MET 214 GO KART PROJECT

The Dirt Devil by www.diyden.net

Design Analysis Performed By:

John Murray
and
Tek Lentine

Problem Statement and Concept Description

Distribution: N. Kundu

Revision: 1
Revision Date: 4/8/2004
Project Number: MET 214

Project Authorization: **Go Kart**

Project Scope:
Design a single passenger Go Kart that does not exceed 1000 lbs. 750 lbs. for the Go Kart 250 lbs. for the Passenger

Background:
Go Kart is to be designed such that it can be sold at a competitive price for this market.

Purchasing / Manufacturing Tasks:
Weld/Build Prototypes for Engineering Materials - Availability & Stocking

Manufacturing Engineering Tasks:
Nesting Programs Tooling Fixturing Time Studies

Marketing Tasks:
Marketing Analysis Reports Competitors Costs and Analysis Reports Product Catalogs

Engineering Tasks:
Project Description
Frame Wheel Loads Friction Factors Power Transmission Braking System

Engineering Tasks Continued:
Engineering Calculations & Parts Selection for the Frame
Loading Distribution Moments and Shear Diagrams Material Selection and Justifications
Drive System
Loads Force Required Torque Required Horsepower Required
Chain/Belt Selections
Length Speed Ratio Large and Small Sprocket/Pulley/Clutch
Axle/Shaft
Torque TE Moment ME Keyways Kt Factor Shaft Size Material
Bearings
C P L10
Braking System
Driving Course Materials (Terrain) Tire Material Friction Torque

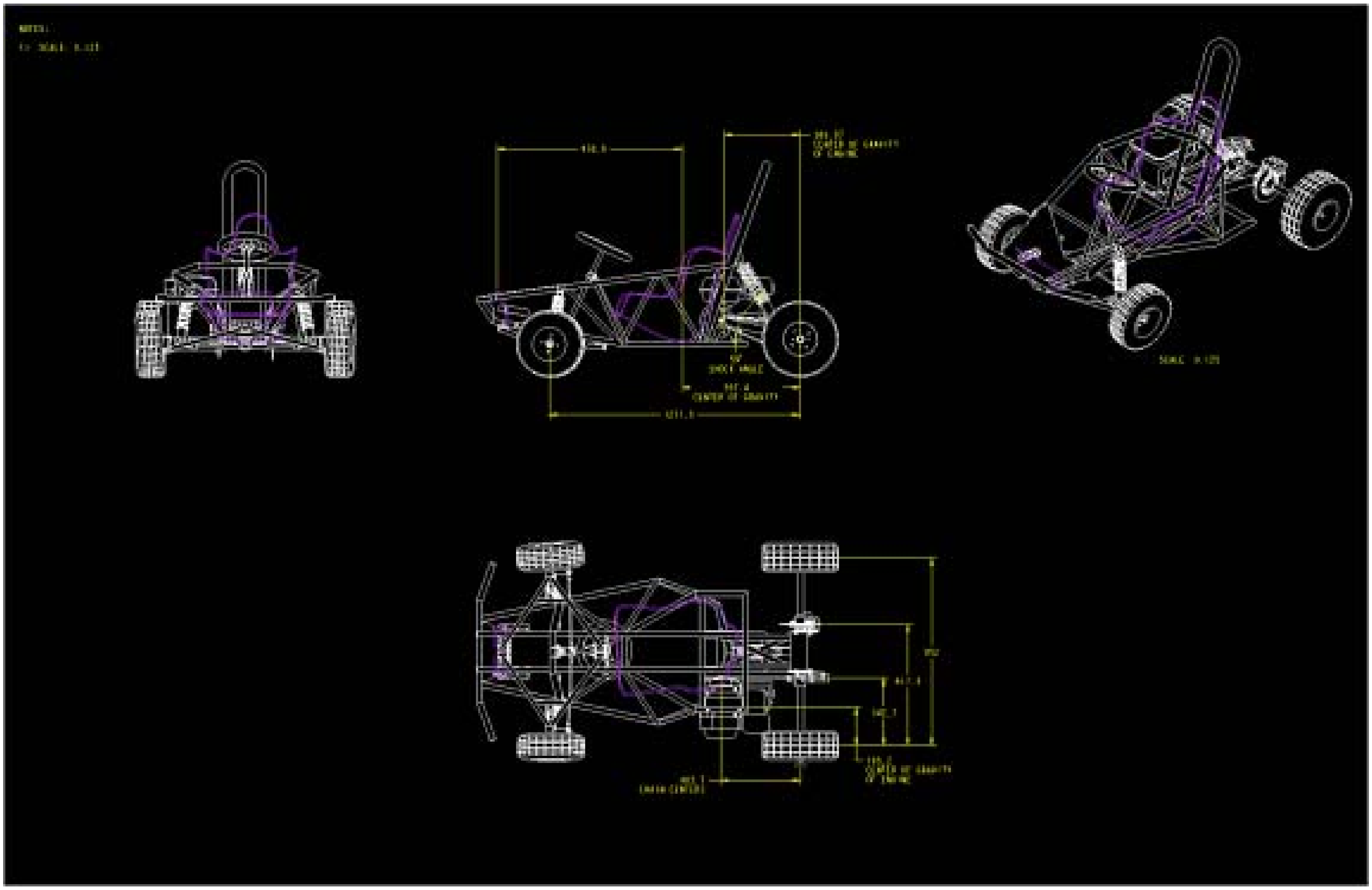
Engineering Tasks Continued:
Bill of Materials
Descriptions Quantities Assemblies / Sub-Assemblies / Parts Breakdowns Part Locations Nesting Locations/Stages Drawing References
Presentation

Target Completion Dates:

04/01/2004	Problem Statement / Concept Description
04/08/2004	Concept Drawings
04/15/2004	Load Distribution on Frame Moment and Shear Diagrams
04/22/2004	Drive System Load, Force, Torque, Power Axle/Shaft
04/29/2004	Bearings Brakes B.O.M.
05/06/2004	Presentation

Assigned To: Tek Lentine & John Murray

Drawing and Critical Dimensions



Drawing is not to scale. A full size plot of this drawing is attached to this report. (Comment from Don- This is not available)

Load Distribution

Overview

Weight is relatively evenly distributed among the four wheels of the vehicle. The right rear wheel showed about 14% heavier load than the left rear wheel. This is due mostly to the positioning of the engine. The following table shows a breakdown of the components that contributed to the vehicle load calculations.
(Comment from Don- The Seat and driver is actually offset to the left-opposite side to the engine for space reasons and to balance the load of the engine on the rear wheels)

Description	Right Front Wheel Load (N)	Left Front Wheel Load (N)	Right Rear Wheel Load (N)	Left Rear Wheel Load (N)
Frame, seat, and driver	451.3	451.3	509.2	509.2
Engine and clutch	46.3	11.9	105.7	27.3
Chain	0	0	14.5	8.1
Rear Shock	0	0	8.8	8.8
Wheels	44.1	44.1	44.1	44.1
Front Shocks	17.7	17.7	0	0
Total	559.4	525	682.3	597.5

Total Weight of Vehicle (kg): 241.0

Data

Weight (kg)	Description	Location
59.8	Frame	Point load at 597.4 mm forward of rear axle center
136.1	Seat and Driver	Point load at 597.4 mm forward of rear axle center
4.5	Wheels	Point load at center of each wheel
19.5	Engine/Clutch	Point load at 386.9 mm forward of rear axle center
2.3	Chain	Point load at center of rear axle
1.8	Front Shocks	Point load at center of each front wheel
1.8	Rear Shock	Point load at center of rear axle

Calculations

Reaction force at wheels due to frame, seat, and driver:

Distance from load to center of rear wheels: $L_R = 597.4$ mm
Distance from load to center of front wheels: $L_F = 674.1$ mm
Total load: $F = 1921.1$ N

$F * L_F + R_R * (L_R + L_F) = 0$
$R_R = -F * L_F / (L_R + L_F)$
$R_R = -1921.1$ N * 674.1 mm / (597.4 mm + 674.1 mm)
$R_R = -1018.5$ N = Reaction force at both rear wheels

Reaction force at each rear wheel: -509.2 N

$R_F = F + R_R$
$R_F = 1921.1$ N – 1018.5 N
$R_F = -902.6$ N = Reaction force at both front wheels

Reaction force at each front wheel: -451.3 N

Reaction force at wheels due to Engine/Clutch

Distance from load to center of rear wheels front to back: $L_R = 386.9$ mm
Distance from load to center of front wheels front to back: $L_F = 884.6$ mm
Distance from load to center of right rear wheel: $L_{RR} = 195.2$ mm
Distance from load to center of left rear wheel: $L_{LR} = 756.8$ mm
Total load: $F = 191.2$ N

$F * L_F + R_R * (L_R + L_F) = 0$
$R_R = F * L_F / (L_R + L_F)$
$R_R = 191.2$ N * 884.6 mm / (386.9 mm + 884.6 mm)
$R_R = 133.0$ N = Reaction force at rear

$R_R * L_{RR} + R_{LR} * (L_{RR} + L_{LR}) = 0$

$R_{LR} = R_R * L_{RR} / (L_{RR} + L_{LR})$
$R_{LR} = -133.0\ N * 195.2\ mm / (195.2\ mm + 756.8\ mm)$

Reaction force at left rear wheel: $\mathbf{R_{LR} = -27.3\ N}$

$R_{RR} = R_R + R_{LR}$
$R_{RR} = 133.0\ N – 27.3\ N$

Reaction force at right rear wheel: $\mathbf{R_{RR} = -105.7\ N}$

$R_F = F - R_R$
$R_F = 191.2\ N – 133.0\ N$
$R_F = 58.2\ N$

$R_{LF} = R_F * (R_{LR} / R_R)$
$R_{LF} = 58.2\ N * (27.3\ N / 133.0\ N)$

Reaction force at left front wheel: $\mathbf{R_{LF} = -11.9\ N}$

$R_{RF} = R_F - R_{LF}$
$R_{RF} = 58.2\ N – 11.9\ N$

Reaction force at right front wheel: $\mathbf{R_{RF} = -46.3\ N}$

Reaction force at wheels due to chain:

Distance from load to center of right wheel:	$L_R = 342.7\ mm$
Distance from load to center of left wheel:	$L_L = 609.3\ mm$
Total load:	$F = 22.6\ N$

$F * L_R + R_L * (L_R + L_L) = 0$
$R_L = -F * L_R / (L_R + L_L)$
$R_L = -22.6\ N * 342.7\ mm / (342.7\ mm + 609.3\ mm)$

Reaction force at left rear wheel: $\mathbf{R_L = -8.1\ N}$

$R_R = F + R_L$
$R_R = 22.6\ N – 8.1\ N$

Reaction force at right rear wheel: $\mathbf{R_R = -14.5\ N}$

Reaction force at wheels due to rear shock:

Shock is centered on rear axle so load is equally distributed on both rear wheels.

Reaction force at each rear wheel: R_R = -8.8 N

Reaction force at wheels due to front shocks:

Shocks are close enough to tires to be considered point loads at the front wheels for the purposes of this analysis.

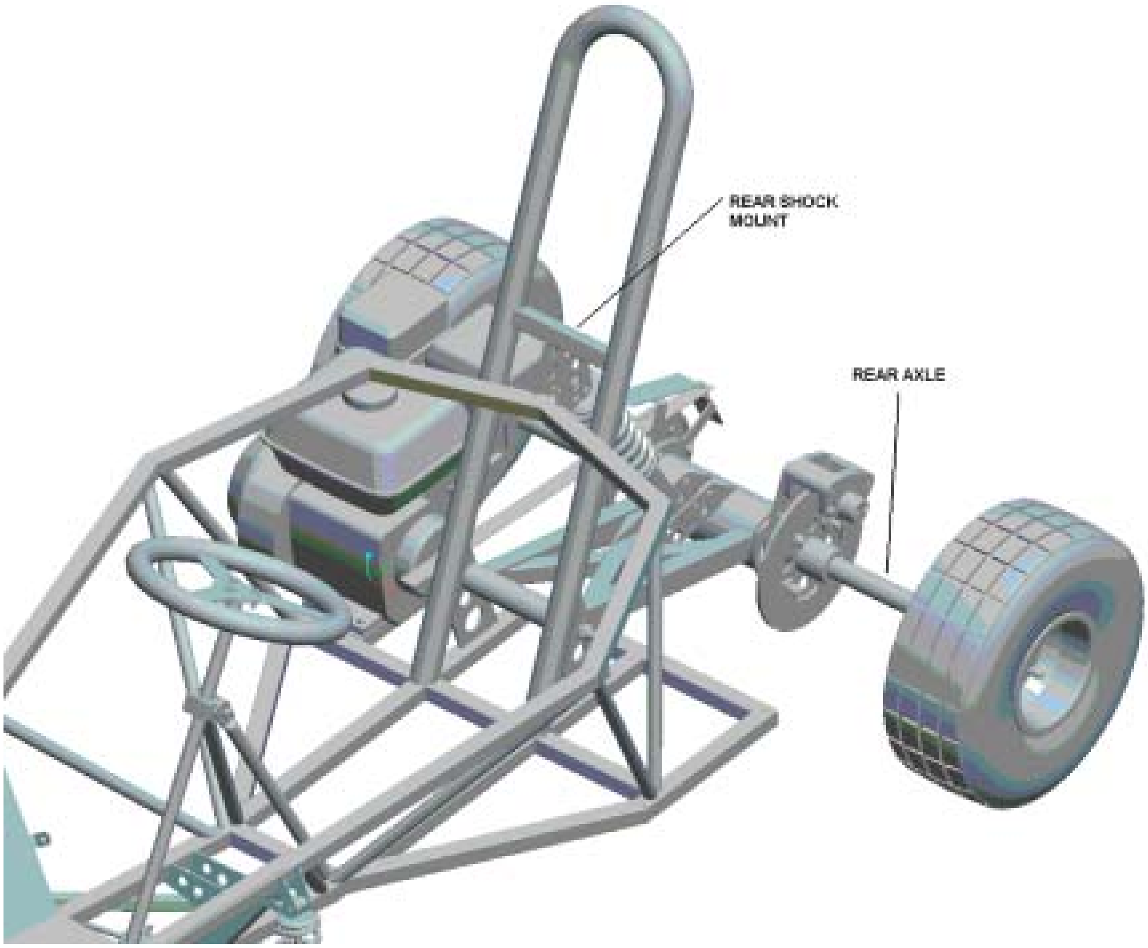
Reaction force at each front wheel: R_F = 17.7 N

Reaction force at wheels due to wheels:

Wheels can each be considered point loads at their centers:

Reaction force at each wheel: R = 44.1 N

Maximum Stress Locations



Overview

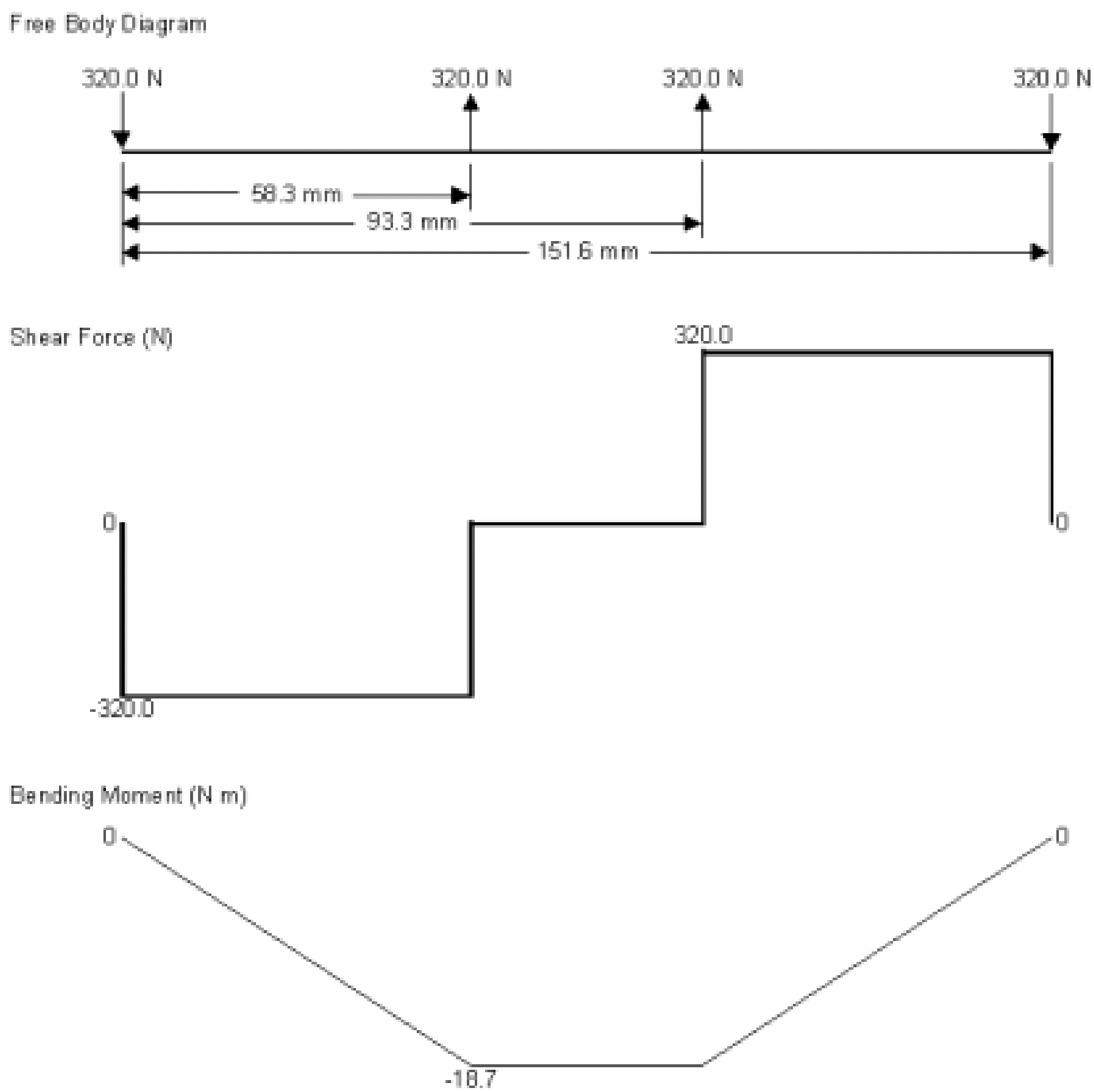
Both the frame and axles were analyzed for maximum stresses during static loading. The load values used can be found in the “Load Distribution” section of this report. A safety factor of 12 was used based on published values for bending stresses in a shock application[1].

The maximum stress in the frame is due to bending and occurs in the cross brace that supports the rear shock mount. It was found that the stresses at the rear shock mount justify the use of common, steel tube such as ASTM A500 Grade A. Moreover, since the rear shock brace represents the maximum stress found in the frame, this same material can be used for the entire frame. (Comment from Don- The load from the rear shock is actually spread by the shock mount plates between both the cross brace and the top frame rails)
A much higher stress occurs in the rear axle because of the distance from where it is constrained to the end of the axle. The stress found necessitates a much stronger material than is needed for the frame. AISI 4140 OQT 700 or stronger should be used. This

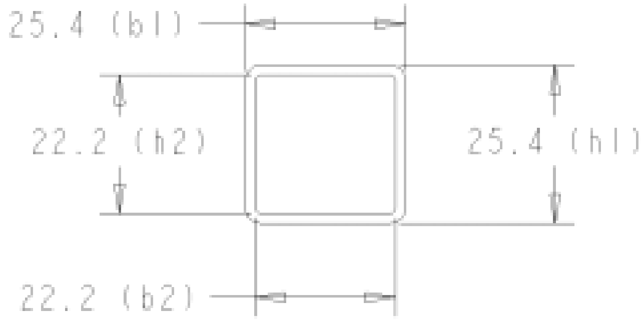
material has an elongation of 12% and should be ductile enough to handle the fatigue that will be present in this application. If another material is substituted, care should be taken not to use one with a lower elongation as premature failure due to fatigue may occur.

Additionally, an FEA software package (Cosmos Works) was used to perform static stress and deformation analysis on both the rear axle and the front spindle. As expected, the front spindle did not show maximum stresses as high as the rear axle, but the stresses were high enough to require the same material. The maximum stress found in the rear axle was close enough to the stress found manually to lend credibility to both analyses; however, the FEA did show a small area of stress that was more than double the stresses seen in the rest of the axle. Testing and special attention should be paid to this area of the axle to ensure that premature failure will not occur here (see FEA graphics below).

Rear Shock Brace Calculations



Cross section (units in mm)
:



Moment of inertia:

Moment of inertia of rectangle[2]: $b\,h^3 / 12$

$I_1 = (25.4\text{ mm})\,(25.4\text{ mm})^3 / 12$
$I_1 = 3.468\text{E-8 m}^4$

$I_2 = (22.2\text{ mm})\,(22.2\text{ mm})^3 / 12$
$I_2 = 2.024\text{E-8 m}^4$

$I = I_1 - I_2 = 1.445\text{E-8 m}^4$

Section modulus[3]:

C = 12.7 mm

$S = I / C = 1.137\text{E-6 m}^3$

Max stress[4]:

M = 18.7 N m (See bending moment diagram)

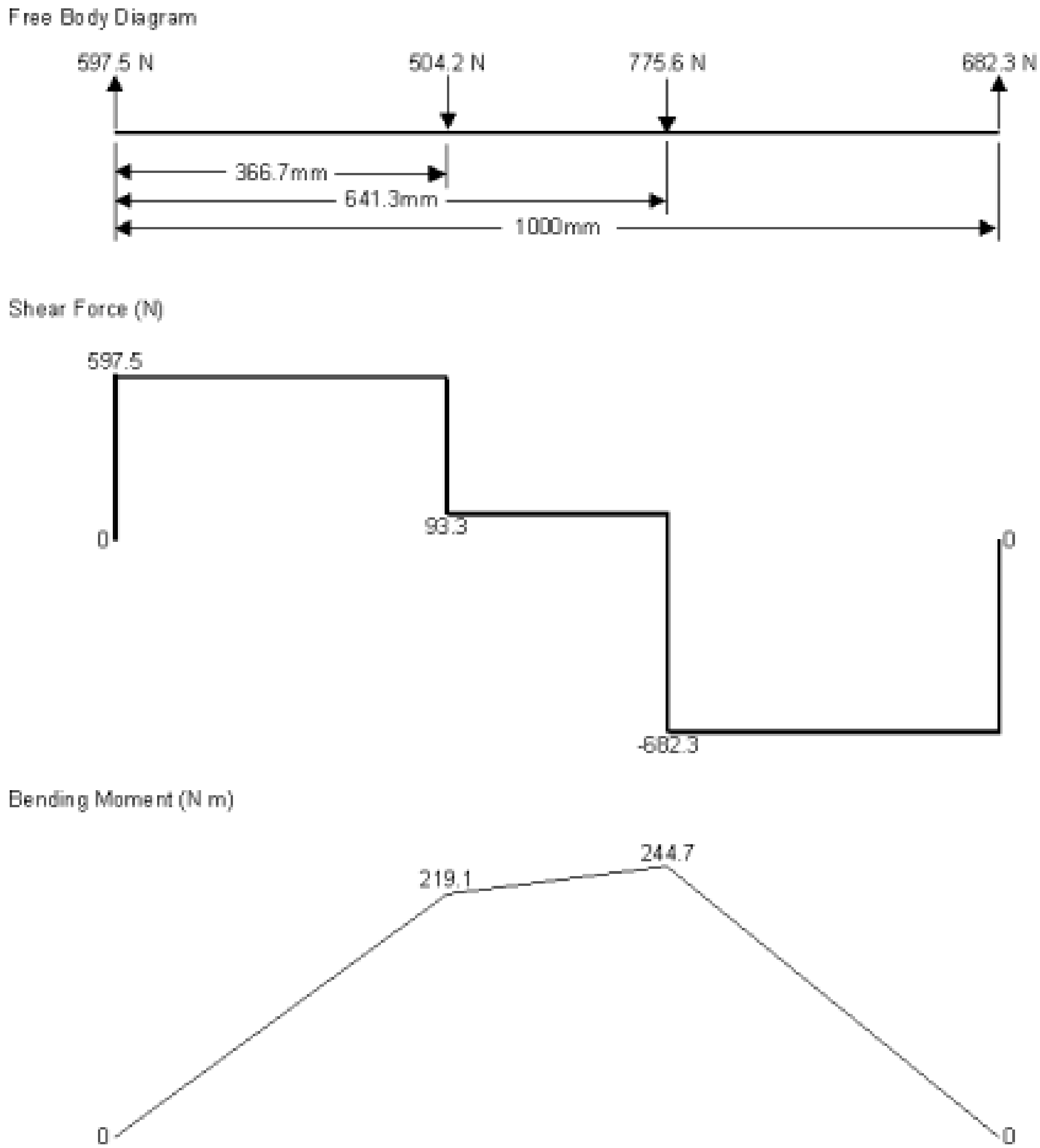
$\sigma_{max} = M / S = 16.4$ Mpa

Material, ASTM A500 Grade A Square Tube:

Safety Factor[1]: N = 12 (Shock Application)
Ultimate Strength: su = 310 MPa
Design Stress: $\sigma_d = su / N = 25.8$ Mpa

$\boldsymbol{\sigma_d > \sigma_{max}}$
Material is adequate

Rear Axle Calculations



Moment of inertia:

Diameter of Axle: D = 30.0 mm

Moment of inertia of cir[2]: $\pi D^4 / 64$

$I = \pi (30.0 \text{ mm})^4 / 64$
$I = 3.976E\text{-}8 \text{ m}^4$

Section modulus[3]:

C = 15.0 mm

$S = I / C = 2.651E\text{-}6 \text{ m}^3$

Max stress[4]:

M = 244.7 N m (See bending moment diagram)

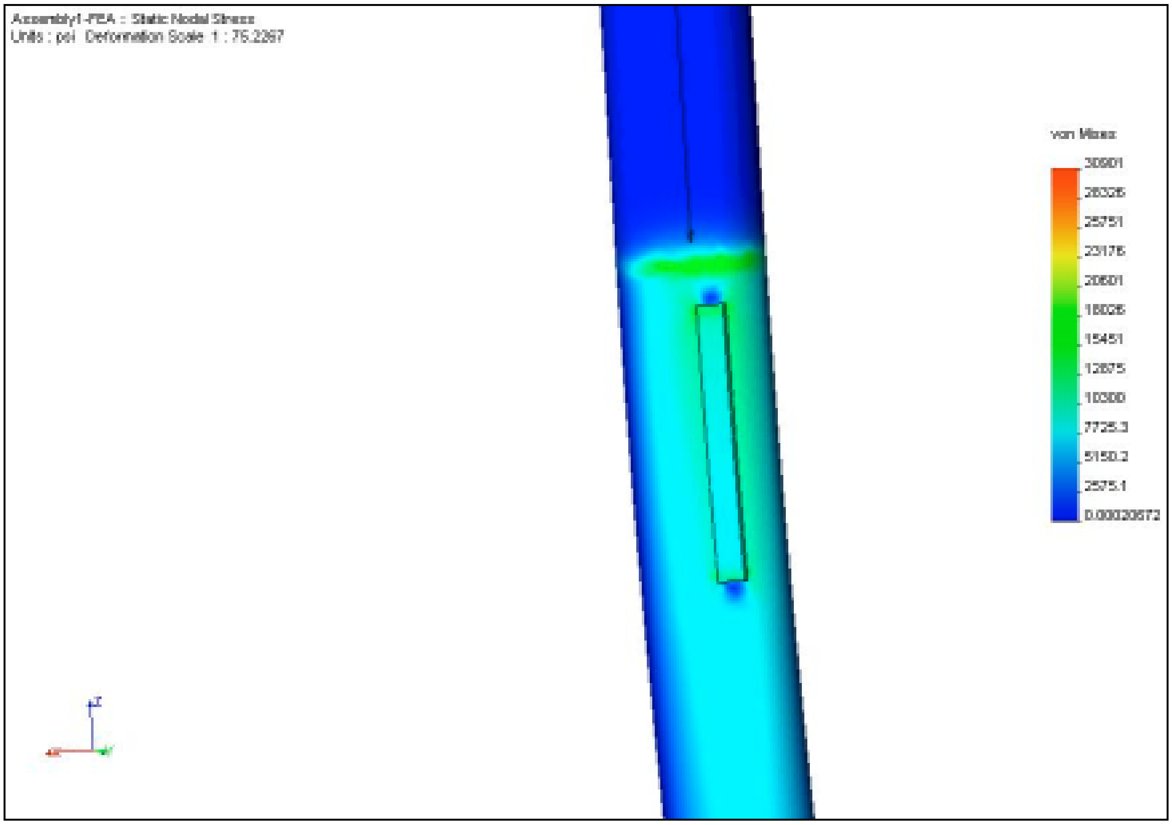
$\sigma_{max} = M / S = 92.3$ Mpa

Material, AISI 4140 OQT 700:

Safety Factor[1]:	N = 12 (Shock Application)
Ultimate Strength:	su = 1593 MPa
Design Stress:	$\sigma_d = su / N = 132.8$ MPa

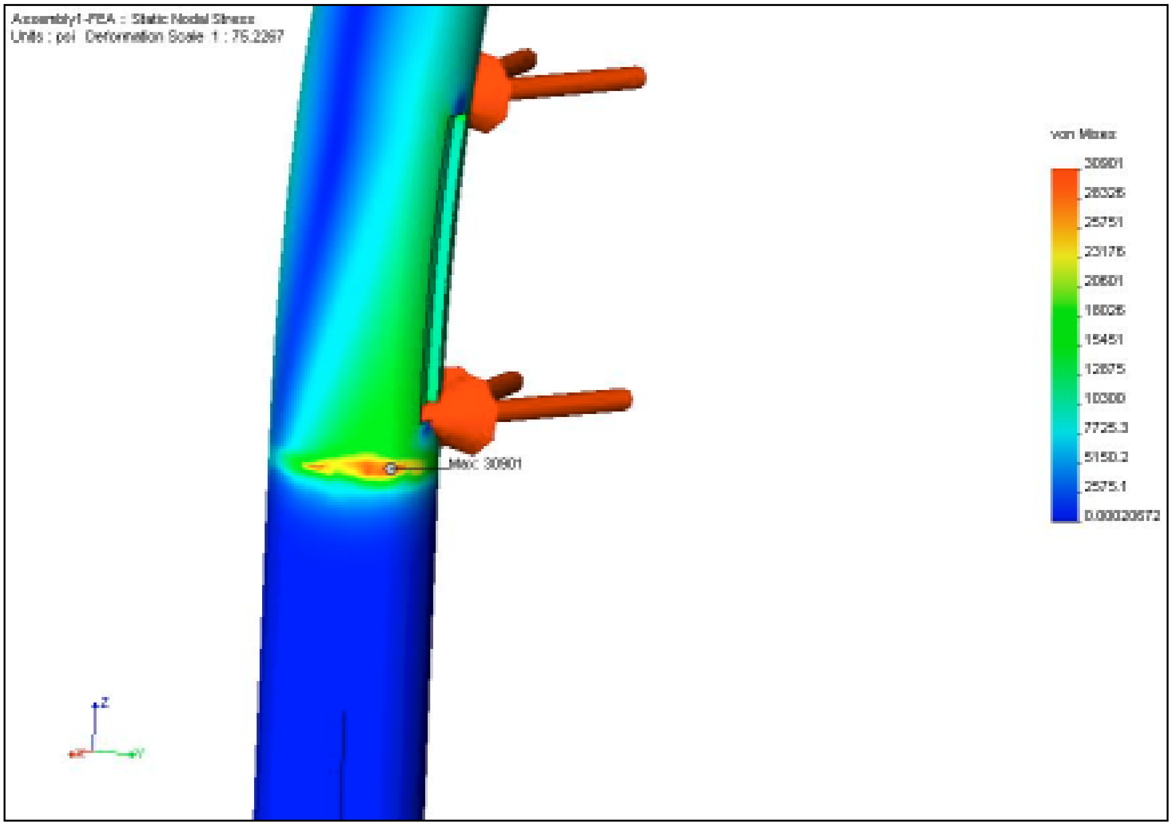
$\boldsymbol{\sigma_d > \sigma_{max}}$
Material is adequate

Rear axle finite element analysis:

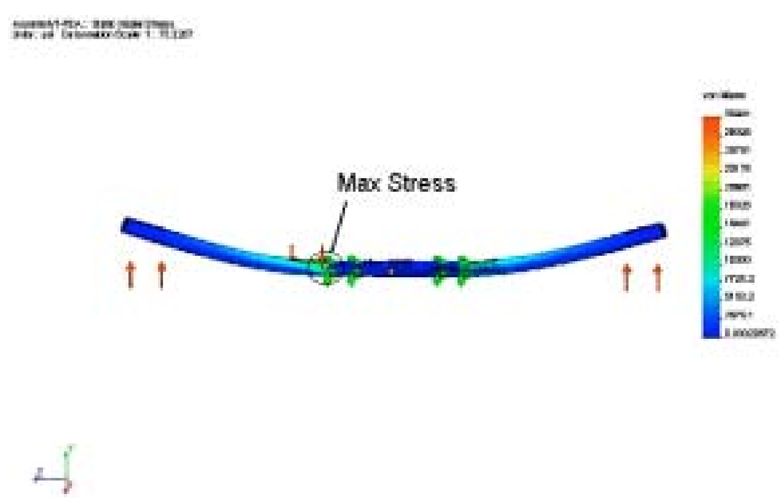


This is the typical max stress found (about 106 Mpa). This is higher than the stress found manually by 15%, but still well below the design stress of the material (133 MPa).

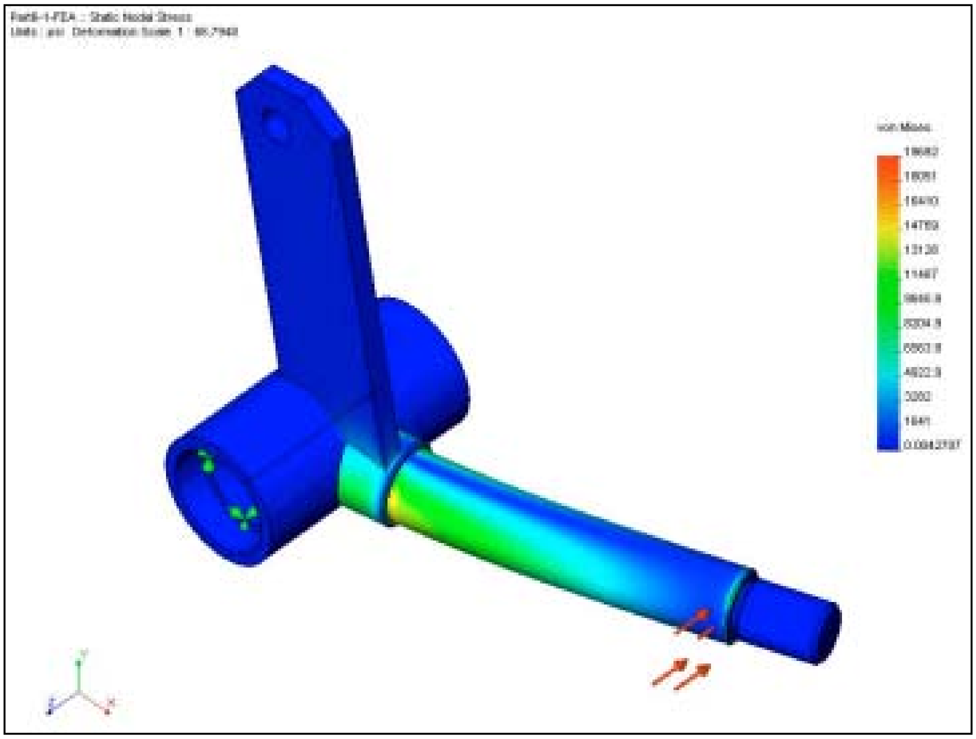
Rear axle finite element analysis:

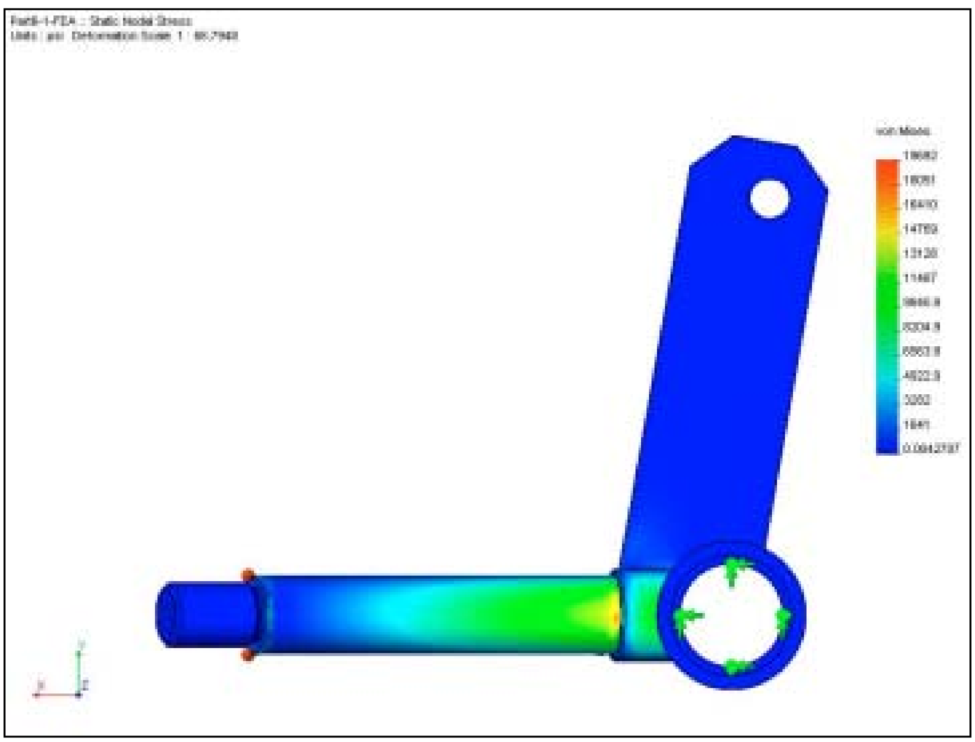


This view shows a stress concentration that was not caught in the manual analysis. The stress in this small area is 213 MPa which is 131% higher than the manually calculated max stress, and 60% higher than the design stress of the material (133 Mpa). Because it is such a small area, the best course is to test the design and pay special attention the behavior of the axle in this area.



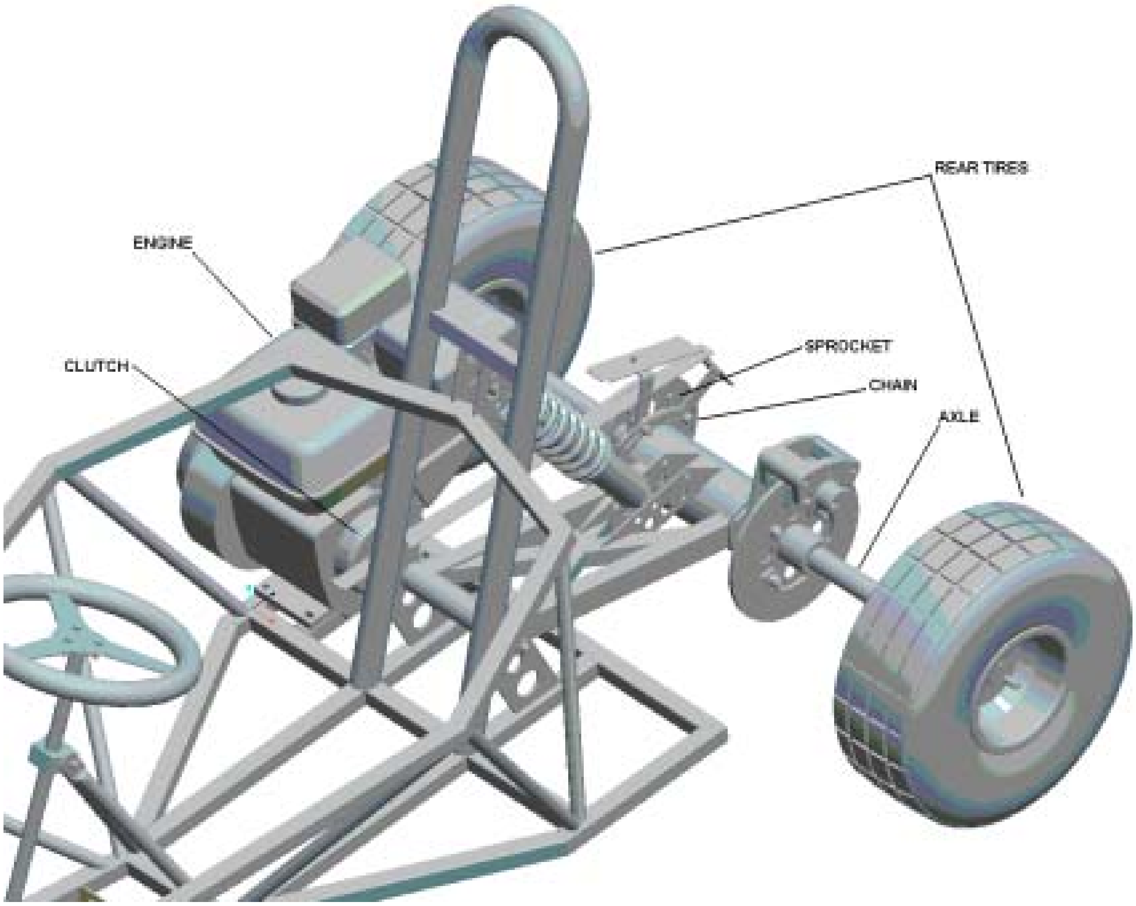
Front spindle finite element analysis:





The maximum stress in the front spindle is shown above in the area colored red. The stress is 136 MPa which exceeds the design stress by 2%. Because of the small margin by which the design stress is exceeded and the small area that experiences the stress, the same material used for the rear axle should be adequate for the front spindle.

Drive System



Overview

The engine is a 7.0 HP Robin-Subaru EX Series overhead cam with a maximum power output of 7.0 HP and a recommended continuous power output of 5.0 HP. Maximum torque for the engine is 10.26 ft-lbs at 2500 RPM.

The clutch is a Comet Industries CSC 400 series with a ¾" bore. This is a very affordable clutch and requires low maintenance when compared with many other popular clutches. It has a twelve tooth sprocket with a diameter of 1.45".

The rear tires are 15" outer diameter and 6" wide (15 x 6.00-6). They are widely available in differing tread patterns at many go kart or lawn and tractor retailers.

The axle is a solid (live) axle and is sprocket driven. The sprocket has 66 teeth and a diameter of 7.88". The stress analyses and recommended material for the axle can be found in the "Maximum Stress Locations" section of this report.

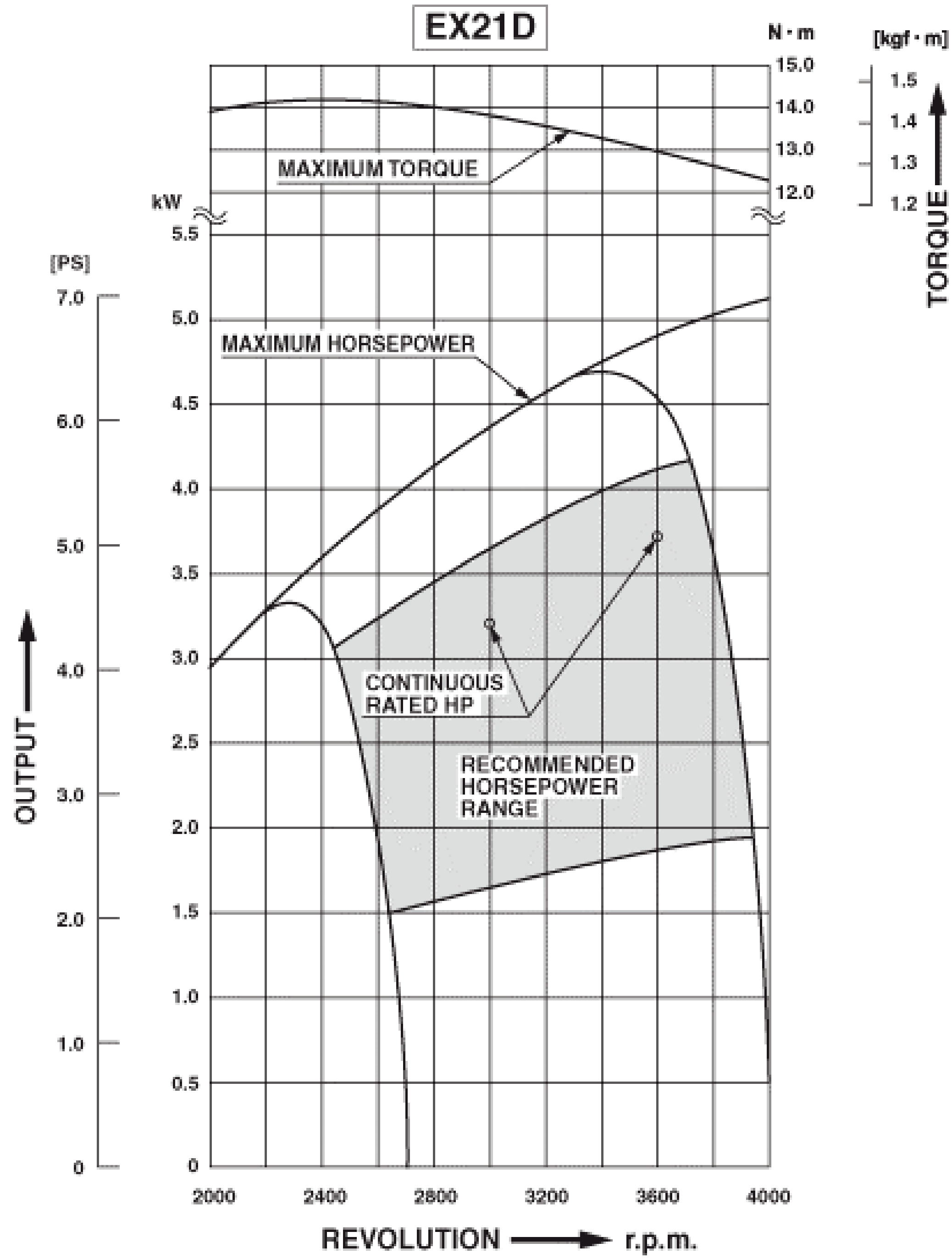
The chain is a standard #35. This chain has a life of approximately 2000 hours. Under average driving conditions, it will need to be replaced once every two years. Because this chain is widely available and very affordable, this life span should be acceptable.

The combination of engine, clutch, tires, and axle sprocket yields a drive ratio of 5.5 to 1 and a top speed of 32 mph. If it is expected that the vehicle will be used to climb exceptionally steep hills regularly, the clutch should be replaced by a torque converter and the engine position shifted to match.

Given Data:

Motor Data*:

Robin-Subaru EX Series (Overhead Cam Engine)
Model EX21
7.0 H.P. (5.1 KW)
Horizontal Shaft
Single Cylinder
Displacement 211 cc
Four Cycle Gasoline
Max H.P. 7.0 @ 4000 RPM
Max KW 5.1 @ 4000 RPM
Recommended Continuous Operation 5.0 H.P. @ 3600 RPM
Recommended Continuous Operation 3.7 KW @ 3600 RPM
Max Torque 10.26 ft.lbs. @ 2500 RPM
Max Nm 13.9 @ 2500 RPM
3/4" Shaft Diameter
3/16" Keyway



*Motor data can be found at:
http://www.americanpowerequipment.com/site/500464/page/117636 and
http://www.robinamerica.com/

Clutch Data:

Manufacturer : Comet Industries
CSC 400 Series
4" Diameter Housing Stamped
3/4" Bore
Rated Up to 8 H.P.
Low/ Cost Maintenance
Bi Directional
12 Tooth

Rear Tires:

15 x 6.00-6
15" Outer diameter
6" Width

Axle:

Solid
Posi
Sprocket Driven
30 mm Diameter

Desired Speed:

28-32 MPH

Chain Size:

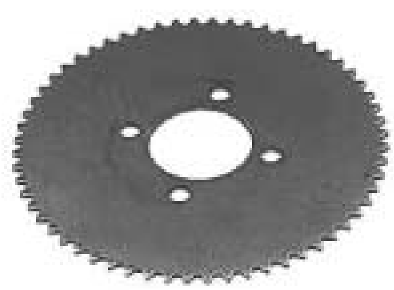
No. 35

Calculations: *

Worst Case Scenario for the Drive (Motor @ 4000 RPM)

Drive Ratio:

12 Tooth Clutch
Driven Sprocket Selected - 66 Tooth
Ratio: 66 Teeth / 12 Teeth = 1 to 5.5



*Chain, sprocket, and clutch data can be found at: http://www.mfgsupply.com/m/c/4-471.html

Go Cart Axle Rotational Speed Calculations:

(1). 4000 RPM Motor x 12 Tooth Drive Clutch = (x) RPM x 66 Tooth Sprocket
(x) RPM = 727.3

(2). 4000 RPM Motor x Ratio (1) = (x) RPM x Ratio (5.5)
(x) RPM = 727.3

(3). 4000 RPM Motor x (3.8637 x .375) = (x) RPM x (21.0164 x .375)
(x) RPM = 735.4
12 Tooth Sprocket Pitch = 3.8637
66 Tooth Sprocket Pitch = 21.0164
Chain Pitch = .375
Machinery's Handbook Page 2343

15" Tire Diameter - Circumference = 47.125" or 3.9375' (Per Revolution of the Tire)

735 RPM x 60 min./1 hour = 44,100 Rev/Hour

44,100 Rev/Hour x 3.9375 Feet/Rev = 173,644 Feet/Hour

173,644 Feet/Hour x 1 Mile/5280 Feet = 32.89 Miles/Hour [MPH] (4000 Motor RPM)

For the Standard/Recommended Motor RPM of 3600 the Speed = 29.6 RPM

Drive Selection Calculations:

Input Speed - 4000 RPM
Output Speed - 735 RPM
H.P. - 7

Service Factor - S.F. -1.7 (Heavy Shock Load, Internal Combustion Engine) [Table 7-8 P.290]

Design Power - S.F. x H.P. = (1.7) x (7) = 12 H.P.

Ratio = 5.5

N_2 (Driven) = N_1 Driver x Ratio = 12 (Teeth) x 5.5 = 66 (Teeth)

$n_2 = n_1(N_1/N_2)$ = 4000 RPM (12 Teeth/66 Teeth) = 727 RPM

$D_1 = p/\sin(180°/N_1)$ = .375 in./sin(180°/12 Teeth) = 1.45 in. (Driver)

$D_2 = p/\sin(180°/N_2)$ = .375 in./sin(180°/66 Teeth) = 7.88 in. (Driven)

Center Distance - "C" - 40 Pitches (Usually between 30 and 50)

$40 \times .375$ in. = 15 in. (Theoretical)

(Actual Center Distance is 16"), so 16" = (x) Pitches x .375 = 42.7 Pitches

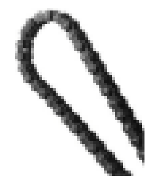
Chain Length - $L = 2C + N_2 + N_1/2 + (N_2-N_1)^2/4\pi^2C$

Chain Length - $L = 2(42.7) + 66+12/2 + (66-12)^2/4\pi^2(42.7)$

Chain Length = 126.130 Pitches

<u>Integral Number of Pitches for the Chain Length and Compute the Actual Theoretical Center Distance</u>

$C = 1/4\ [\ (L-(N_2+N_1/2)) + ((L-N_2+N_1/2)^2 - (8(N_2-N_1)^2/4\pi^2))^{1/2}\ ]$



$C = 1/4\ [\ (126 -(66+12/2)) + ((126-66+12/2)^2 - (8(66-12)^2/4\pi^2))^{1/2}\ ]$

C = 42.6 Pitches = 42.6 pitches x (.375 in.) = 16 in.

<u>Angle of Wrap</u>

Small $\theta_1 = 180° - 2\ \text{Sin}^{-1}\ [D_2-D_1/2C]$

$\theta_1 = 180° - 2\ \text{Sin}^{-1}$ [7.88 in. -1.45 in. / 2(16 in.)]

$\theta_1 = 156.8°$ P 120° Acceptable

Large $\theta_2 = 180° + 2\ \text{Sin}^{-1}\ [D_2-D_1/2C]$

$\theta_2 = 180° + 2\ \text{Sin}^{-1}$ [7.88 in. -1.45 in. / 2(16 in.)]

$\theta_2 = 203.2°$

<u>Summary</u>

Pitch of No. 35 Chain = .375 inch

Length = 126 Pitches = 126(.375 inch) = 47.25 inches

Center Distance = C = 16.0 in. (Maximum)

Sprockets = Single Strand, N0. 35 , .375 in. Pitch

Small: 12 Teeth, D = 1.45 in.

Large: 66 Teeth, D = 7.88 in.

Lubrication: The Charts recommend the use of Type "B", but this is not typically used in this industry.
It is also not practical on a Go-Kart. So we will use a Manual Lubrication System.

Chain Type/Number Selection:

An assumption was made for the chain selected. Per the selection charts, the chain we selected with the drive components, is only rated for 2.17 H.P. This rating is for an average life of 15,000 Hours per The American Chain Association. In their Design Factors Section there is a note stating " Increase in rated speeds and loads may be utilized when a service life of less than 15,000 hours is satisfactory, or when full load operation is encountered only during a portion of the required service life. It is beyond the scope of this publication to present selection procedures for all conditions. Consult chain manufacturers for assistance with these or any special application requirements."

Our Design is based on a design for 2000 hours and this how it was derived:

12 Tooth Sprocket @ 4000 RPM, Chain is rated for 2.17 H.P. for 15,000 Hours of Life

So for 7,500 Hours @ 4000 RPM, Chain H.P. = 4.34 H.P.

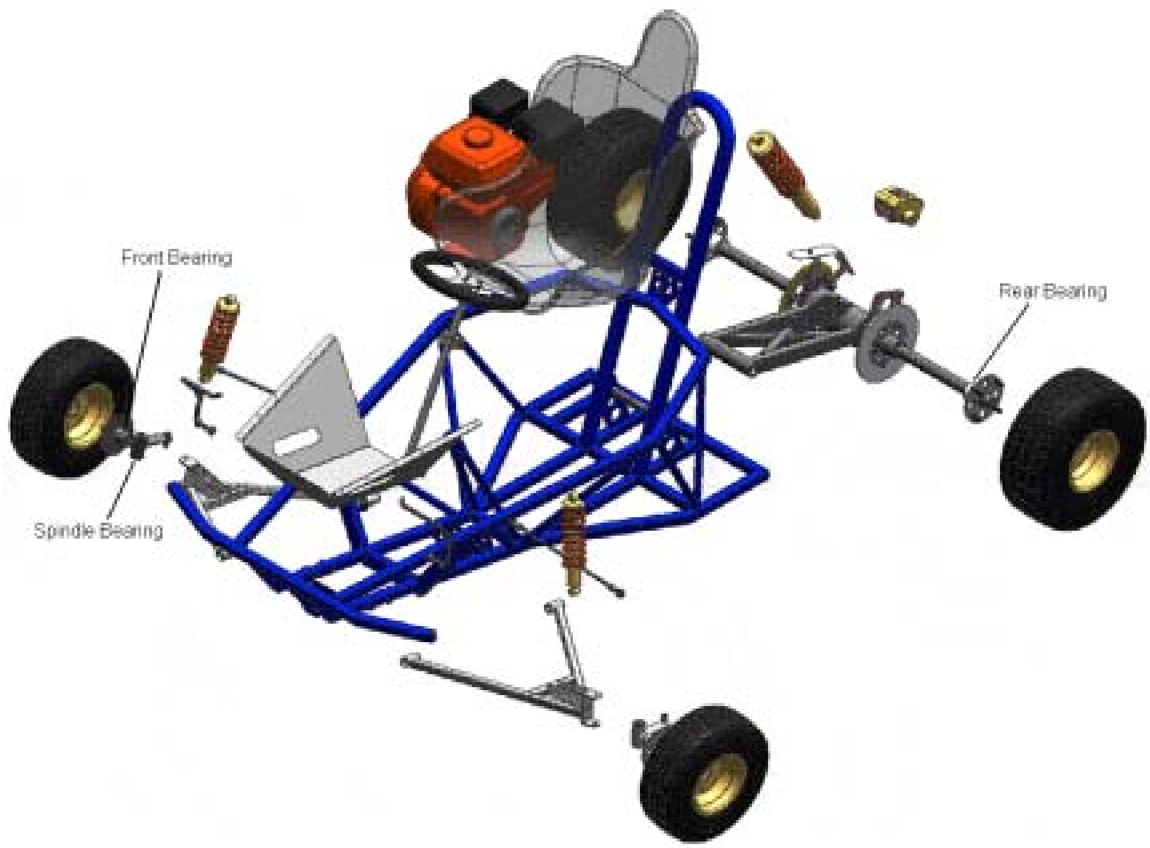
So for 3,750 Hours @ 4000 RPM, Chain H.P. = 8.68 H.P.

So for 1,875 Hours @ 4000 RPM, Chain H.P. = 17.36 H.P.

So for the Design Life of 2000 Hours @ 4000 RPM, Chain H.P. = 16.3 H.P.
(Design Power 12 H.P.)

Go Kart ridden on average of 20 Hours a week for 52 weeks a year = 1040 Hours a Year, thus having a design life of two years.

Bearing Analysis



Overview

Loads used in the calculations used to select bearings can be found in the "Load Distribution" section of this report. For dynamic side loading experienced when the go kart is in motion, it was assumed that the limiting factor was the amount of force that would cause the tires to skid under ideal conditions (coefficient of friction = 2.0). Rotational speeds of the bearings were 735 rpm at the rear axle and 850 rpm at the front axle. The difference in revolutions per minute is due to the size difference of the front and rear wheels.

Based on the data, #6206 bearings should work for the rear axle and #6203 bearing should be adequate for the front axle. The calculations used in drawing this conclusion are shown below. Although a formal analysis was not done on the spindle bearings, their application is not nearly as severe as the hub bearings. This in mind, #6000 bearings should be adequate. If premature problems develop in these bearings, they can be replaced with heavier duty bearings.

Input Data:

Rear Axle RPM =735 RPM @ 4000 RPM Motor Speed (Worst Case Scenario)
Go-Kart Speed is 32.89 MPH @ 4000 RPM Motor Speed (Worst Case Scenario)
Rear Tires are 15” O.D. Diameter

13” Tires – Circumference = 40.841” = 3.403’ / Revolution
3.403 Feet/Revolution x 1 Mile/5280 Feet = .000645 Mile/Revolution
32.89 Miles/Hour x 1Rev/.000645 Mile = 50992.25 Rev/Hour
50992.25 Rev/Hour x 1 Hour/60 Minute = 850 RPM

Front Axle RPM =850 RPM @ 4000 RPM Motor Speed (Worst Case Scenario)

Rear Bearings – 735 RPM – Inner Race Rotating and Outer Race is Stationary
Front Bearings – 850 RPM – Outer Race Rotating and Inner Race is Stationary

Radial Loads	Front -	560 lbs.	Worst Case Scenario
	Back -	690 lbs.	

An assumption was made for the axial loads. These loads only occur during the time the Go-Kart is turning or skidding on its terrain. The highest axial loads on the bearings would occur at the point where the Go-Kart is going at a high speed, turning while all wheels are on the ground, and is at it’s fastest point before it overcomes the friction between the tires and its terrain. According to the Machinery’s Handbook, the rubber can have a coefficient of friction as high as 4.0 depending what material it is riding/working on. For the worst case scenario the following calculations were made:

Coefficient of Friction = 2.0
Normal Force at Right Rear Tire = 153 lbs
Maximum Force on Tire Before Skidding = 306 lbs

For everyday use, normal driving and turning conditions, and various terrains, a friction factor of 2.0 was used and an axial load of 200 lbs. will be used for the bearing selections.

Calculations:

Rear Bearing Calculations

Radial Loads - 690 lbs.
Axial Loads - 200 lbs.
Speed - 735 RPM
Design Life of 2000 Hours (2 years)
Shaft Diameter - 30 mm

Bearing Calcs, Tables, and Bearing Selections were all completed from our School Book.

V = 1.0 (Inner Race Rotates)
X = .56 (Table 14-5)
R = 690 lbs.
Y = 1.5 Assumption
T = 200 lbs.

P = VXR +YT
P = (1.0)(.56)(690 lbs.) + (1.5)(200 lbs.)
P = 684.4 lbs.

f_N = .355 (Figure 14-12)
f_L = 1.58 (Figure 14-12)

$C = Pf_L/f_N$ C = (686.4)(1.58)/(.355) C = 3055 lbs.

(6206) Bearing [30 mm Shaft] (Table 14-3)
(6206) Bearing C_o = 2320 lbs.

T/C_o = 200 lbs./2320 lbs. = .086 (Table 14-3)
e = .281 (Table 14-5)
T/R = 200 lbs./690 lbs. = .290
T/R > e
Y = 1.54

P = (1.0)(.56)(690 lbs.) + (1.54)(200 lbs.) = 694.4 lbs.

C = (694.4)(1.58)/(.355) = 3090.6 lbs.

Bearing # 6206 "C" = 3350 lbs., which is > than calculated "C". [This Bearing is acceptable]

Front Bearing Calculations
Radial Loads - 560 lbs. (280 lbs. per Bearing)
Axial Loads - 200 lbs. (100 lbs. per Bearing)
Speed - 850 RPM
Design Life of 2000 Hours (2 years)
Shaft Diameter - 17 mm

Bearing Calcs, Tables, and Bearing Selections were all completed from our School Book.

The Front Wheels will use (2) Ball Bearings per Wheel.

V = 1.2 (Outer Race Rotates)
X = .56 (Table 14-5)
R = 280 lbs.
Y = 1.5 Assumption
T = 100 lbs.

P = VXR +YT
P = (1.2)(.56)(280 lbs.) + (1.5)(100 lbs.)
P = 338.16 lbs.

f_N = .34 (Figure 14-12)
f_L = 1.58 (Figure 14-12)

$C = Pf_L/f_N$ C = (338.16)(1.58)/(.34) C = 1571 lbs.

(6203) Bearing [17 mm Shaft] (Table 14-3)

(6203) Bearing C_o = 1010 lbs.

T/C_o = 100 lbs./1010 lbs. = .099 (Table 14-3)
e = .30 (Table 14-5)
T/R = 100 lbs./280 lbs. = .36
T/R > e
Y = 1.45

P = (1.2)(.56)(280 lbs.) + (1.45)(100 lbs.) = 333.2 lbs.

C = (333.2)(1.58)/(.34) = 1548.2 lbs.

Bearing # 6203 "C" = 1660 lbs., which is > than calculated "C". [This Bearing is acceptable]

Brake Analysis

Overview

A standard 8" rotor and matching caliper were used in the location shown above. A wide variety of pads are available. For the purposes of this analysis, a circular pad with a diameter of 2" was used. This is most likely smaller than most pads, and provides a worst-case scenario for the analysis. Loads used can be found in the "Load Distribution" section of this report.

The caliper should be steel or a metal of comparable strength due to the pressure required to provide maximum stopping force. Heat dissipation ratings for pads were difficult to find, and it is not known if the pads and rotor sourced will dissipate the heat produced adequately. Additionally, driving and braking habits have a large effect on the amount of heat produced during vehicle operation. If brake "fade" (loss of braking power) is experienced frequently, it may be necessary to use a larger pad and rotor, and/or use a slotted or drilled rotor.

The wear rating of the pads analyzed was 830 % above the optimal value for this application. This may result in short life span of the brake pads. Considering the minimal cost and service time to replace the brake pads, this should be acceptable. If replacement frequency becomes excessive, larger brake pads and rotor may be necessary.

Calculations

Strength of motor material:

Coefficient of friction between tires and ground: $\mu_t = 2.0$
Mass of vehicle: $m = 281$ kg
Normal force at right rear tire: $N_R = 682.3$ N
Normal force at left rear tire: $N_L = 597.5$ N

Force required to cause both tires to skid: $F_T = \mu (N_R + N_L) = 2560$ N
Radius of tires: $r_t = 190.5$ mm
Mean radius of rotor: $r_r = 63.5$ mm
Ratio of rotor mean radius to tire radius: $\text{Ratio} = r_t / r_r = 3.00$

Force at rotor required to make tires skid: $F_R = F_T \text{ Ratio} = 7680$ N

Radius of brake pads: $r_p = 25.4$ mm
Area of brake pads: $A = 2 \pi r_p^2 = 0.00405 \text{ m}^2$

$\text{Press} = F_T / A = 1894$ KPa

Max pressure allowed for steel rotor[7]: $\text{Press}_{max} = 2070$ KPa

Rotor material is adequate for application.

Max heat dissipation of brake:

Rotational speed of engine: $n_m = 4000$ rpm
Gear Ratio: $GR = 5.5$
Rotational speed of rotor: $n_r = n_m / GR = 727 \text{ rpm} = 12.1 \text{ rev/s}$

$P = F_R (2 \pi r_r) n = 12.4$ KW

Min braking time and distance:

Mass of vehicle: $m = 241$ Kg
Speed of vehicle: $v = 14.31$ m/s (32 mph)

Kinetic energy of vehicle: $KE = \frac{1}{2} m v^2 = 24.7$ KJ

Braking time: $t = KE / P = 2.0$ s

Deceleration: $a = v / t = 7.15$ m/s^2

Stopping distance: $s = \frac{1}{2} a t^2 = 14.31$ m

These values are under optimal conditions. Most likely the actual stopping time and distance will be much greater than the numbers seen above.

Wear rating:

Tangential speed of rotor at center of pad: $\omega = 2 \pi r_r / n = 4.83$ m/s

Frictional power rating: $P_f = F_R \omega = 37.1$ W

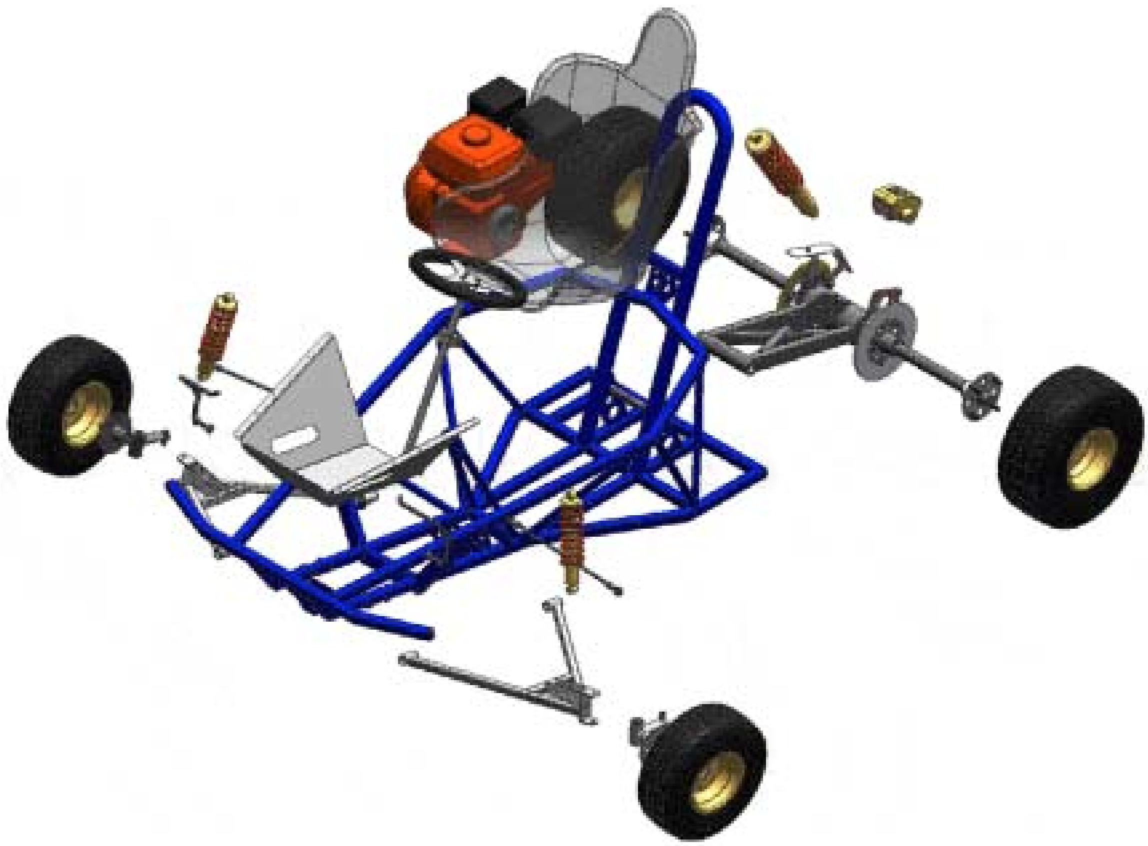
Wear rating: $WR = P_f / A = 9.15E6$ W / m^2

Max wear rating for application: $WR_d = 4.41E5$ W / m

$WR > WR_d$ by 830%

The pads may wear out quickly. Evaluate at testing and use larger pads and/or rotor if necessary.

Bill of Material



Tubing (ASTM A500, A511, or A513 Grade A)[6]

Part #	Description	Qty	Cost Each	Total Cost
1	Tube - Front Bumper, 25x25x1.6x918	1	$5.39	$5.39
2	Tube - Base Rail, 25x25x1.6x2816	2	$16.54	$33.08
3	Tube - Top Rail, 25x25x1.6x3036	2	$17.83	$35.66
4	Tube - Bulkhead LH, 38x25x1.6x295	1	$2.33	$2.33
5	Tube - Bulkhead Base, 38x25x1.6x220	1	$1.74	$1.74
6	Tube - Bulkhead RH, 38x25x1.6x220	1	$1.74	$1.74
7	Tube - Mid Cross, 25x25x1.6x310	2	$1.82	$3.64
8	Tube - Motor Hoop, 25x25x1.6x1954	1	$11.48	$11.48
9	Tube - Motor Cross, 25x25x1.6x356	2	$2.09	$4.18
10	Tube - Motor Mount, 25x25x1.6x192	1	$1.13	$1.13
11	Tube - Roll Bar, 38x1.6x2010 Round	1	$35.13	$35.13
12	Tube - Roll Bar Cross, 25x25x1.6x152	1	$0.89	$0.89
13	Tube - Steering Shaft, 19x1.6x530 Round	1	$5.38	$5.38
14	Tube - Steering Strut LH, 19x1.6x385 Round	1	$3.91	$3.91
15	Tube - Steering Strut RH, 19x1.6x395 Round	1	$4.01	$4.01
16	Tube - Swingarm Axle, 89x1.6x200 Round	1	$8.16	$8.16
17	Tube - Swingarm Diagonal, 25x25x1.6x652	1	$3.83	$3.83

18	Tube - Swingarm, 38x1.6x173 Round	1	$3.02	$3.02
19	Tube - Swingarm Split Diag, 25x25x1.6x314	2	$1.84	$3.68
20	Tube - Swingarm Arm, 38x25x1.6x726	2	$5.73	$11.46
21	Tube - Side, 19x1.6x3000 Round	7	$30.48	$213.36
22	Tube - Wishbone Cross, 19x19x1.6x326	2	$1.43	$2.86
23	Tube - Wishbone, 19x19x1.6x1852	4	$8.13	$32.52
24	Tube - Wishbone Pivot, 25x1.6x1080 Round	2	$12.58	$25.16

Angle (ASTM A500 Grade A)[5]

Part #	Description	Qty	Cost Each	Total Cost
25	Angle - Spindle Bracket, 40x40x5x150	4	$3.73	$14.92
26	Angle - Cable Bracket, 30x30x3x2816	2	$19.80	$39.60
27	Angle - Throttle Stop, 40x40x5x60	1	$1.49	$1.49
28	Angle - Master Cyl Mount, 30x30x3x295	1	$2.07	$2.07

Machined Parts[6]

Part #	Description	Qty	Cost Each	Total Cost
29	Front Hub Tube	2	$1.72	$3.44
30	Spindle Stub Axle	2	$2.46	$4.92
31	Rear Axle	1	$19.78	$19.78
32	Pedal Tube	6	$0.67	$4.02
33	Steering Block	2	$1.18	$2.36
34	Spindle Tube	2	$1.73	$3.46
35	Universal Hub	3	$3.71	$11.13
36	Brake Pedal	1	$2.80	$2.80
37	Accelerator Pedal	1	$2.80	$2.80
38	Swingarm Bushing	2	$1.11	$2.22
39	Swingarm Pivot	1	$2.31	$2.31
40	Wishbone Bushing	8	$0.29	$2.32
41	Wishbone Pivot	4	$0.76	$3.04

Flat Parts (AISI 1020 Cold Rolled)[5]

Part #	Description	Qty	Cost Each	Total Cost
42	Bearing Mount	1	$4.92	$4.92
43	Bearing Retainer	4	$1.65	$6.60
44	Brake Mount	1	$7.87	$7.87
45	Chain Guard Bracket	2	$0.48	$0.96
46	Chain Guard	1	$1.21	$1.21
47	Pitman Arm	1	$6.12	$6.12
48	Rim/Brake Plate	3	$6.89	$20.67
49	Steering Wheel Plate	1	$15.00	$15.00
50	Steering Wheel Mount	1	$3.23	$3.23
51	Foot Tray	1	$28.45	$28.45
52	Front Hub Plate	2	$6.74	$13.48
53	Shock Mount (A)	2	$6.23	$12.46
54	Shock Mount (B)	2	$6.23	$12.46
55	Spindle Arm	2	$4.87	$9.74

56	Steering Tab LH	1	$1.86	$1.86
57	Steering Tab RH	1	$1.86	$1.86
58	Swing Adjustor	2	$1.65	$3.30
59	Swingarm Mount	2	$1.56	$3.12
60	Swingarm Shock Mount	2	$2.03	$4.06
61	Sprocket Plate	1	$6.52	$6.52
62	Wishbone Mount	4	$3.48	$13.92
63	Wishbone Shock Mount	2	$2.99	$5.98

Purchased Assemblies

Part #	Description	Qty	Cost Each	Total Cost
64	Engine	1	$500.00	$500.00
65	Rim	4	$13.60	$54.40
66	Tire - Rear	2	$15.00	$30.00
67	Tire - Front	2	$16.95	$33.90
68	Shock - Rear	1	$30.00	$30.00
69	Shock - Front	2	$20.00	$40.00
70	Clutch - Centrifugal	1	$25.00	$25.00
71	Chain	1	$15.00	$15.00
72	Sprocket - Axle	1	$15.00	$15.00
73	Bearing - Insert, Rear Axle	2	$8.75	$17.50
74	Bearing - Front Hub	4	$2.00	$8.00
75	Bearing - Front Spindle	4	$2.00	$8.00
76	Brake - Caliper & Rotor, 8"	1	$59.95	$59.95
			Total Cost:	**$1,596.96**

(Comment from Don- There are some serious errors highlighted in red above- please do not take these prices too seriously)

References

1: Machine Elements in Mechanical Design by Robert L. Mott 4'th Ed. (Table 8-1 pg 317)

2: Applied Strength of Materials by Robert L. Mott 4'th Ed. (Appendix A-1 pg 632)

3: Applied Strength of Materials by Robert L. Mott 4'th Ed. (Equation 8-4 pg 318)

4: Applied Strength of Materials by Robert L. Mott 4'th Ed. (Equation 8-5 pg 318)

5: McMaster-Carr Catalog #107. Values have been adjusted to estimate cost of sizes not listed.

6: MSC Industrial Supply Co. Catalog 2001/2002. Values have been adjusted to estimate cost of sizes not listed.

7: Machine Elements in Mechanical Design by Robert L. Mott 4'th Ed. (Table 22-2 pg 850)