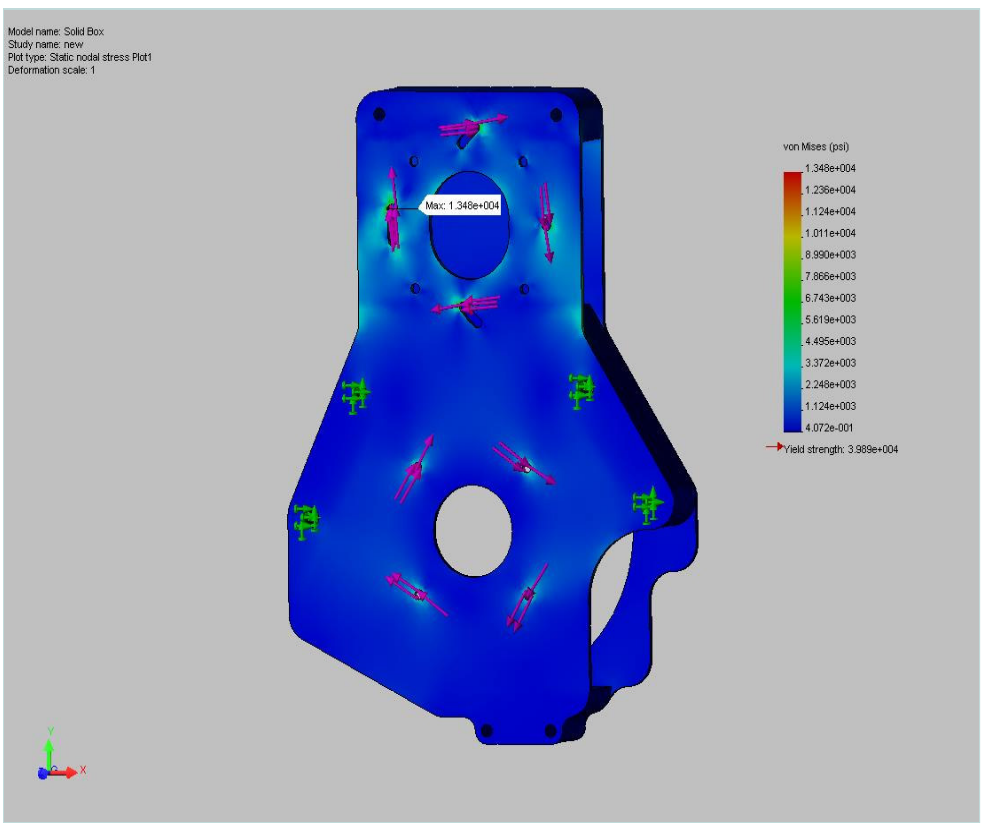
Appendix C.4 – Adapter Plate Analysis

6. Stress Results

Name	Type	Min	Location	Max	Location
Plot1	VON: von Mises stress	0.407201 psi Node: 2561	(3.74451 in, -7.4489 in, 0.25 in)	13484.8 psi Node: 800	(-3.48822 in, 11.9104 in, 0.25 in)

Solid Box-new-Stress-Plot1

JPEG

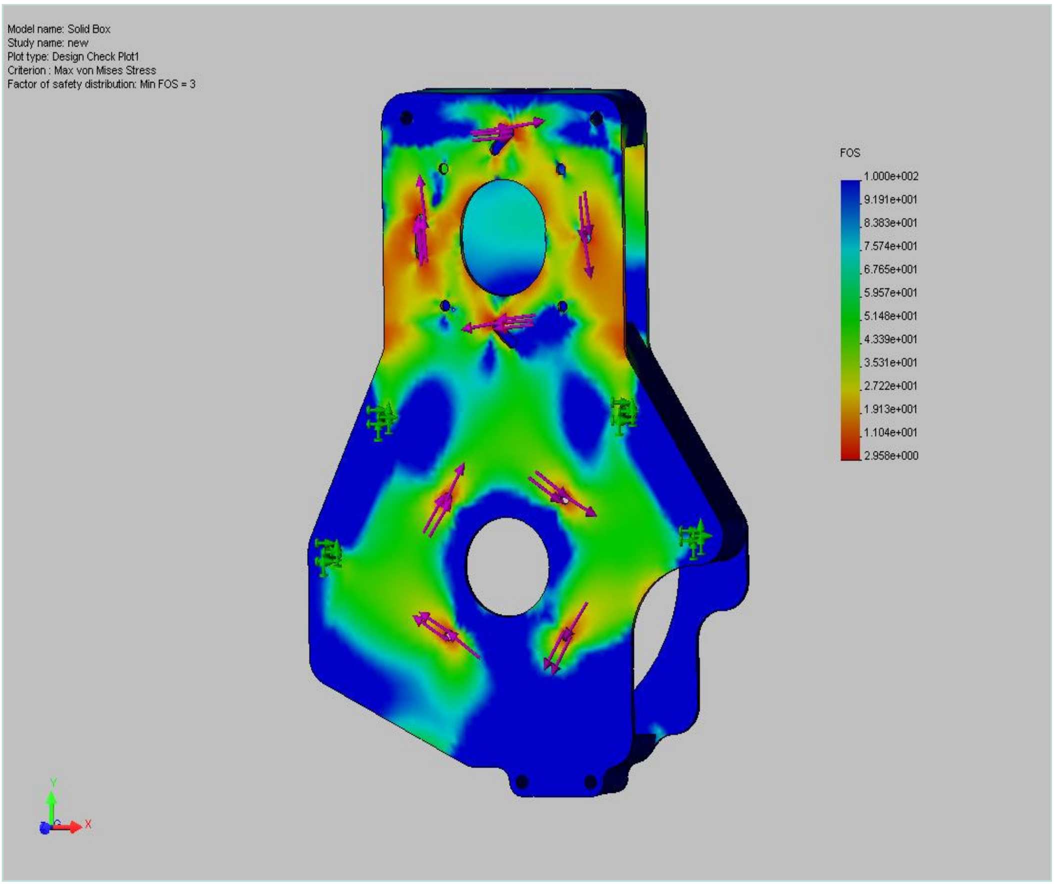


Appendix C.4 – Adapter Plate Analysis

7. Design Check Results

Solid Box-new-Design Check-Plot1

JPEG



Appendix C.4 – Adapter Plate Analysis

8. Appendix

Material name:	6061-T6*
Description:	Aluminum
Material Source:	Library files
Material Library Name:	cosmos materials
Material Model Type:	Linear Elastic Isotropic

Property Name	Value	Units	Value Type
Elastic modulus	6.9e+010	N/m^2	Constant
Poisson's ratio	0.33	NA	Constant
Shear modulus	2.6e+010	N/m^2	Constant
Mass density	2700	kg/m^3	Constant
Tensile strength	3.1e+008	N/m^2	Constant
Yield strength	2.75e+008	N/m^2	Constant
Thermal expansion coefficient	2.4e-005	/Kelvin	Constant
Thermal conductivity	166.9	W/(m.K)	Constant
Specific heat	896	J/(kg.K)	Constant
Hardening factor (0.0-1.0; 0.0=isotropic; 1.0=kinematic)	0.85	NA	Constant

The assembled belt box was found through a Cosmos 2005 FEA simulation to have a minimum factor of safety of 3. This is sufficient considering the belt box will never experience this level of torsion due to the added support of the rear motor plate.

Appendix D.1 – Clutch and Flywheel Options

Clutch Selection Information

Clutch City Online (866 762-5141)
    $115 for a stock flywheel / clutch kit
        (for the ’88 Fiero Getrag 5 Speed 2.5L) (not for drag use)
Clutch Net (Igor - 626 448-7432)
    $485 for a custom built Double Sprung Pressure plate and a three pad sintered iron spring hub clutch disc. Over “Stage 5” performance.
        (for the ’88 Fiero Getrag 5 Speed 2.5L – 215mm disc diameter)
Spec Clutch (205 491-8581)
    $355 for a “Stage 4” SC784 Clutch disc and pressure plate.
        (for the ’88 Fiero Getrag 5 Speed 2.5L)

Interviews with local drag strip enthusiasts as well as the altered function of this clutch led to the decision to order the most aggressive clutch system. Shifting is not required in drive initiation, and motor operation is not dependent on a minimum flywheel rpm. A non streetable clutch was deemed acceptable, and so, the sintered iron three puck disc in conjunction with a double sprung pressure plate was ordered. A sprung hub clutch disc was ordered to ease the initial impulse of the electric motors on the transmission gearing. Sintered iron friction pucks are primarily used where heat is an issue, as in driving situations involving slippage induced through high torque. Such situations will not be experienced with the electric motor system due to the ability of the electric motor system to quickly match any flywheel rpm. Sintered iron clutch disc pucks posses a marginally higher friction coefficient for the same cost as an organic clutch disc, and were thusly selected.

Clutch Kit Order Information

Request for Purchase Submitted March 3rd, 2005.

Requisition Number:	968757
Budget Number:	65-0298
Budget Name:	Student EV Project
Technical Contact Name:	Nicholas SL Smith
Authorizing Signature:	A. Emery
Total Encumbrance:	$467.00

Appendix D.1 – Clutch and Flywheel Options

Flywheel Selection Information

KFM Enterprises (John - 604 657-4850)
    $399 for a custom built Aluminum Flywheel
        (for the ’88 Fiero Getrag 5 Speed 2.5L)
Clutch Net (Igor - 626 448-7432)
    $500 for a custom built Aluminum Flywheel
        (for the ’88 Fiero Getrag 5 Speed 2.5L)
TheFieroShop.com (416 747-5728)
    $550 for a custom built Aluminum Flywheel RT4032
        (for the ’88 Fiero Getrag 5 Speed 2.5L)
Spec Clutch (205 491-8581)
    $399 for a custom built Aluminum Flywheel
        (for the ’88 Fiero Getrag 5 Speed 2.8L) (not compatible)
Coximport.com
    $373 for a Fidanza Aluminum Flywheel FD-198261
        (for the ’88 Fiero Getrag 5 Speed 2.8L) (not compatible)

Each flywheel was guaranteed to work with a high pressure clutch pressure plate and high performance clutch disc. Flywheels all weighed less than 8lb and came balanced and ready to mount any compatible pressure plate.

Until a flywheel can be purchased, the current stock 18lb steel flywheel will be used for construction and testing.

Flywheel Order Information

A reputable supplier has not been found. Each manufacturer listed above was unavailable throughout the design process. The stock flywheel will be used in the design, ensuring maximum compatibility with racing flywheels.

Appendix D.2 – Flywheel Analysis

Stress analysis of 88 stock Fiero 2.5l 5sp (getrag) flywheel in torsion

Author: Nicholas SL Smith

Company: BJCANE

Date: 12Feb2005

Appendix D.2 – Flywheel Analysis

1. Introduction

Summarize the FEM analysis on 88 stock fiero 5sp (getrag) flywheel

---

2. File Information

Model name:	88 stock fiero 5sp (getrag) flywheel
Model location:	C:\Documents and Settings\nick\Desktop\ME 495\measured models\88 stock fiero 5sp (getrag) flywheel.sldprt
Results location:	c:\temp
Study name:	Inital (-Default-)

---

3. Materials

No.	Part Name	Material	Mass	Volume
1	88 stock Fiero 5sp (Getrag) Flywheel	Cast Alloy Steel	15.3762 lb	58.3031 in^3

Appendix D.2 – Flywheel Analysis

4. Load & Restraint Information

Restraint	
clutch stationary <88 stock fiero 5sp (getrag) flywheel>	on **7 Face(s)** fixed.
Description:	Restraints on pressure plate mount points and clutch disc friction face.

Load		
1000 ft lb torque on mount <88 stock fiero 5sp (getrag) flywheel>	on **6 Face(s)** apply torque **6000 in lb** with respect to selected reference **Flywheel Center** using uniform distribution	Sequential Loading
Description:	6000in lb equiv. to 500ftlb of torque – on crankshaft mounting hole interior faces	

---

5. Study Property

Mesh Information	
Mesh Type:	Solid mesh
Mesher Used:	Standard
Automatic Transition:	Off
Smooth Surface:	On
Jacobian Check:	4 Points
Element Size:	0.19392 in
Tolerance:	0.0096963 in
Quality:	High
Number of elements:	64020
Number of nodes:	100029

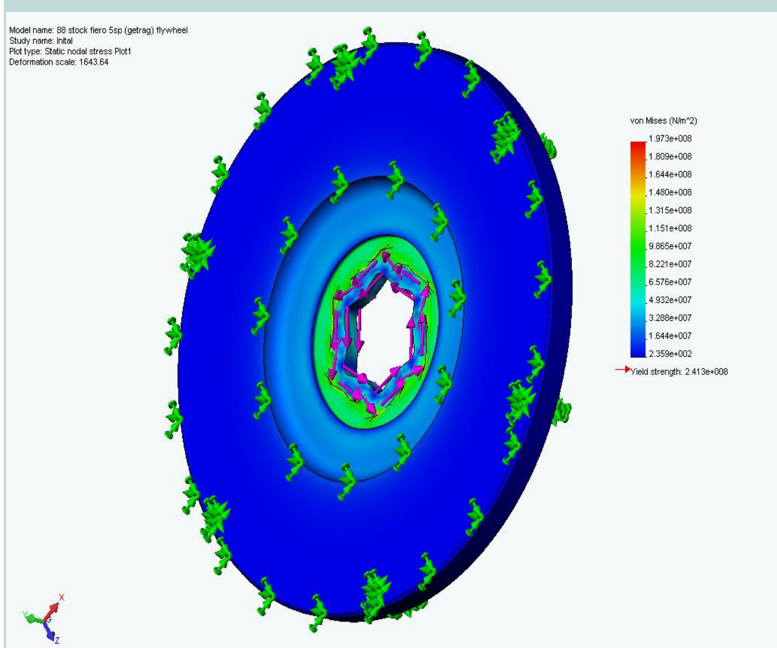
Solver Information	
Quality:	High
Solver Type:	FFEPlus

Appendix D.2 – Flywheel Analysis

6. Stress Results

Name	Type	Min	Location	Max	Location
Plot 1	VON: von Mises stress	235.869 N/m^2 Node: 99327	(-2.59756 in, 0.8 in, -4.66108 in)	1.97292e+008 N/m^2 Node: 97934	(-1.19625 in, 0.261375 in, 0.510204 in)

88 stock fiero 5sp (getrag) flywheel-Inital-Stress-Plot1
JPEG – Plot 1



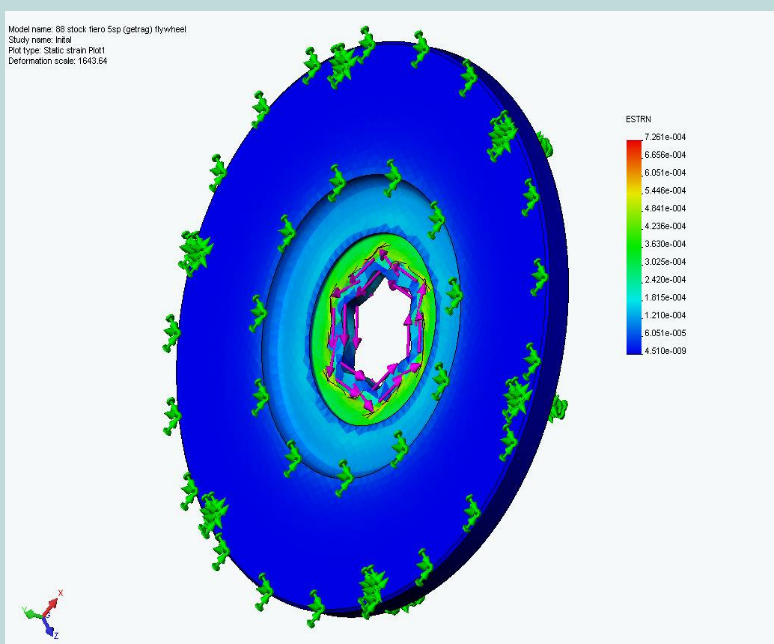
Appendix D.2 – Flywheel Analysis

7. Strain Results

Name	Type	Min	Location	Max	Location
Plot 2	ESTRN: Equivalent strain	4.50952e-009 Element: 44015	(-5.51389 in, 0.221495 in, -0.00502636 in)	0.000726084 Element: 31952	(0.106301 in, 0.0501516 in, 1.34686 in)

88 stock fiero 5sp (getrag) flywheel-Inital-Strain-Plot1

JPEG - Plot 2



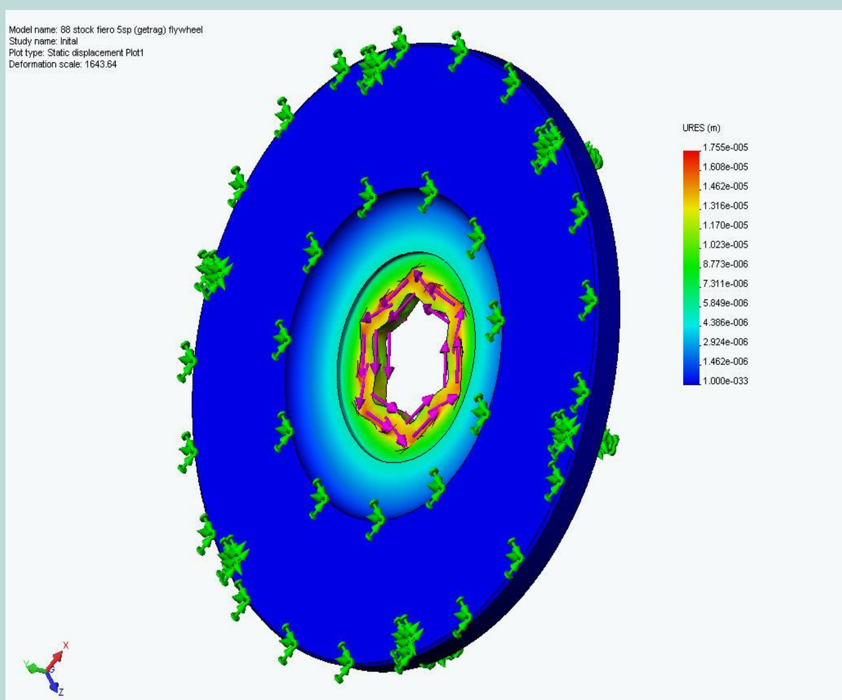
Appendix D.2 – Flywheel Analysis

8. Displacement Results

Name	Type	Min	Location	Max	Location
Plot 3	URES: Resultant displacement	0 m Node: 1	(5.3754 in, 0.8 in, 9.20633e-014 in)	1.75455e-005 m Node: 235	(0.856409 in, 1.02907e-009 in, 0.43345 in)

88 stock fiero 5sp (getrag) flywheel-Inital-Displacement-Plot1

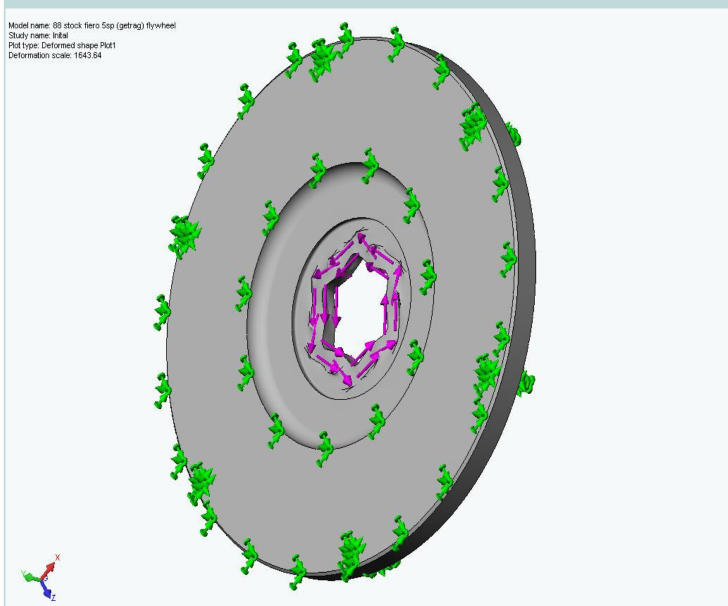
JPEG – Plot 3



Appendix D.2 – Flywheel Analysis

9. Deformation Results

Plot No.	Scale Factor
4	1643.6

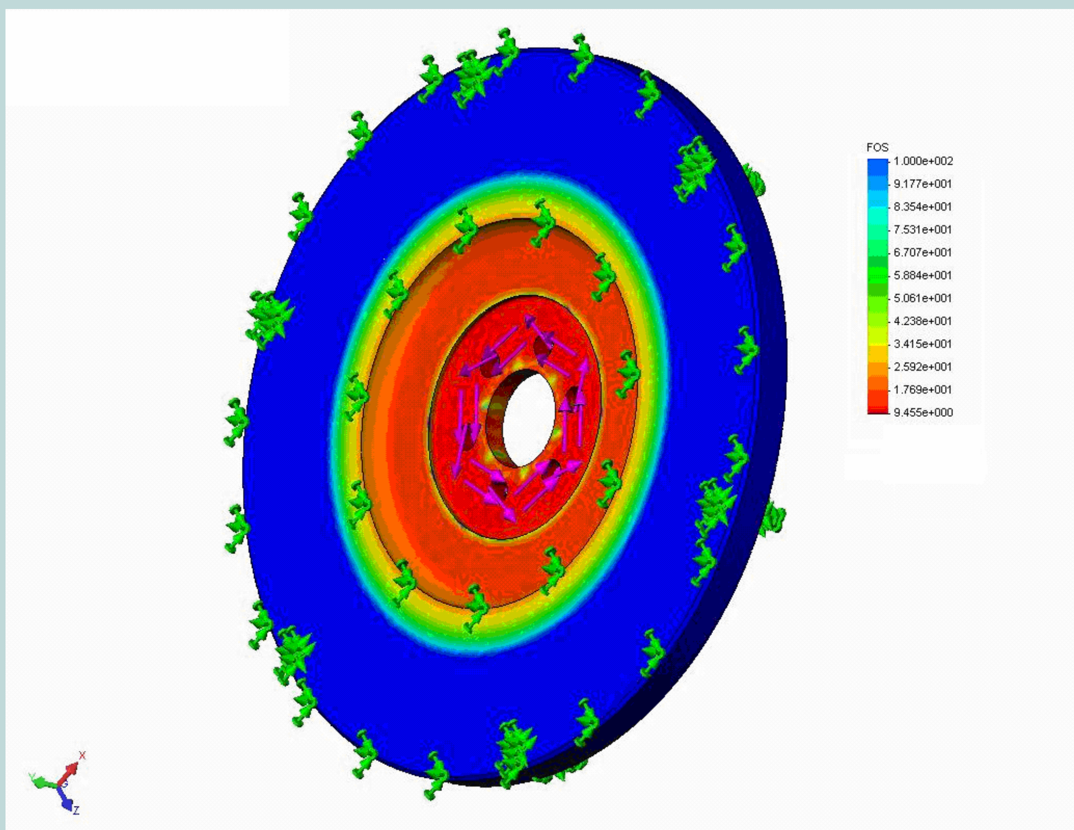
88 stock fiero 5sp (getrag) flywheel-Inital-Deformation-Plot1
JPEG – Plot 4


Appendix D.2 – Flywheel Analysis

10. Design Check Results

88 stock fiero 5sp (getrag) flywheel-Inital-Design Check-Plot1

JPEG – Plot 5



Appendix D.2 – Flywheel Analysis

11. Appendix

Material name:	Cast Alloy Steel
Description:	1988 Pontiac 2.5l 5sp Flywheel
Material Source:	GM – Pontiac Division (1988)
Material Library Name:	solidworks materials
Material Model Type:	Linear Elastic Isotropic

Property Name	Value	Units	Value Type
Elastic modulus	2.7562e+007	psi	Constant
Poisson's ratio	0.26	NA	Constant
Shear modulus	1.1315e+007	psi	Constant
Mass density	0.26373	lb/in^3	Constant
Tensile strength	65000	psi	Constant
Compressive strength	0	psi	Constant
Yield strength	35000	psi	Constant
Thermal expansion coefficient	8.3333e-006	/Fahrenheit	Constant
Thermal conductivity	0.00050839	BTU/(in.s.F)	Constant
Specific heat	0.10511	Btu/(lb.F)	Constant

Appendix D.2 – Flywheel Analysis

Solid Model Computations – Solidworks 2004

*Acutal Weight: 16.5 lb

Mass properties of 88 stock fiero 5sp (getrag) flywheel (Part Configuration - Default)

Output coordinate System: -- default --

Density = 0.26 pounds per cubic inch

Mass = 16.35 pounds

Volume = 61.98 cubic inches

Surface area = 264.04 square inches

Center of mass: (inches)
X = 0.00
Y = 0.42
Z = 0.00

Principal axes of inertia and principal moments of inertia: (pounds * square inches)
Taken at the center of mass.

Ix = (0.00, 0.00, 1.00)	Px = 151.13
Iy = (1.00, 0.00, 0.00)	Py = 151.13
Iz = (0.00, 1.00, 0.00)	Pz = 300.92

Moments of inertia: (pounds * square inches)
Taken at the center of mass and aligned with the output coordinate system.

Lxx = 151.13	Lxy = 0.00	Lxz = 0.00
Lyx = 0.00	Lyy = 300.92	Lyz = 0.00
Lzx = 0.00	Lzy = 0.00	Lzz = 151.13

Moments of inertia: (pounds * square inches)
Taken at the output coordinate system.

Ixx = 153.97	Ixy = 0.00	Ixz = 0.00
Iyx = 0.00	Iyy = 300.92	Iyz = 0.00
Izx = 0.00	Izy = 0.00	Izz = 153.97

Appendix D.2 – Flywheel Analysis

Stress analysis of Aluminum Flywheel Design in torsion -

Author: Nicholas SL Smith

Company: BJCANE

Date: 12Feb2005

Appendix D.2 – Flywheel Analysis

1. Introduction

Summarize the FEM Torsional analysis on Colin and Nick's aluminum flywheel concept

3. Materials

No.	Part Name	Material	Mass	Volume
1	Colin flywheel_concept	[SW]6061 Alloy	3.00962 kg	0.00111467 m^3

---

4. Load & Restraint Information

Restraint		
Restraint-1 <Nick Redesign 0>	on **4 Face(s)** fixed.	
Description:	Keyed to match motor – friction and keyway restraint	

Load		
Force-1 <Nick Redesign 0>	on **7 Face(s)** apply torque **6000 lb** with respect to selected reference **Center** using uniform distribution	Sequential Loading
Description:	500 ft lb on clutch disc friction surface and pressure plate mount locations	

Appendix D.2 – Flywheel Analysis

5. Study Property

Mesh Information	
Mesh Type:	Solid mesh
Mesher Used:	Standard
Automatic Transition:	Off
Smooth Surface:	On
Jacobian Check:	4 Points
Element Size:	0.19972 in
Tolerance:	0.0099861 in
Quality:	High
Number of elements:	71609
Number of nodes:	112247

Solver Information	
Quality:	High
Solver Type:	FFEPlus
Option:	Include Thermal Effects
Thermal Option:	Input Temperature
Thermal Option:	Reference Temperature at zero strain: 298 Kelvin

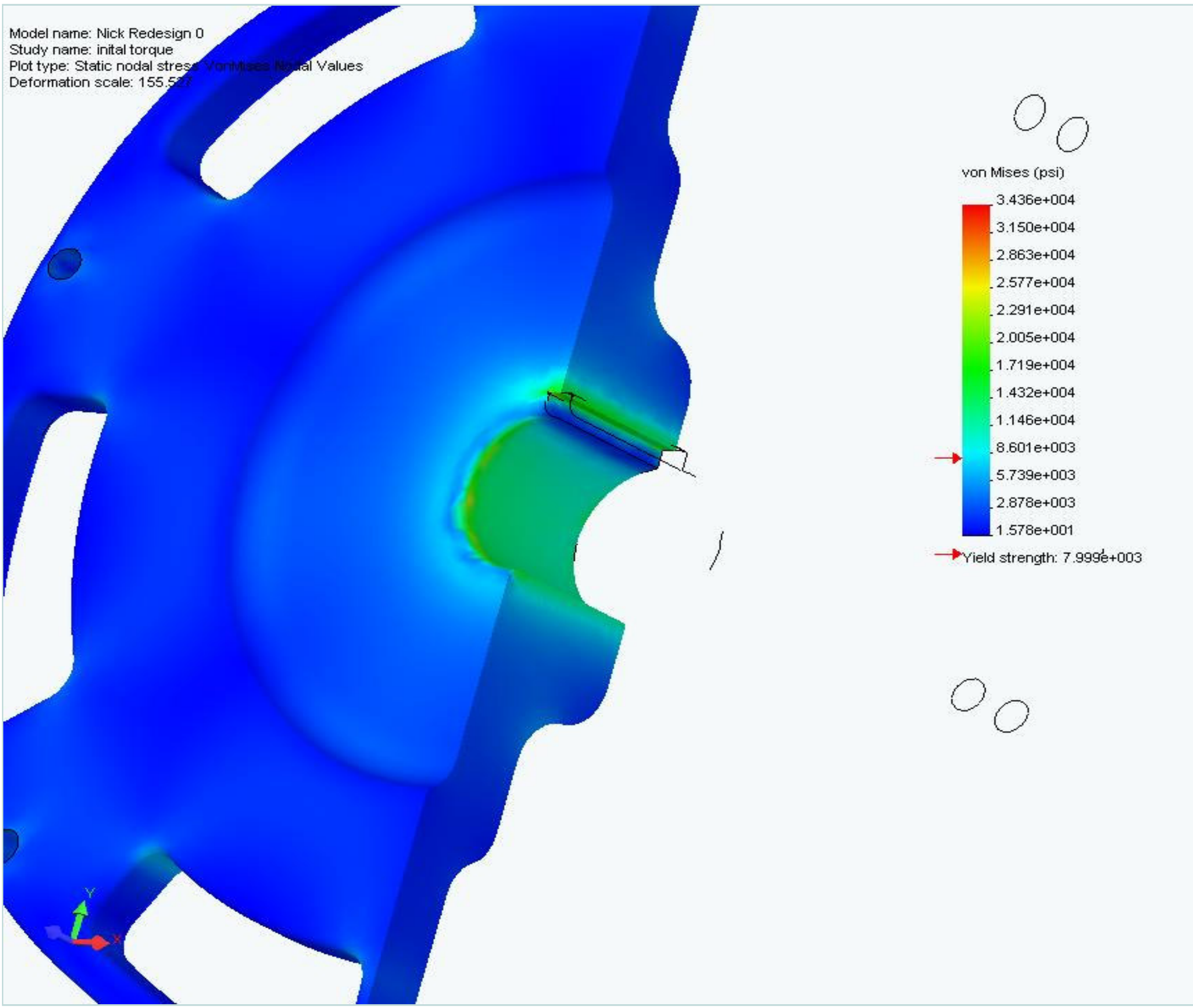
Appendix D.2 – Flywheel Analysis

6. Stress Results

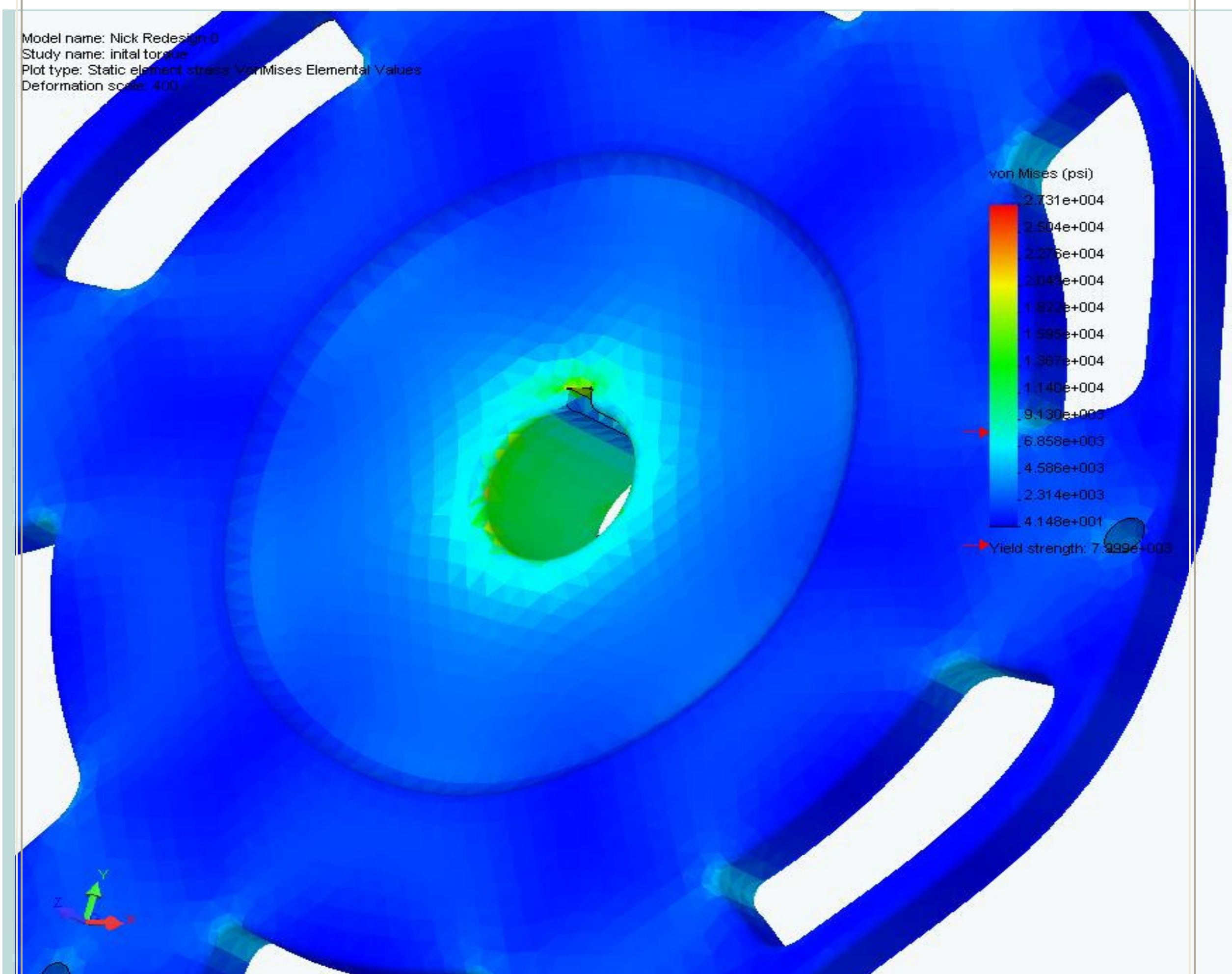
Name	Type	Min	Location	Max	Location
VonMises Nodal Values	VON: von Mises stress	15.7847 psi Node: 2025	(-5.875 in, 0 in, 0.3 in)	34356.8 psi Node: 60129	(0.117132 in, 0.925826 in, 0.344429 in)
Intensity Elemental Values	VON: von Mises stress	41.4762 psi Element: 14127	(5.84323 in, -0.0318101 in, 0.366757 in)	27307.2 psi Element: 62545	(0.122047 in, 0.924978 in, 0.344658 in)
Error	ERR: Energy norm error	0.0150996 Element: 42807	(5.53309 in, 0.141811 in, 0.520006 in)	189.593 Element: 17647	(0.0138692 in, 0.978481 in, 0.0208662 in)
VonMises Elemental Values	VON: von Mises stress	41.4762 psi Element: 14127	(5.84323 in, -0.0318101 in, 0.366757 in)	27307.2 psi Element: 62545	(0.122047 in, 0.924978 in, 0.344658 in)

Appendix D.2 – Flywheel Analysis

Nick Redesign 0-inital torque-Stress-VonMises Nodal Values
JPEG



Appendix D.2 – Flywheel Analysis

Nick Redesign 0-inital torque-Stress-VonMises Elemental Values
JPEG


Appendix D.2 – Flywheel Analysis

7. Strain Results

Name	Type	Min	Location	Max	Location
Equiv Strain	ESTRN: Equivalent strain	3.67476e-006 Element: 14127	(5.84323 in, -0.0318101 in, 0.366757 in)	0.00241939 Element: 62545	(0.122047 in, 0.924978 in, 0.344658 in)
Strain Energy	SEDENS: Strain energy density	0.853308 Element: 14127	(5.84323 in, -0.0318101 in, 0.366757 in)	282671 Element: 62545	(0.122047 in, 0.924978 in, 0.344658 in)

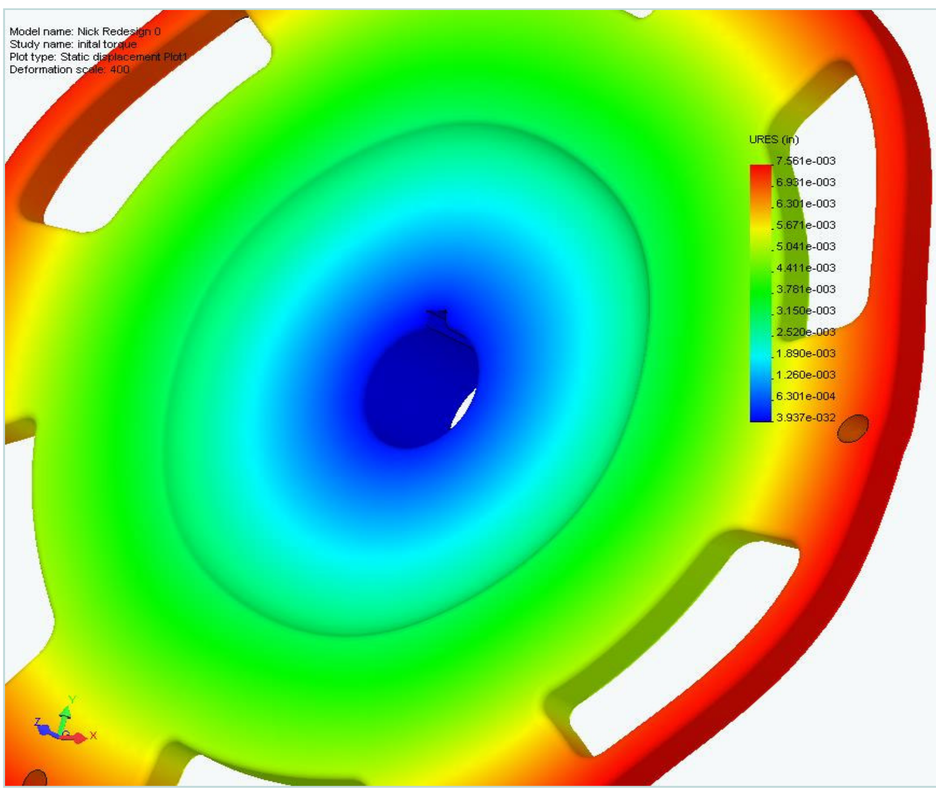
Appendix D.2 – Flywheel Analysis

8. Displacement Results

Name	Type	Min	Location	Max	Location
Plot1	URES: Resultant displacement	0 in Node: 7	(0.76 in, 8.53702e-017 in, -1.25 in)	0.00756119 in Node: 3573	(-0.298835 in, -5.86739 in, 0.1 in)

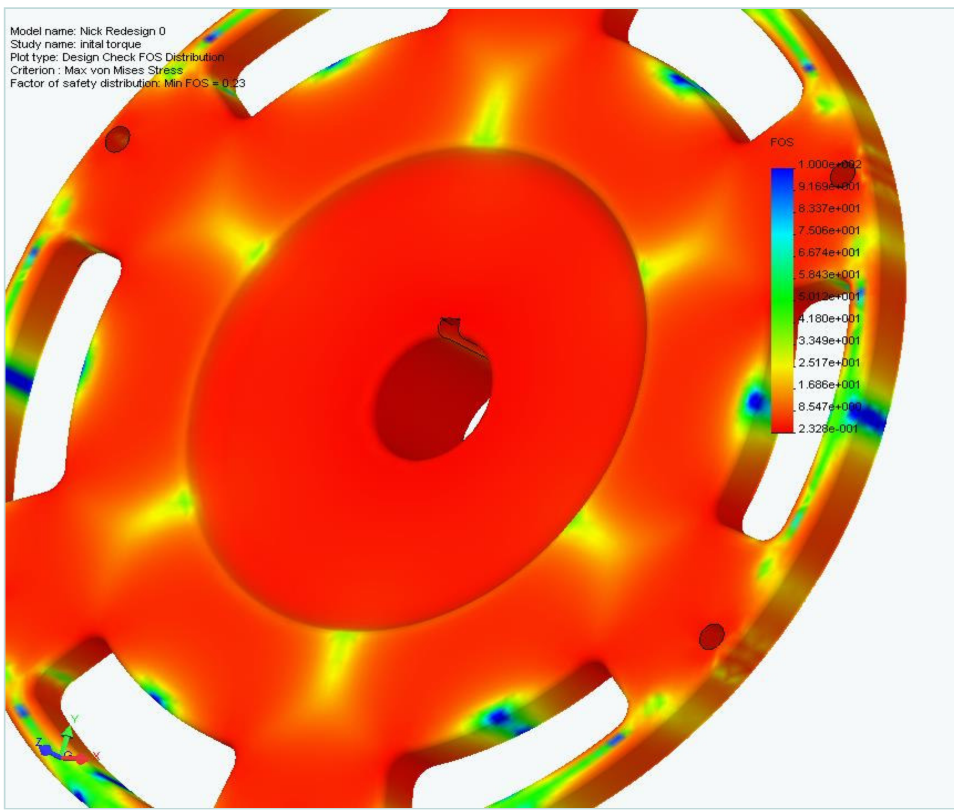
Nick Redesign 0-inital torque-Displacement-Plot1

JPEG



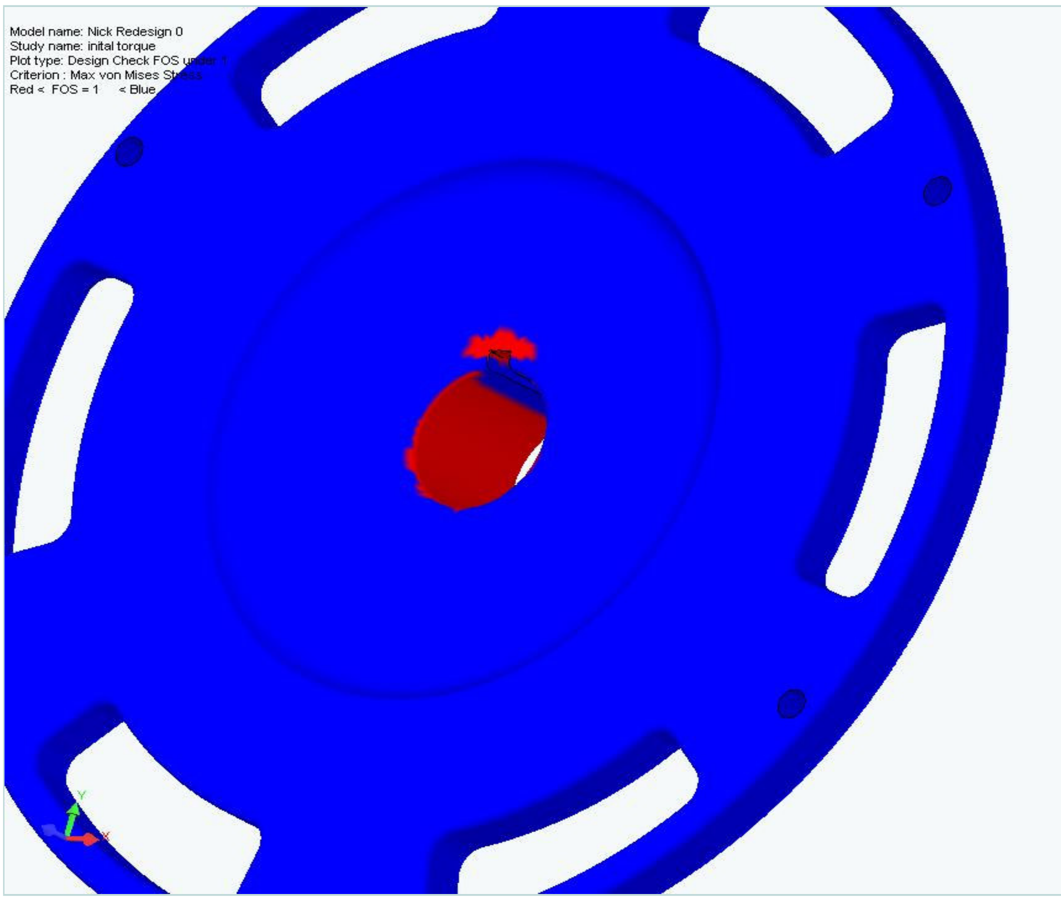
Appendix D.2 – Flywheel Analysis

10. Design Check Results

Nick Redesign 0-inital torque-Design Check-FOS Distribution
JPEG


Appendix D.2 – Flywheel Analysis

Nick Redesign 0-inital torque-Design Check-FOS under 1
JPEG



Appendix D.2 – Flywheel Analysis

11. Appendix

Material name: [SW]6061 Alloy

Description: Custom Flywheel

Material Source: SolidWorks material

Material Library Name: solidworks materials

Material Model Type: Linear Elastic Isotropic

Property Name	Value	Units	Value Type
Elastic modulus	6.9e+010	N/m^2	Constant
Poisson's ratio	0.33	NA	Constant
Shear modulus	2.6e+010	N/m^2	Constant
Mass density	2700	kg/m^3	Constant
Tensile strength	1.2408e+008	N/m^2	Constant
Yield strength	5.5149e+007	N/m^2	Constant
Thermal expansion coefficient	2.4e-005	/Kelvin	Constant
Thermal conductivity	170	W/(m.K)	Constant
Specific heat	1300	J/(kg.K)	Constant

Appendix D.2 – Flywheel Analysis

Mass properties of Nick Redesign 0 (Part Configuration - Default)

Output coordinate System: -- default --

Density = 0.10 pounds per cubic inch

Mass = 6.64 pounds

Volume = 68.02 cubic inches

Surface area = 240.23 square inches

Center of mass: (inches)
X = 0.00
Y = -0.00
Z = 0.06

Principal axes of inertia and principal moments of inertia: (pounds * square inches)
Taken at the center of mass.

Ix = (1.00, 0.00, 0.00)	Px = 45.10
Iy = (0.00, 1.00, 0.00)	Py = 45.10
Iz = (0.00, -0.00, 1.00)	Pz = 88.17

Moments of inertia: (pounds * square inches)
Taken at the center of mass and aligned with the output coordinate system.

Lxx = 45.10	Lxy = 0.00	Lxz = 0.00
Lyx = 0.00	Lyy = 45.10	Lyz = 0.00
Lzx = 0.00	Lzy = 0.00	Lzz = 88.17

Moments of inertia: (pounds * square inches)
Taken at the output coordinate system.

Ixx = 45.12	Ixy = 0.00	Ixz = 0.00
Iyx = 0.00	Iyy = 45.13	Iyz = 0.00
Izx = 0.00	Izy = 0.00	Izz = 88.17

** The Aluminum concept has 29.3 percent of the moment of inertia about the rotational axis and 40.6 percent the weight of the stock flywheel; however, it has a factor of safety below 1. This concept is not viable. A commercially available aluminum flywheel with a guaranteed factor of safety over unity is desirable, and would similarly have a much lower moment of inertia than the stock steel flywheel.

Appendix E.1 – Parts List / Budget Assessment

Part	Part Number	Item Call Out	Description	Quantity	Purchase Per Unit	Material Cost	Total Cost
Goodyear Eagle Pd Blue Belt	B-994	Belt	Coupling System Pd Belt	1	Donated	--	$0.00
Goodyear Eagle Pd Blue Sprocket	B-30S-MPB	Sprocket	Coupling System Sprockets	2	Donated	--	$0.00
Custom Lower Sprocket Bushing		Bush 1	Custom made to fit to motor shaft	1		$115.00	$115.00
Custom Upper Sprocket Bushing		Bush 2	Custom made to fit to motor shaft.	1		$115.00	$115.00
Warp 8" DC Motor	Warp 8	Motor	Dual Shaft, Advanced Timing DC motor	2	$1,300.00		$2,600.00
Custom Aluminum Adapter Plates		P1	6061 Aluminum, 0.25" thickness	1	--	$97.80	$97.80
Custom Aluminum Adapter Plates		P2	6061 Aluminum, 0.25" thickness	1	--	$97.80	$97.80
Custom Aluminum Adapter Plates		P3	6061 Aluminum, 0.25" thickness	1	--	$97.80	$97.80
Custom Plate Support Structures		S1, S2, S3, S4	6061 Aluminum block spacers between P1&2	4	--	$91.65	$366.60
McMaster 1/4" x 1/4" Key	98870A405	K1	Transmits torque from motor to Hub	2	$4.67	--	$9.34
Flywheel	1988 2.5L 5sp	Flywheel	Original Fiero Flywheel	1	Donated	--	$0.00
ClutchNet "E-Z Lock Pro" clutch disk	65005	Clutch	8.5 in., 3-button, Srung Hub	1	$467.00	--	$467.00
ClutchNet Pressure Plate		Pressure Plate	Double Diaphragm Pressure Plate	1	With Clutch Disk	--	$0.00
88 Pontiac Fiero Transmission	Getrag	Transmission	5-speed Getrag for 2.5L Fiero engine	1	$100.00	--	$100.00
Custom Alignment Pins	A1	A1	Alignment pins for adapter plates to transmission	2		$2.62	$2.62

Part	Part Number	Item Call Out	Description	Quantity	Cost per Pkg	Pkg. Qty.
McMaster-Carr Nut	90685A110	M12 Nut	Stainless Steel M12 nut	7	$10.43	50
McMaster-Carr Split-ring Lock Washer	91190A570	M12 Washer	Black Steel M12 Split-ring Lock washer	7	$6.67	100
McMaster-Carr Bolt	92316A581	B1	Grade 8, 3/4 inch, 5/16"-18, fully threaded	8	$9.07	50
McMaster-Carr Bolt	92316A624	B2	Grade 8, 1 inch, 3/8"-16, fully threaded	8	$5.46	25
McMaster-Carr Bolt	91310A538	B3	Class 10.9, 30mm, M8, partially threaded	12	$7.18	25
McMaster-Carr Bolt	91310A712	B4	Class 10.9, 30mm, M12, fully threaded	6	$11.82	25
Pontiac Pressure Plate Bolt	O98	B6	Pressure plate to flywheel bolts	6	$0.00	0
McMaster-Carr Bolt	91310A728	B6	Class 10.9, 60mm, M12, partially threaded	2	$6.70	10
McMaster-Carr Bolt	91310A742	B7	Class 10.9, 100mm, M12, partially threaded	5	$4.77	5
McMaster-Carr Bolt	91310A750	B8	Class 10.9, 120mm, M12, partially threaded	3	$6.42	5
McMaster-Carr Bolt	92316A595	B9	Grade 8, 2 1/2 Inch, 5/16-18, partially threaded	4	$8.38	25

Part Total $3,966.34

Fastener Total $76.90

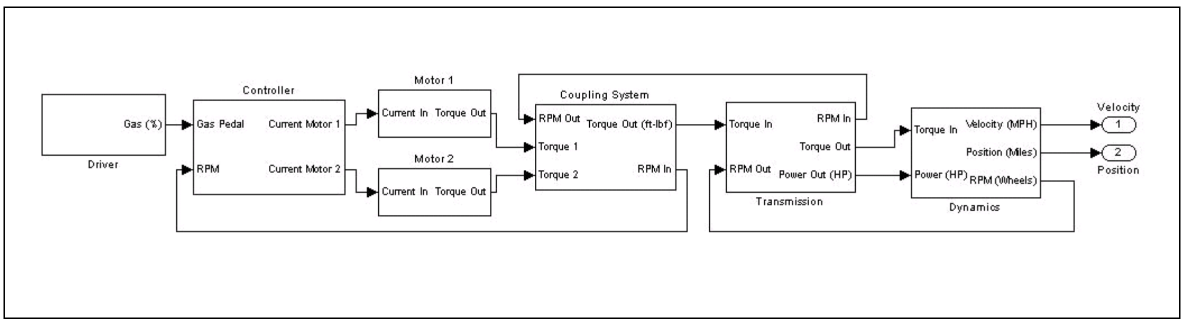
TOTAL SYSTEM COST $4,145.86

Appendix E.2 – BJ CANE Vehicle Dynamics Simulation Description

The model was created in Simulink using block diagrams. The main components of the car were modeled individually and then combined to model the entire car. This allowed each component of the vehicle to be tested and tweaked until it behaved appropriately. This approach also makes it easier to follow the flow of the block diagram signals.

Subcomponents:

Driver
Controller
Motor
Coupling System
Transmission
Car Dynamics



Driver:

Inputs:

None

Outputs:

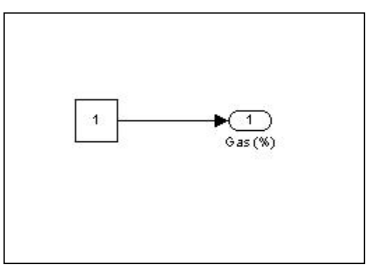
Amount accelerator is suppressed sent to controller (%)

Parameters:

None

Description:

Assumes the pedal is to the metal



Appendix E.2 – BJ CANE Vehicle Dynamics Simulation Description

Controller:

Inputs:

Amount accelerator is suppressed from driver (%)
RPM from coupling system

Outputs:

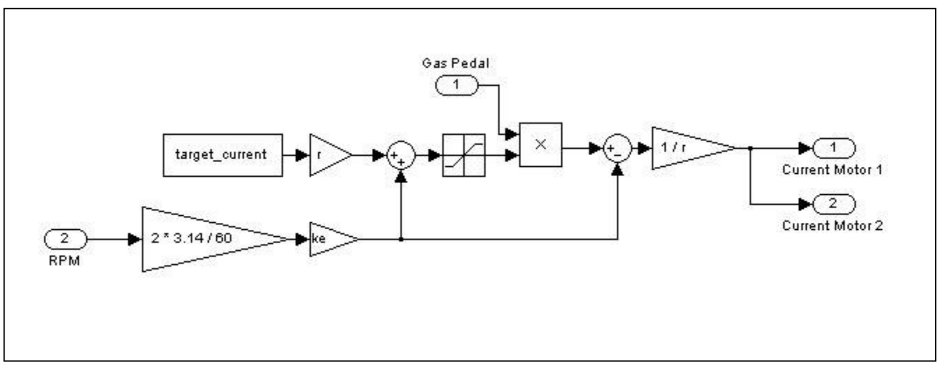
Motor 1 current to motor 1 (amps)
Motor 2 current to motor 2 (amps)

Parameters:

Target Current (Amps)
Resistance (Ohms)
Motor Constant (N-M / Amp)
Maximum Voltage (V)
Minimum Voltage (V)

Description:

To determine the voltage applied across the motors: $V = I R + Ke\ \omega$
This voltage is capped at 196 volts
To determine the current applied through the motors: $I = (V - Ke\ \omega) / R$



Appendix E.2 – BJ CANE Vehicle Dynamics Simulation Description

Motor:

Inputs:

Current from controller (amps)

Outputs:

Motor torque to coupling system (ft-lbf)

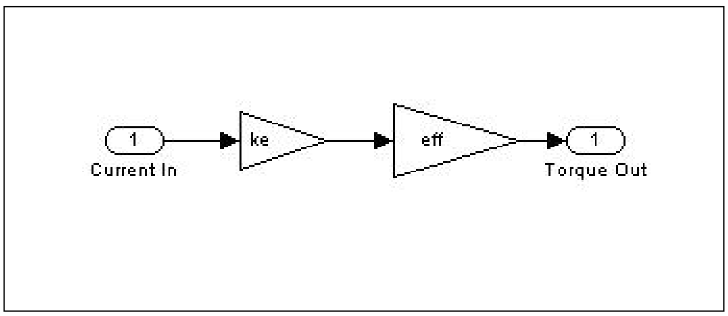
Parameters:

Motor Constant (N-M / Amp)

Efficiency

Description:

The formula $M = k_e I$ is used to determine the torque output



Coupling System:

Inputs:

Torque from motor 1 (ft-lbf)

Torque from motor 2 (ft-lbf)

RPM from transmission

Outputs:

Coupling torque to transmission (ft-lbf)

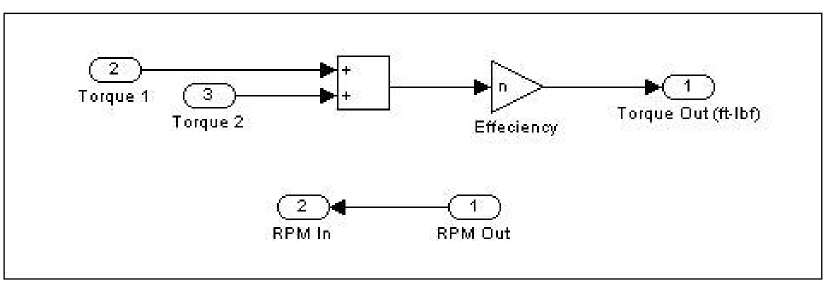
Motor RPM to controller

Parameters:

Efficiency

Description:

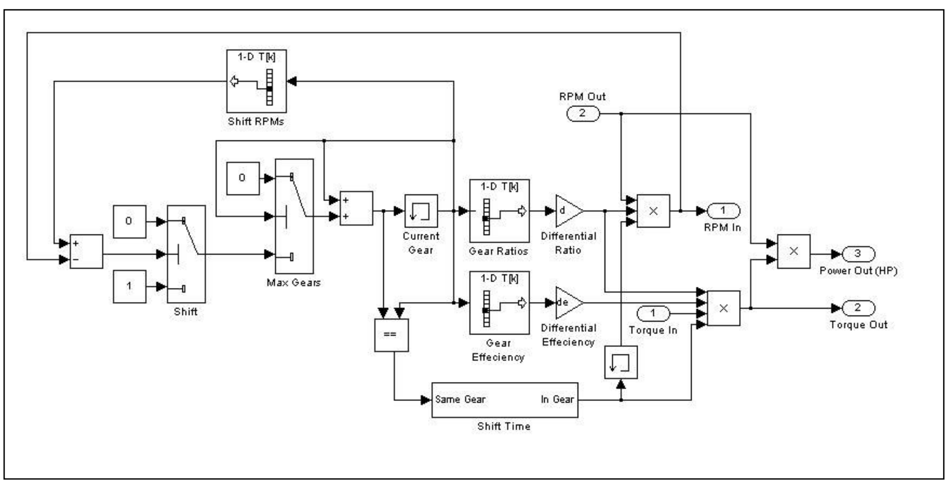
Adds the torques of the two motors and sets the motors' RPM to the RPM of the motor end of the transmission



Appendix E.2 – BJ CANE Vehicle Dynamics Simulation Description

Transmission:

- Inputs:
  - Torque from coupling system (ft-lbf)
  - RPM from the wheels
- Outputs:
  - Transmission torque to car dynamics (ft-lbf)
  - Transmission power to car dynamics (HP)
  - Motor side of transmission RPM to coupling system
- Parameters:
  - Start Gear
  - Gear Shift RPM
  - Gear Ratio
  - Gear Efficiency
  - Differential Ratio
  - Differential Efficiency
  - Shift Time (sec)
- Description:
  - Simulates gear ratios and shifting. When motor RPM reaches the shifting point, the gear ratios change, simulating a shift.



Appendix E.2 – BJ CANE Vehicle Dynamics Simulation Description

Car Dynamics:

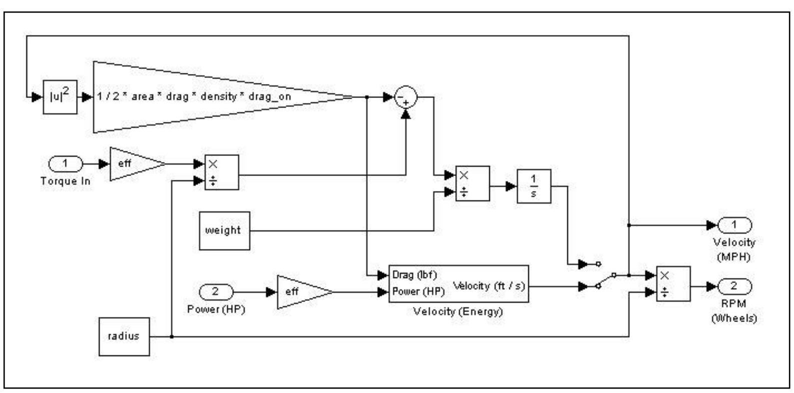
- Inputs:
  - Torque from transmission (ft-lbf)
  - Power from transmission (HP)
- Outputs:
  - Car acceleration (ft/s^2)
  - Car velocity (MPH)
  - Car position (miles)
- Parameters:
  - Car Weight (lbf)
  - Wheel Radius (ft)
  - Drag Coefficient
  - Efficiency
- Description:

The user can choose between two approaches to determine the car's velocity. The force method uses the equation $F = m \cdot a$ to determine the acceleration of the car. This acceleration is then integrated to find the car's velocity. The energy method uses the equation:

$$\int (T \cdot \omega) = \tfrac{1}{2} \cdot m \cdot v^2$$

where T is the torque, ω is the angular speed of the transmission, m is the car's mass and v is the car's velocity. Solving for velocity gives:

$$v = \sqrt{\tfrac{2}{m} \cdot \int (T \cdot \omega)}$$



Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results

Setup:
The simulation was run with four different setups:

- Ideal settings with two 8'' motors
- Realistic settings with two 8'' motors
- Ideal settings with single 9'' motor
- Realistic settings with single 9'' motor

The two 8'' motors had the following assumed properties:

- Motor constants of 0.33 N-m / amp
- Motor resistances of 0.02 ohms
- Target current of 1000 amps through each motor
- Maximum voltage of 196 volts

The single 9'' motor had the following assumed properties:

- Motor constant of 0.35 N-m / amp
- Motor resistance of 0.0178 ohms
- Target current of 1000 amps through the motor
- Maximum voltage of 196 volts

In the ideal setting the following assumptions were made:

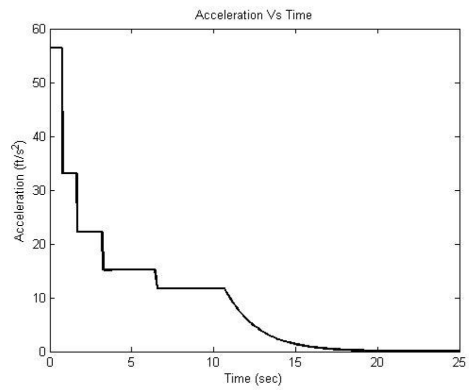
- All unknown efficiencies were set to 100% efficient
- No shift time
- No tire slippage, which allowed the vehicle to start in first gear
- No drag

In the realistic setting the following assumptions were made:

- All unknown efficiencies were set to 90% efficient
- There was a shift time of 0.2 seconds
- The tires were allowed to slip, which forced the vehicle to start in second gear
- The tire coefficient of friction was 1.5
- There was drag

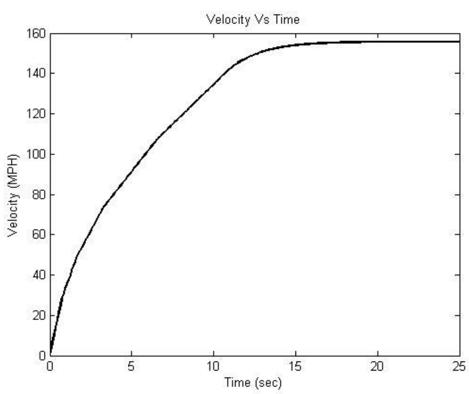
Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results

Results:
Ideal settings with two- 8” motors:
0-60 time: 2.371 seconds



Acceleration Vs Time:

The reason the acceleration has a discontinuous drop off is because it is proportional to the force. The transmission torque equals the couple torque multiplied by the gear ratio. At the instant of drop off, the gear ratio decreases, so the torque to the wheels decreases.



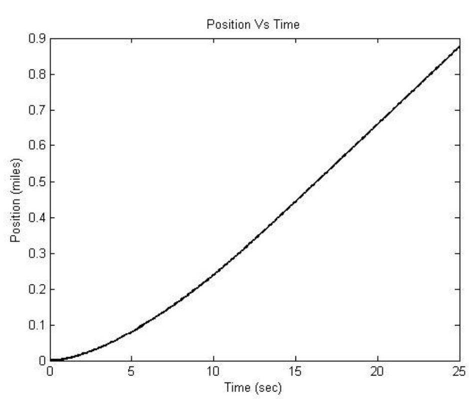
Velocity Vs Time:

Integrating the vehicle acceleration gives this graph. An identical graph can be created using the equation:

$$\int T \cdot \omega = \frac{1}{2} \cdot m \cdot v^2$$

and solving for velocity.

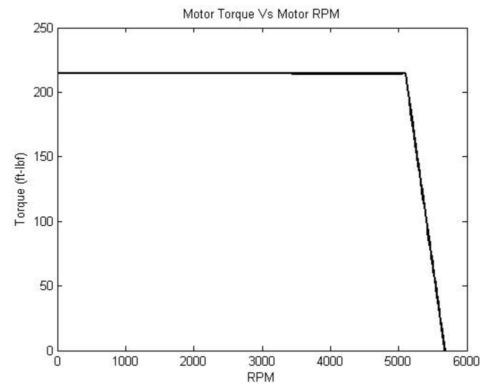
Notice that the velocity is greater than 60 well before 5 seconds.



Position Vs Time:

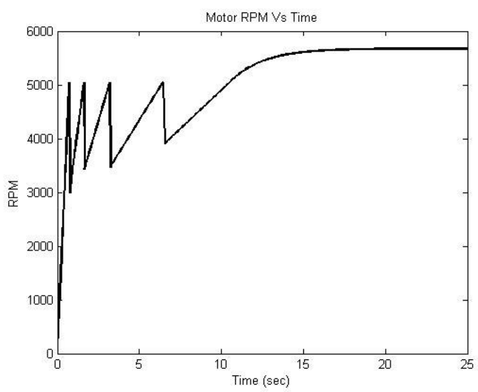
Integrating the vehicle acceleration gives this graph.

Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results



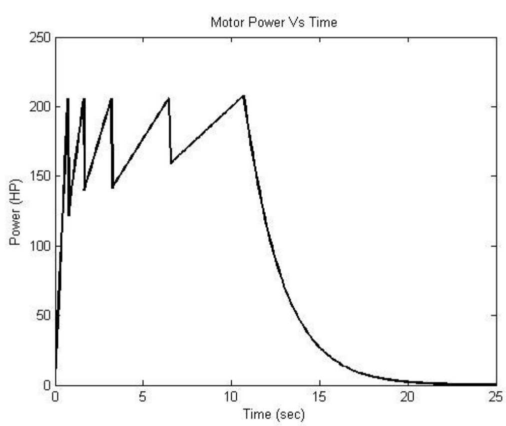
Motor Torque Vs Motor RPM:

This graph shows that electric motors have constant torque up until a certain EMF. The graph also demonstrates the effects of back EMF. When the motor's RPM is higher than around 5050, the current through the motor begins to decrease. Motor torque is directly proportional to the current.



Motor RPM Vs Time:

This graph demonstrates the transmission shift points and the maximum motor RPM. When the motor RPM reaches 5050 and the transmission is in 4th gear or lower, the transmission shifts to the next gear. This decreases the motor RPM instantaneously because there is no shift time.

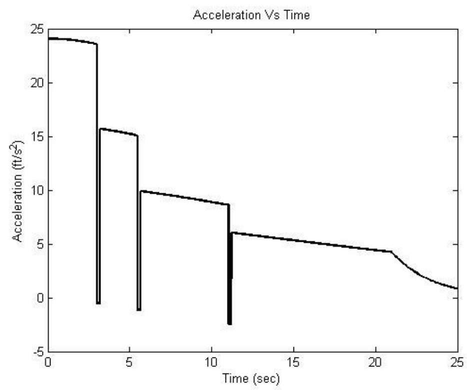


Motor Power Vs Time:

This graph demonstrates the power drop off after shifts and as the back EMF takes affect. The power drops off because power equals T ω, where T is the motor torque and ω is the motor angular velocity. Before the shift, the motor angular velocity equals the wheel angular velocity times the gear ratio. After the shift, the motor angular velocity equals the wheel angular velocity times a new, lower ratio. So at the instant the gear ratio changes the torque remains the same, but the angular velocity decreases, and thus T ω decreases. Note that this is only one motor.

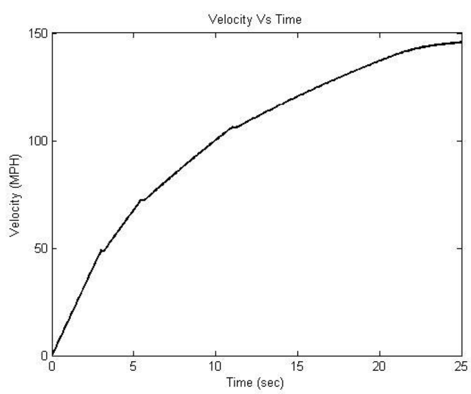
Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results

Realistic settings with two- 8" motors:
0-60 time: 4.262 seconds



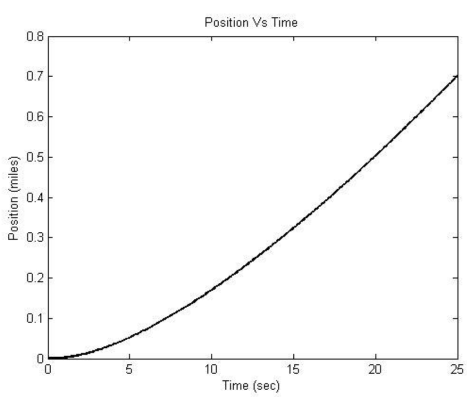
Acceleration Vs Time:

There are two main differences between this graph and the ideal case. The first difference is that while the gear is engaged the acceleration still decreases with time. This is because the drag increases with the square of the velocity. The second difference occurs when the gears are beings shifted. There is no power being transmitted for 0.2 seconds while the gears are being shifted. During this time the acceleration is negative because of drag.



Velocity Vs Time:

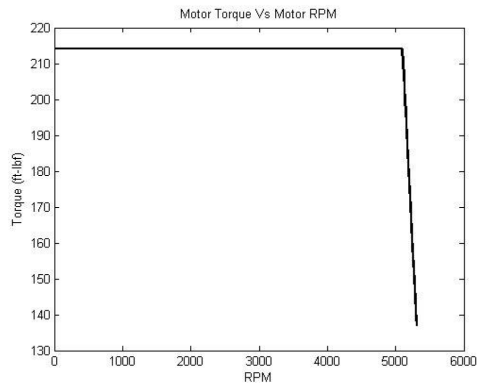
The only major difference between this velocity graph and the last one is the small plateaus during shifting. The car is also accelerating slower and the maximum speed is about 10 MPH less.



Position Vs Time:

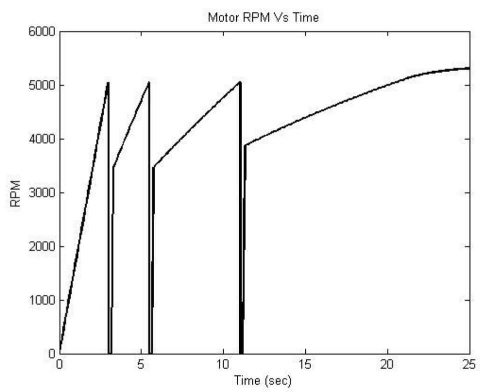
In the realistic version the car travels about 0.2 miles less in 25 seconds.

Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results



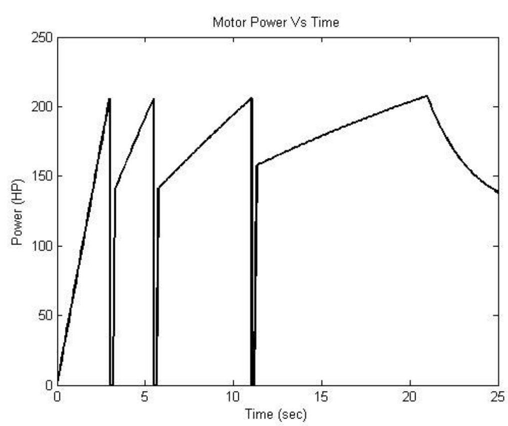
Motor Torque Vs Motor RPM:

In this graph the motor torque only decreases until it exactly counters the drag force. At this point the sum of forces on the car is zero and so its velocity is constant. At this point the motor RPM will not increase any more, and so the back EMF will not reduce the torque any more.



Motor RPM Vs Time:

This graph demonstrates that the transmission shifts at the same motor RPM for each gear. There are only four active gears because in this case the car starts in 2nd gear because if the car starts in 1st gear there is too much torque and the tires slip. While the transmission is out of gear, the simulation assumes there is no motor RPM (although this is incorrect it has no bearing on other calculations).

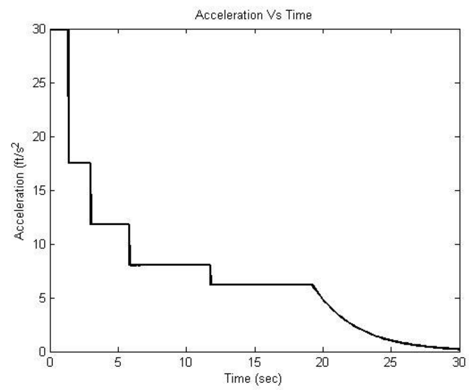


Motor Power Vs Time:

A maximum horse power of approximately 206 can be seen in this graph. In the lower gears as soon as this power is reached a shift occurs (to prevent back EMF at high RPM). In 5th gear the RPM is allowed to increase and the torque then begins to drop off.

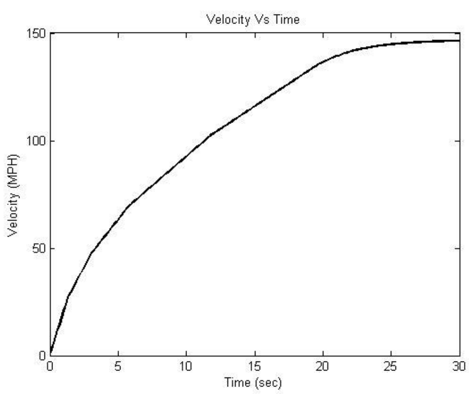
Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results

Ideal settings with single- 9" motor:
0-60 time: 4.606 seconds



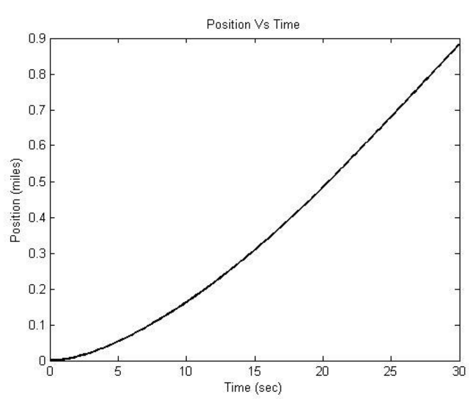
Acceleration Vs Time:

The acceleration in this case is almost half of the ideal case for the two 8". There is more time between shifts because the motor RPM is decreasing slower.



Velocity Vs Time:

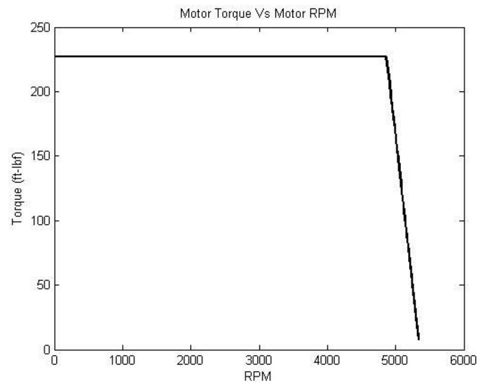
Slower acceleration compared to the two 8".



Position Vs Time:

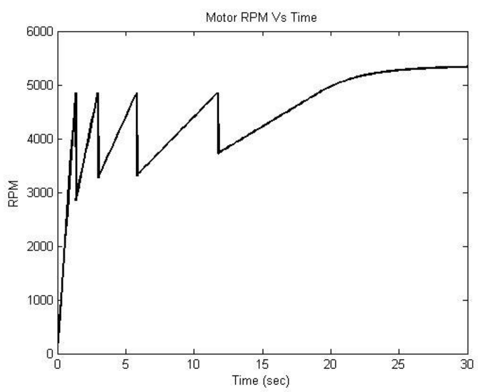
Although each simulation seems to travel the same distance, the 9" takes almost 5 seconds more time to do it.

Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results



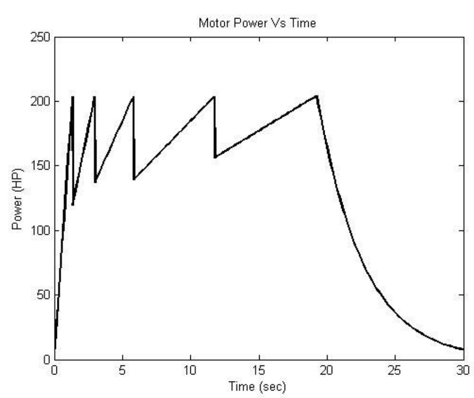
Motor Torque Vs Motor RPM:

Although the torque is higher for a single 9" compared to a single 8", since there are two 8" motors the total torque is greater in the other system.



Motor RPM Vs Time:

The transmission shifts at a lower RPM for this simulation because the power drops off at a lower RPM.

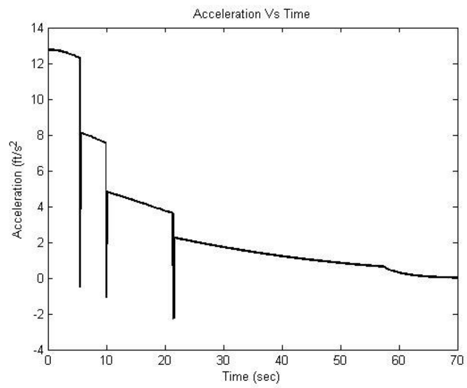


Motor Power Vs Time:

The single 9" has slightly more maximum power than a single 8".

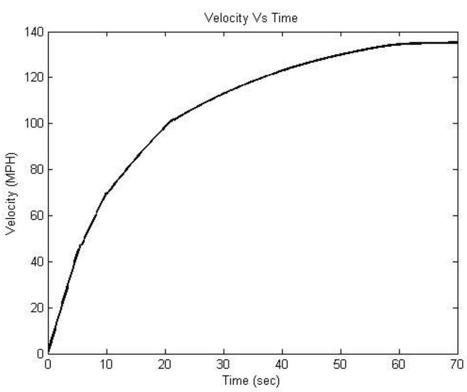
Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results

Realistic settings with single- 9" motor:
0-60 time: 8.090 seconds



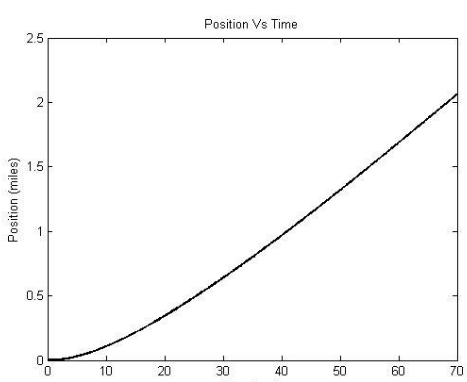
Acceleration Vs Time:

Very similar to the realistic 8" simulation, except the acceleration is significantly less. The shift times are also a lot longer because of the slower acceleration.



Velocity Vs Time:

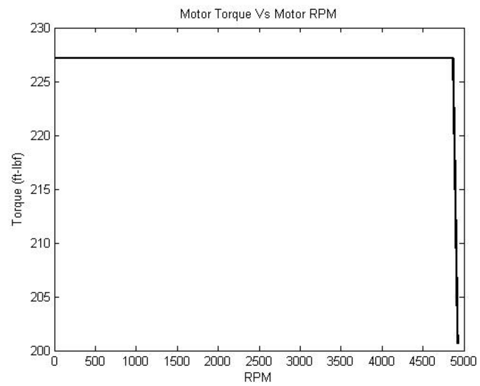
The maximum velocity is nearly 10 MPH less than for the 8" simulation.



Position Vs Time:

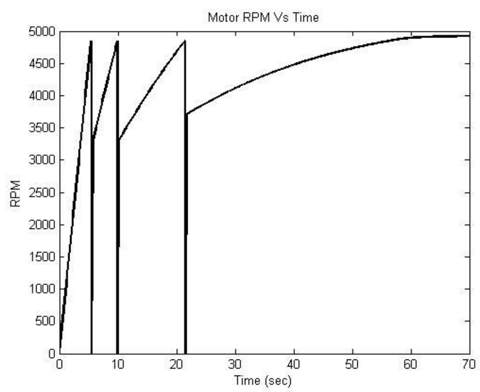
This graph is difficult to compare to the 8" because the time range is so different. This time range was chosen to allow time for the back EMF to set in.

Appendix E.3 – BJ CANE Vehicle Dynamics Simulation Results



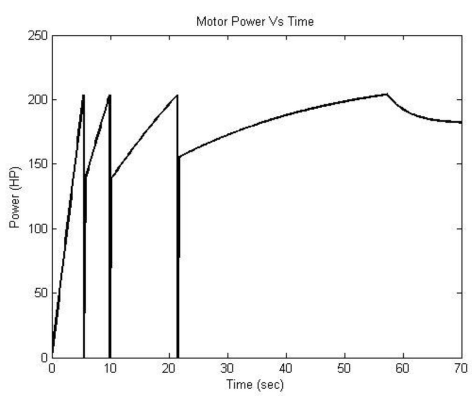
Motor RPM Vs Motor Torque:

The torque begins to drop off at around 4850 RPM in this case.



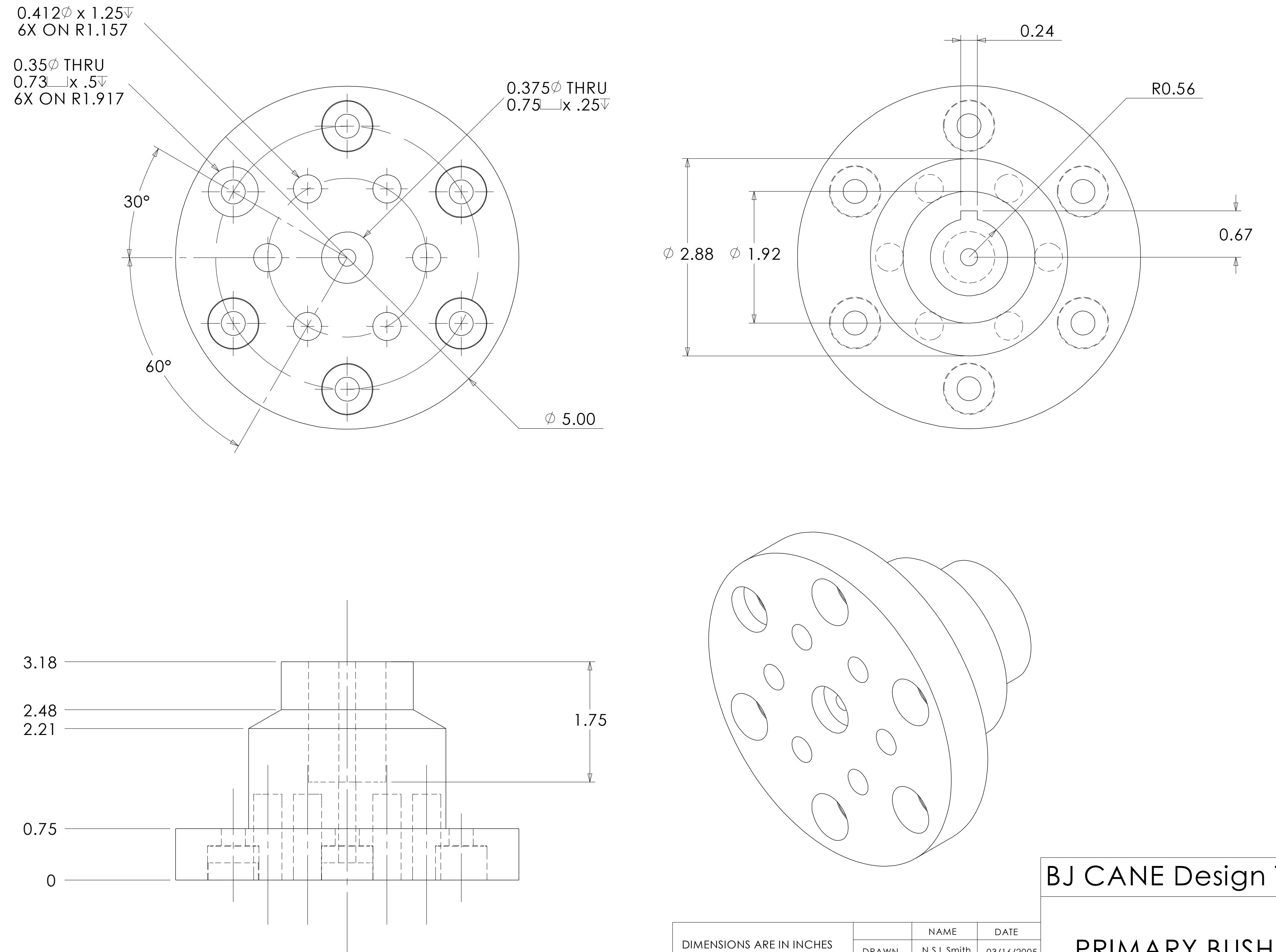
Motor RPM Vs Time:

For this case it takes nearly 60 seconds for back EMF to set in, compared to around 20 seconds for the other setups.



Motor Power Vs Time:

In this case it takes a long time for each gear's maximum power to be reached. For 5th gear it takes nearly 40 seconds to max out, while in the other simulations it took 10 seconds or less.



BJ CANE Design Team

PRIMARY BUSHING

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

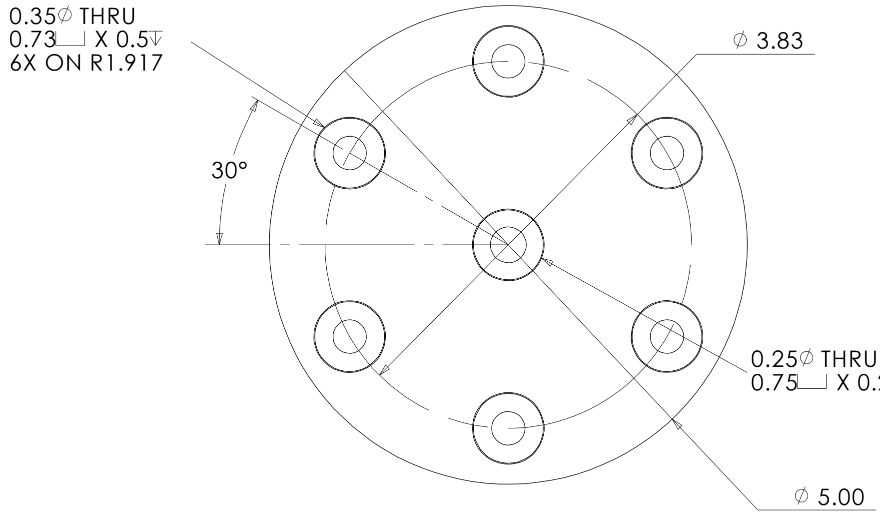
	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

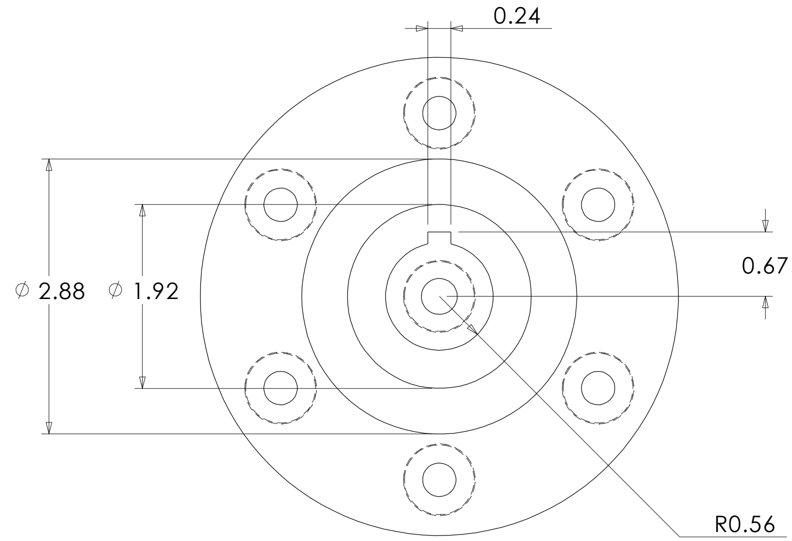
COMMENTS:
DO NOT SCALE DRAWING

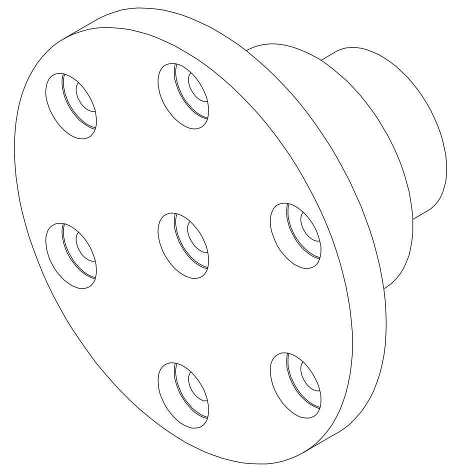
MAKE FROM: ALUMINUM 7075-T651

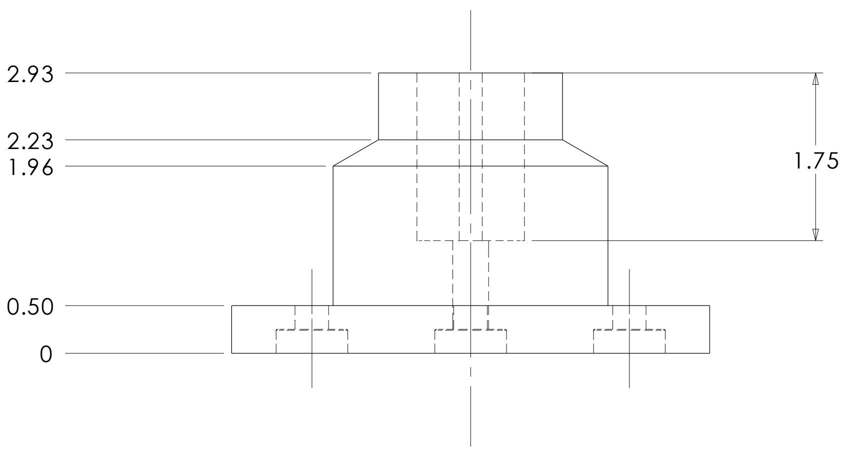
SIZE A | 12Mar05 Bushing (Primary).SLDDRW | REV A

SCALE 1:2 | P/N: BUSH 1 | SHEET 1 OF 1









BJ CANE Design Team

SECONDARY BUSHING

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

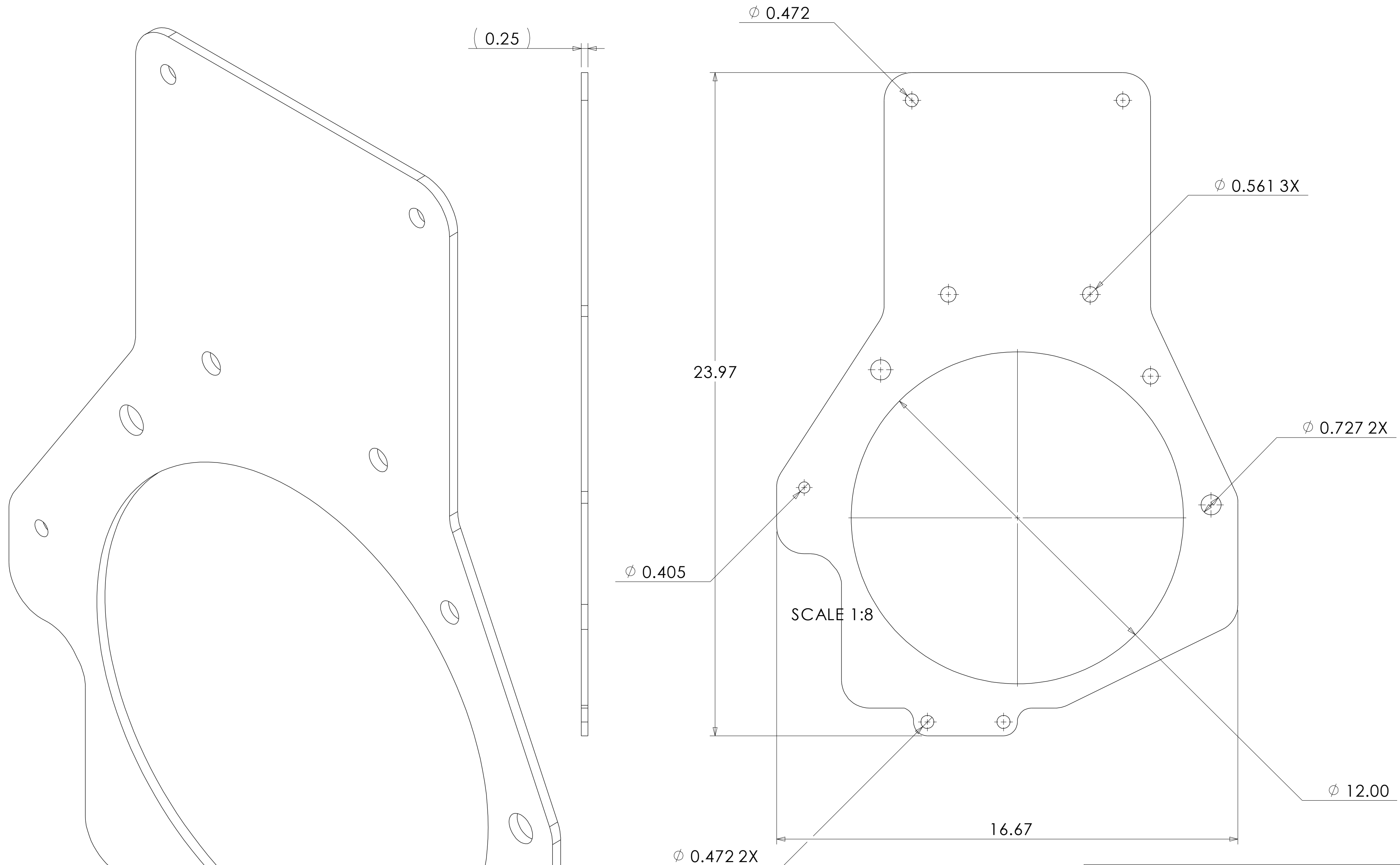
COMMENTS:

DO NOT SCALE DRAWING

MAKE FROM: ALUMINUM 7075-T651

SIZE A | 12Mar05 Bushing (Secondary).SLDDRW | REV A

SCALE 1:2 | P/N: BUSH 2 | SHEET 1 OF 1



BJ CANE Design Team

ADAPTOR PLATE TRANSMISSION END

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/17/2005

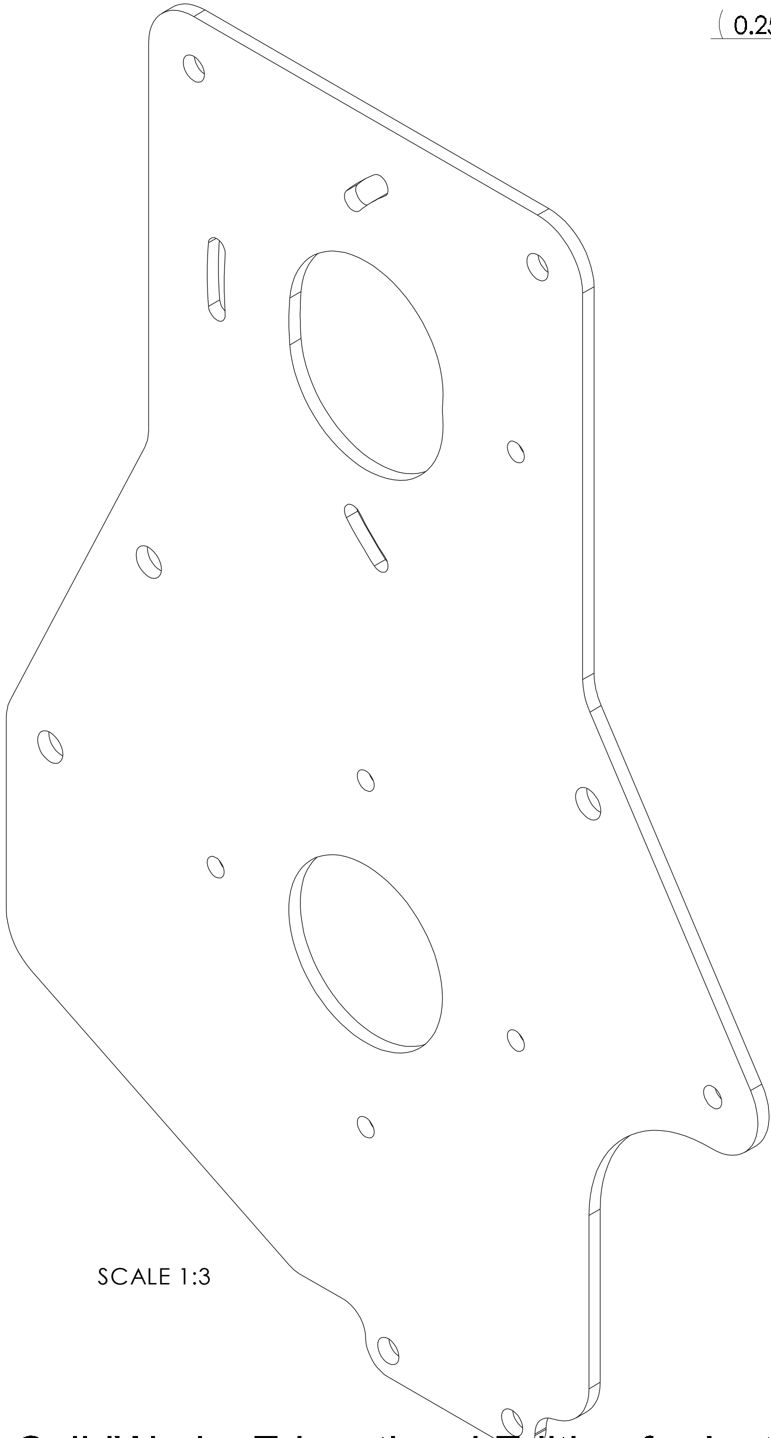
COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

DO NOT SCALE DRAWING

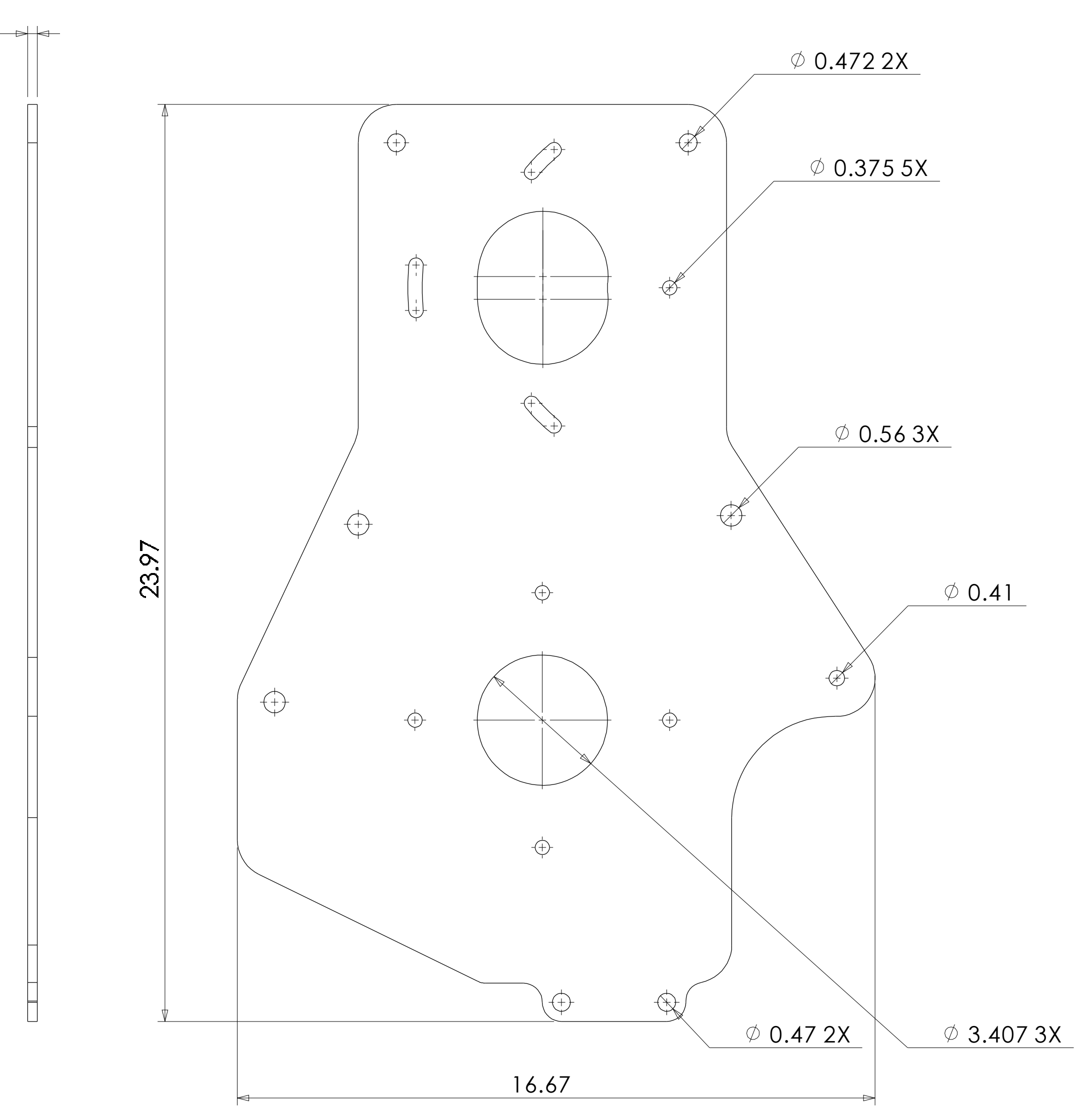
MAKE FROM:
0.25" THK ALUMINUM 6061-T6

SIZE A | 16mar05 Adaptor Plate (Transmission End).SLDDRW | REV A

SCALE 1:5 | P/N: P1 | SHEET 1 OF 1



SCALE 1:3



BJ CANE Design Team

ADAPTOR PLATE MOTOR-DRIVE END

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MAKE FROM: 0.25" THK ALUMINIUM 6061-T6

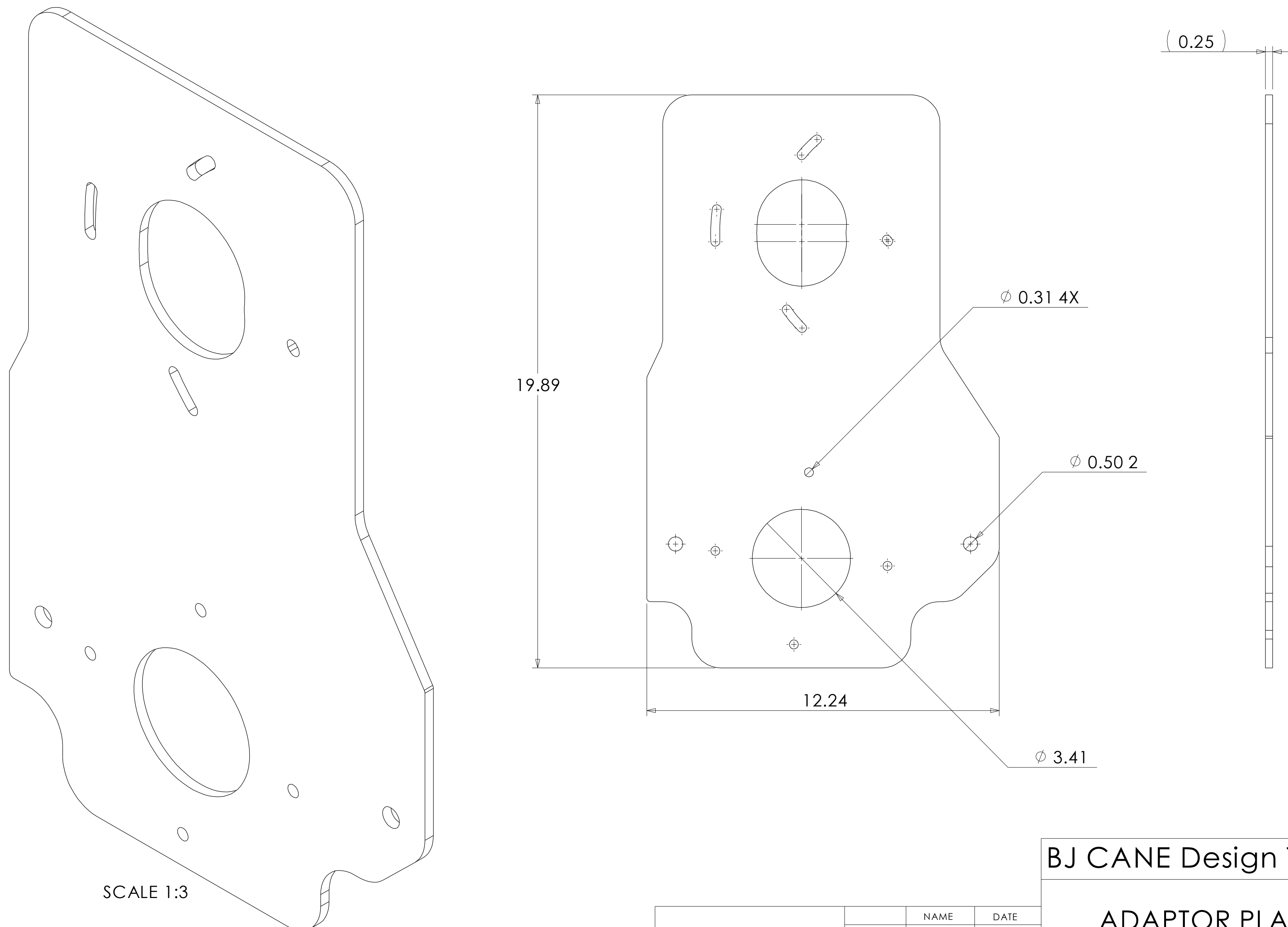
	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/17/2005

COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

DO NOT SCALE DRAWING

SIZE		REV
A	16mar05 Adaptor Plate (Motor Drive End).SLDDRW	A

SCALE 1:5	P/N: P2	SHEET 1 OF 1



BJ CANE Design Team

ADAPTOR PLATE REAR MOTOR SUPPORT

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

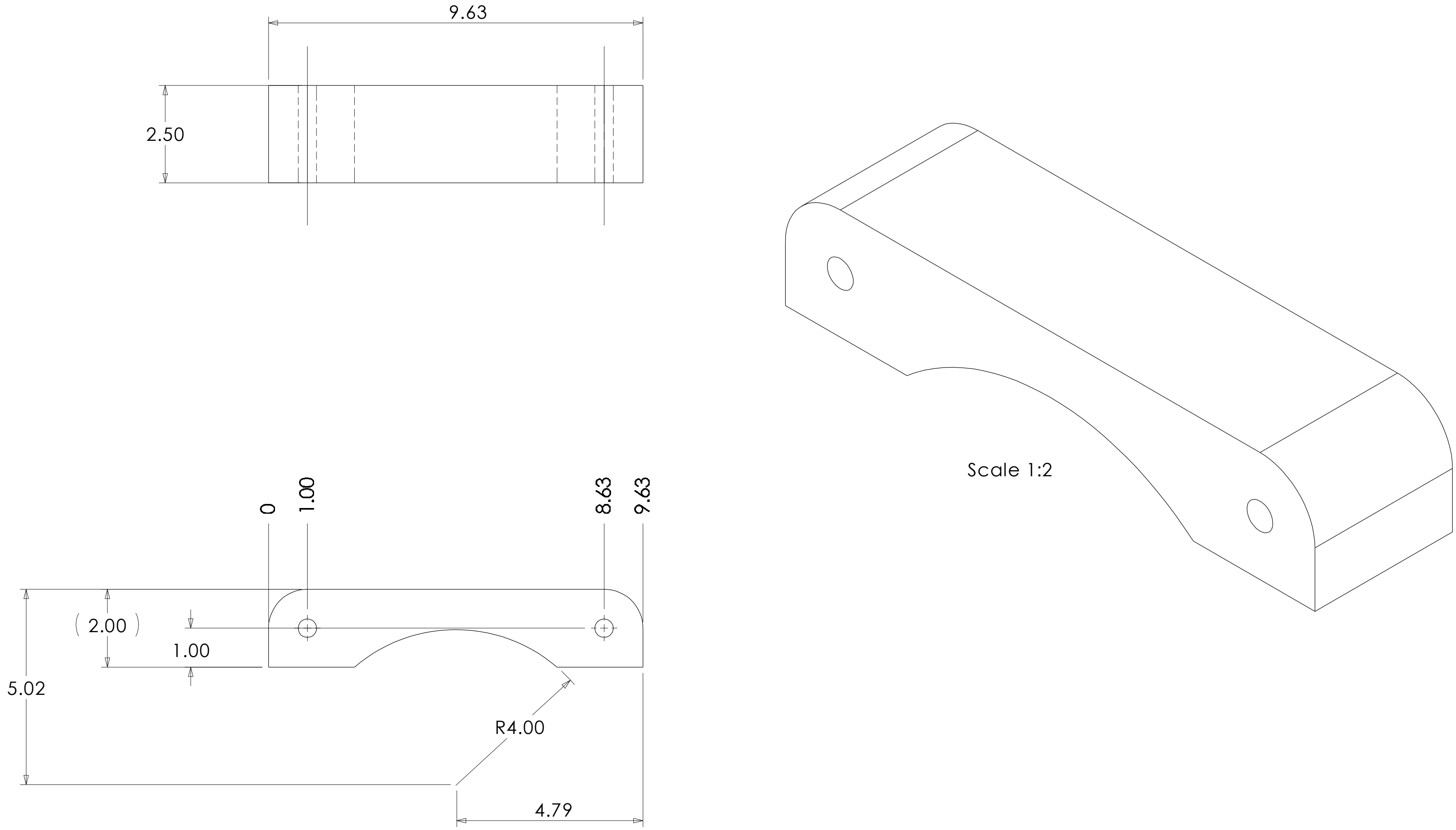
MAKE FROM: .25 THK ALUMINUM 6061-T6

	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J.JOLLY	03/17/2005

COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE
DO NOT SCALE DRAWING

SIZE		REV
A	16mar05 Adaptor Plate (Rear Motor Support).SLDDRW	A

SCALE 1:5 | P/N: P3 | SHEET 1 OF 1



BJ CANE Design Team

ADAPTOR PLATE
TOP SUPPORT STRUCTURE

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MAKE FROM: 2" X 3" STOCK
ALUMINUM 6061-T6

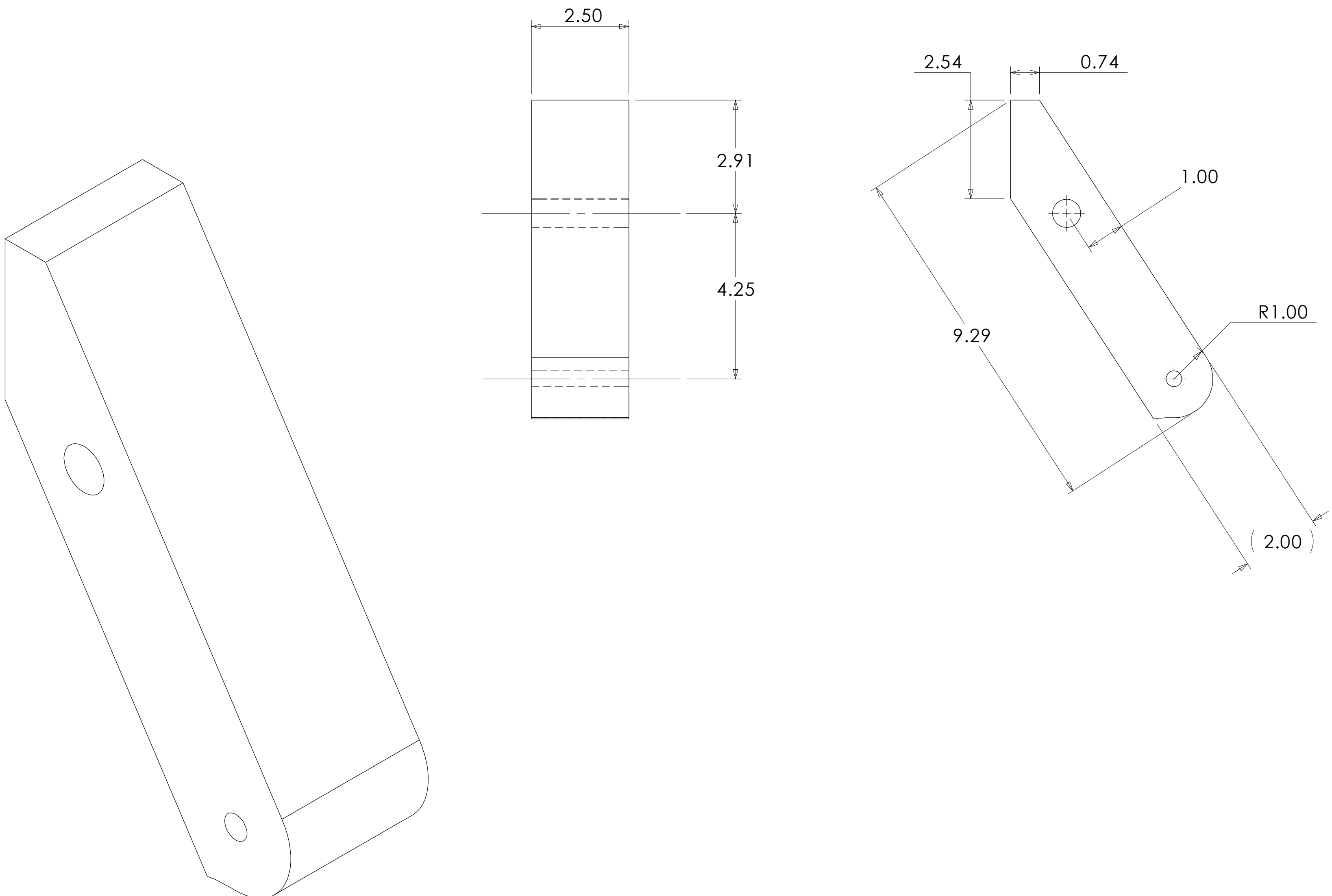
	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

DO NOT SCALE DRAWING

SIZE		REV
A	15mar05 Top Block.SLDDRW	A

SCALE 1:4 | P/N: S1 | SHEET 1 OF 1



Scale 1:3

BJ CANE Design Team

ADAPTOR PLATE
LEFT SUPPORT STRUCTURE

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MAKE FROM: 2" X 3" STOCK
ALUMINIUM 6061-T6

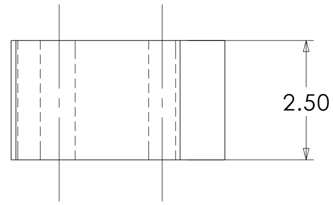
	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

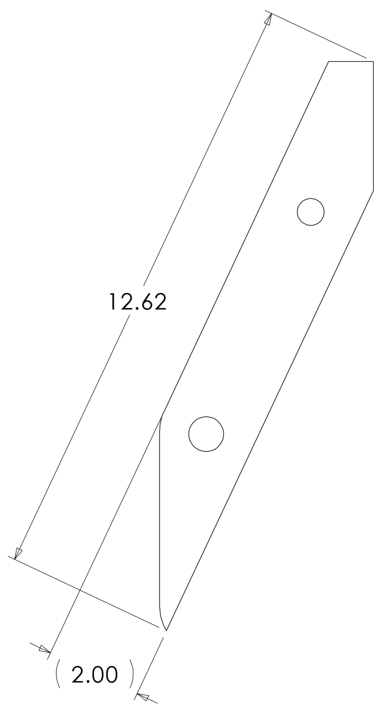
COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

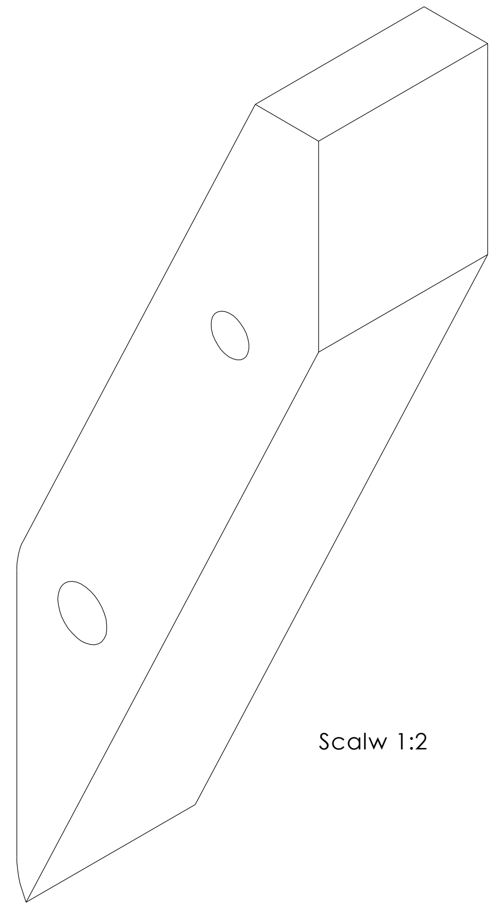
DO NOT SCALE DRAWING

SIZE		REV
A	15mar05 Left Block.SLDDRW	A

SCALE 1:4 P/N: S2 SHEET 1 OF 1







Scalw 1:2

	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MADE FROM: 2" X 3" STOCK
ALUMINIUM 6061-T6

COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

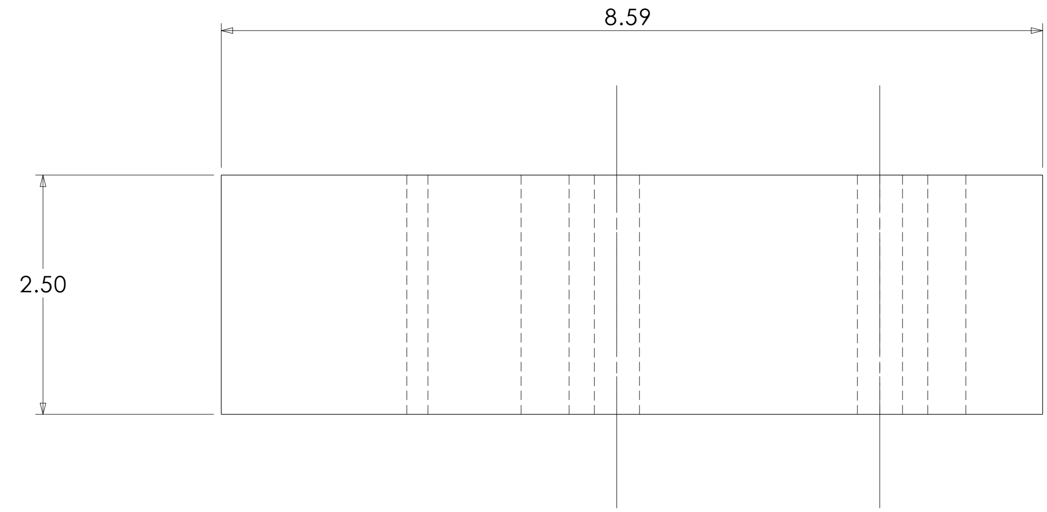
DO NOT SCALE DRAWING

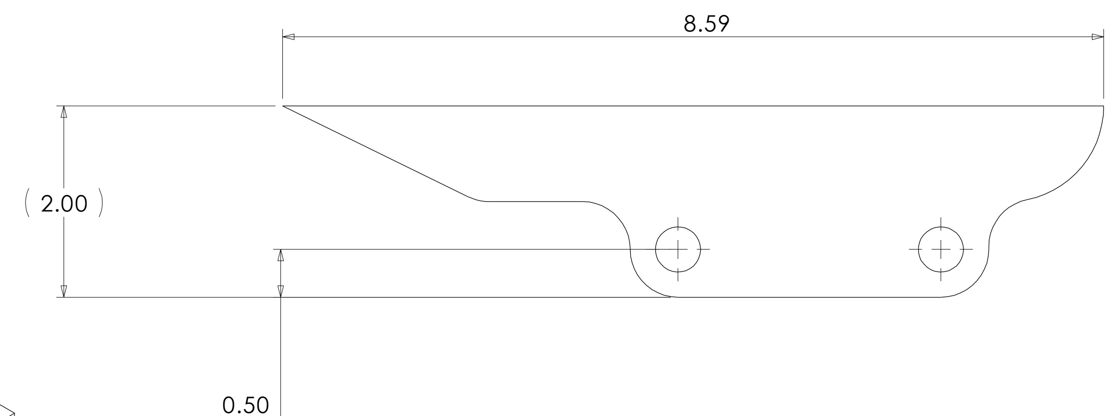
BJ CANE Design Team

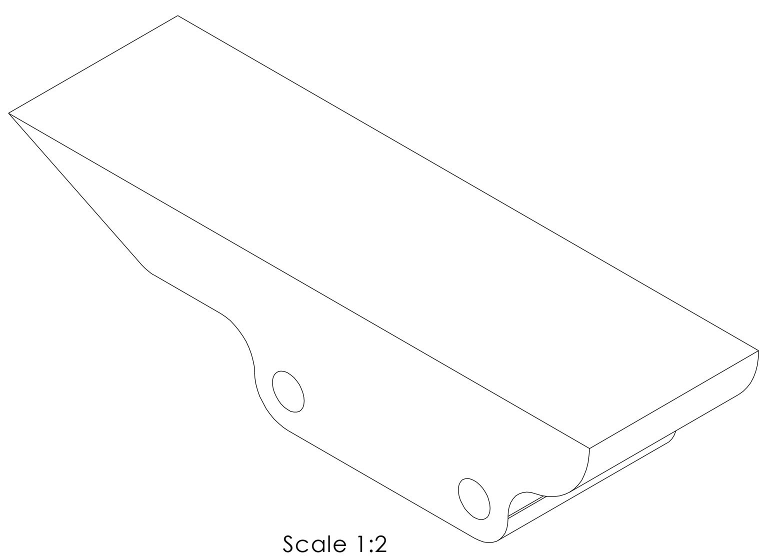
ADAPTOR PLATE
RIGHT SUPPORT STRUCTURE

SIZE A | 15mar05 Right Block.SLDDRW | REV A

SCALE 1:4 | P/N: S3 | SHEET 1 OF 1







Scale 1:2

BJ CANE Design Team

ADAPTOR PLATE
LOWER SUPPORT STRUCTURE

DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MAKE FROM: 2" x 3" STOCK ALUMINIUM 6061-T6

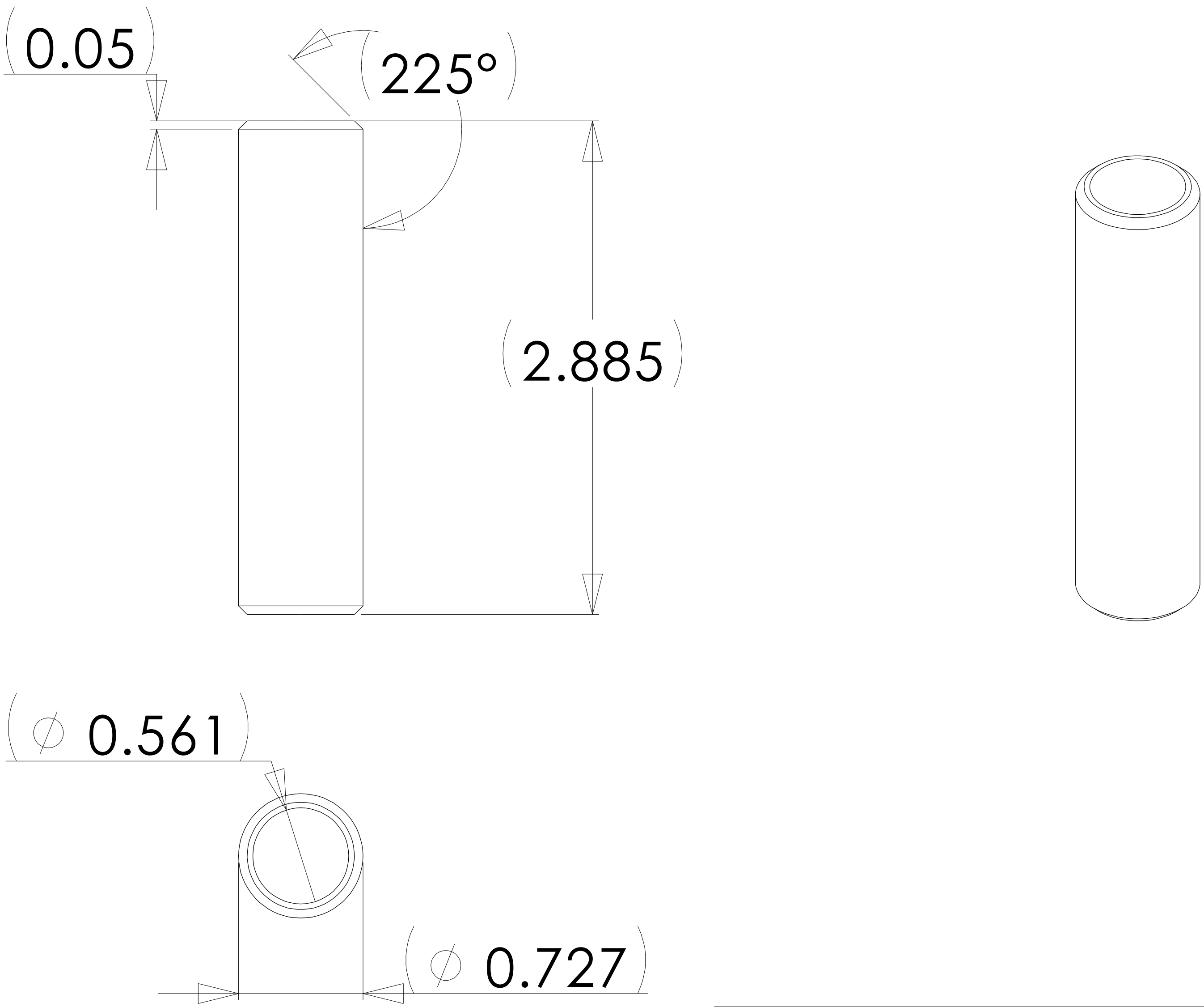
	NAME	DATE
DRAWN	N.S.L.Smith	03/16/2005
CHECKED	J. JOLLY	03/16/2005

COMMENTS:
OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

DO NOT SCALE DRAWING

SIZE		REV
A	15mar05 Lower Block.SLDDRW	A

SCALE 1:2	P/N: S4	SHEET 1 OF 1



DIMENSIONS ARE IN INCHES
TOLERANCES: ±.003
ANGULAR: ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

MATERIAL

1018 Cold Rolled Steel

	NAME	DATE
DRAWN	C. Bates	3/17/05
CHECKED	N.S.L. Smith	3/17/05

COMMENTS:

OBTAIN SOLID MODEL FROM BJCANE DESIGN TEAM FOR MANUFACTURE

DO NOT SCALE DRAWING

BJ CANE Design Team

Alignment Pin

SIZE	DWG. NO.	REV.
A	17Mar05 Alignment Pin.SLDPRT	A

SCALE:1:1	WEIGHT:	SHEET 1 OF 1

Appendix G – References

Material Reference Information

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Appendix G – References

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http://greenplanet3.org/ev/

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http://evcl.com/914/

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Interview

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