Mechanical Engineering Design 1

Final Project

2004 Ninja 250 Rebuild/Analysis

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Introduction | 1

This report will give a brief synopsis of the rebuild process of a 2004 Kawasaki Ninja 250 motorcycle. Furthermore, it will perform a power analysis of the Ninja 250 engine and compare the results to two other motorcycles (Yamaha R6 & Yamaha R1). The theoretical torque and horsepower numbers will be derived and will be used to calculate the forces on the front and rear sprockets of the Ninja 250. An analysis of the brake system of the Ninja 250 will be performed and used to calculate the net friction force at the brake rotors and to find an estimated stopping distance. A von-mises stress analysis of the swing arm will be performed in order to show areas of high stress concentrations/-concern. An analysis of the chain chain tensioner will be performed and the applied force on the cam chain through the tensioner spring system will be calculated.

1.1 Horsepower and Torque Calculations

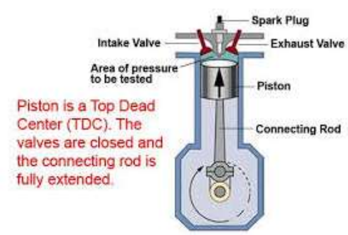

Figure 1.1: A piston at top dead center. The area being pressurized is the compressed at a certain compression ratio with respect to the outside air. [1]

Every engine is made out of piston, connecting rod, and a crankshaft. The area of the piston head is known as cylinder bore. This is because the cylinder needs to be "bored out" in order to fit the piston. The length that the piston travels is known as the stroke length of the piston. When the piston is at top dead center, right before the spark plug fires, the gas inside the piston chamber is compressed in relation with the outside air at a certain compression ratio. From these values along with a few other distinct properties of the engine, the horsepower output of the engine can be approximated.

From this theoretical horsepower output, a torque value can be produced. Using the revolutions per minute (RPM) of the engine and the output power, the torque at the engine can be calculated. The gear ratios of the engine output shaft to the transmission input shaft, the transmission gears, the transmission output sprocket and the wheel sprocket, and the wheel sprocket to the wheel can all be approximated into one gear train value. In theory, at 100% efficiency, this torque can be "carried" through the gear train to find the torque at the rear wheel. Dividing this torque by the radius of the rear wheel will result in the force applied at the edge of the wheel in order to create said torque value.

1.2 Brake System

The Ninja 250's braking system functions using basic hydraulic pressure. The force applied at the handle is multiplied by the system's mechanical advantage ratio. This mechanical advantage comes from

the fact that the incompressible brake fluid experiences only a constant pressure. By increasing the area of the brake pistons relative to the master cylinder area, the applied force is amplified.
This amplified force is then applied to the brake rotor which causes a frictional force opposing the motion of travel. This effectively drains kinetic energy from the system and slows down the motorcycle.

1.3 Swing Arm Stress Analysis

The Ninja 250 rear suspension design uses a coil-over shock absorber/-damper and uni-trak shock linkage system.[2] This system is standard on many motorcycles and its primary purpose is to prevent small forces generated at the rear wheel from road obstacles, such as bumps and potholes, from being transmitted to the chassis. Its secondary purpose is to maintain even contact of the rear wheel to the road surface. However, the shock absorber is very weak to any bending or torsion stress. Thus, a metal swing-arm is used to prevent these torsional and bending stresses from reaching the shock absorber. The bending and torsional stresses are loaded onto the swingarm while the axial stresses are loaded onto the shock absorber, which subsequently dampens or loads the system depending on the situation.

1.4 Cam Chain Tensioner

The Ninja 250 is a dual overhead camshaft engine design. It utilizes a self adjusting cam chain tensioner system to ensure the engine never goes off timing. This self adjusting tensioner is made up of the pieces as shown. The small diameter spring drives the push rod into the engine. Through a system of levers, this push rod applies tension onto the engine cam chain. The large diameter spring is compressed when the tensioner system is screwed into the engine and serves the purpose of preventing the push rod bearing from falling into the engine.

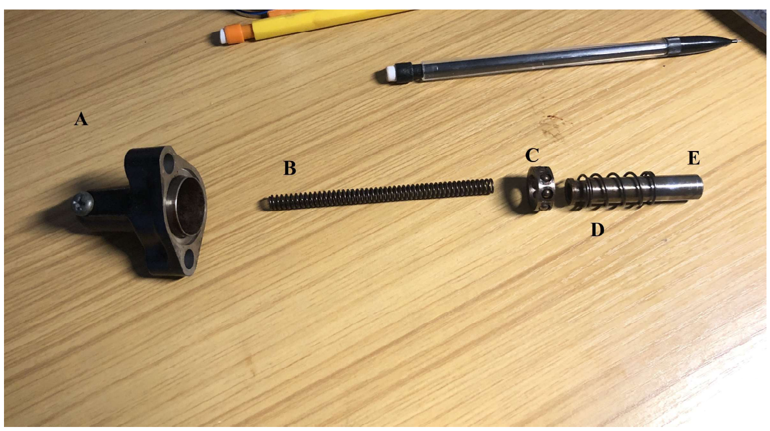

Figure 1.2: Cam chain tensioner internals: A) Upper Housing, B) Small diameter push rod spring, C) Push rod bearing, D) Large diameter bearing spring, and E) Push Rod

Procedure 2

This project consisted of a mixture of theoretical calculations using data found online from the manufacturer and data physically obtained off of the motorcycle.

2.1 Horsepower and Torque Calculations

An estimated max horsepower (P) of the engines was calculated using a relationship between the number of cylinders (n), compression ratio (ρ), bore diameter (D), stroke length (L), and firing strokes per second (N) [3].

$$P = \frac{(\pi D^2)(n)(L)(\rho)(N)}{4} \tag{2.1}$$

This was then used to calculate the torque (τ) and tangential force (W_t) on the rear wheel in conjunction with a calculation of rear wheel speed. The primary drive ratio, final drive ratio, and transmission gear ratios were all combined into a train value and used to calculate the speed of the rear wheel at max horsepower revolution per minute (RPM) (2).

$$\tau = \frac{33000P}{2\pi(RPM)} \tag{2.2}$$

$$W_t = \tau r \tag{2.3}$$

2.2 Brake System

The brake system of the Ninja 250 was analyzed as a simple hydraulic fluid system. The expansion of the rubber hoses was ignored as this change in volume was negligible. Using calipers, measurements of the length of the distributed load (L), diameter of the master cylinder piston (D_1), distance from center of distributed load to pivot point O (D), and the distance from center of the master cylinder piston to pivot point O (d) were taken. A free body diagram was created, and a relationship between the distributed load force (ω) and the force at the master cylinder piston (F_1) was established by summing the moments around pivot point O.

$$F_1 = \frac{\omega L D}{d} \tag{2.4}$$

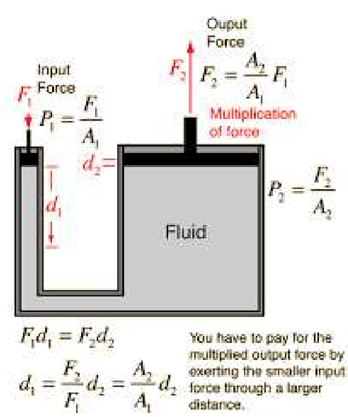

Figure 2.1: Example of hydraulic system similar to brake system [4]

The brake caliper piston diameters were estimated by measuring the piston imprints on the back of the old brake pads. The areas of both of the pistons were combined to find a total fluid area at the brake caliper

(A_2). The brake caliper piston area was then divided by the area of the master cylinder piston (A_1) to find the mechanical advantage ratio (X). This ratio was multiplied by the force applied at the master cylinder to find the force applied at the brake caliper (F_2).

$$F_2 = \frac{A_2}{A_1} F_1 \tag{2.5}$$

2.3 Swingarm Stress Analysis

Figure 2.2: The swingarm after it was removed from the bike

In order to get the dimensions for the swingarm CAD model, the swingarm needed to be removed from the motorcycle. Thus, the rear wheel and suspension system needed to be removed. With both motorcycle wheels on the ground, all relevant bolts were slightly loosened. This prevented the movement of the bike while lifted off the ground when trying to loosen these bolts. The rear axle pressed out of the inner wheel bearing ring. The brake caliper and all other rear wheel auxiliaries were removed, and the rear wheel was rolled out for under the bike. The suspension connection bolts and swing arm pivot bolts were removed and the swingarm was removed from the motorcycle. The swingarm was thoroughly cleaned, and the dimensions were taken using a ruler and caliper. The swingarm was then modeled in solid works and a von mises stress simulation was run.

2.4 Cam Chain Tensioner

The cam chain tensioner system was held into the motorcycle engine with two bolts. However, one of the bolts suffered from a clearance issue with the engine block. In order to remove this bolt, a universal joint socket adapter was necessary. The cam chain tensioner was under spring load so it shot out once the bolts are removed. Once the tensioner was out of the engine, all the internal parts were easily removed. The spring rate (K) was modeled by measuring the outer diameter (O.D.), the diameter of the spring coils (d) and counting the coils on each spring (N_a) for each spring. Both springs were assumed to be made out of HD 227 Steel and the shear modulus (G) was found to be 11.6 Mpsi [5].

$$K = \frac{d^4 G}{8D^3 N_a} \tag{2.6}$$

With the spring rate value, the calculation of max force was possible using Hooke's law. Unfortunately, the small diameter spring was found to yield before solid length was achieved. Therefore, the spring force was found at a factor of safety of 1 using the following equations.

$$S_y = 0.45 \frac{A}{d^m} \tag{2.7}$$

$$F_{max} = \frac{S_y}{k_b \frac{8d}{\pi D^3}} \tag{2.8}$$

Results 3

3.1 Horsepower and Torque Calculations

The horsepower of each of the motorcycle engines was calculated using the power equation. The RPM value that was used was the RPM of maximum power according to the manufacturer. The torque was calculated using the gear ratios of the first gear of the transmission as specified by the manufacturer.[6-8] First gear was chosen as it produces the most torque at the expense of top speed. This torque was then converted into the force applied at the wheel to the ground using the radius of the stock tire for each bike[9](Table 3.1).

Motorcycle	Power (hp)	Torque ($lbf * ft$)	Force (lbf)
Ninja 250	38.69	426.85	465.97
R6	128.88	703.18	726.62
R1	198.27	781.92	821.02

Table 3.1: These were the primary values that were used in the failure analysis of the rear sprocket. All the intermediate values can be found in the appendix.

3.2 Brake System

Information regarding the rear master cylinder was unable to be recorded physically off the bike and unable to be found online from a reliable source. Therefore, only the front brake system was modeled in this report.

The force applied by the hand (w) on the brake lever was assumed to be a uniform distributed load of 2 lbf/in. The rubber hose expansion of the brake system was ignored as this expansion was negligible and would be very hard to measure without precision tools.

Table 3.2: Intermediate values used to calculate the force applied by the brake lever onto the brake fluid. The force at the bottom of the table is the force applied onto the fluid

Name	Value
w (lbf/in)	2.00
D (in)	6.00
L (in)	4.00
d (in)	1.60
Force (lbf)	30.21

Name	Value	Name	Value
Input Force (lbf)	30.21	Master Cylinder Area (in^2)	0.196
Caliper Total Area (in^2)	1.598	Output Force (lbf)	245.9
Brake Pad Coefficient	0.35	Force on Rotor (lbf)	86.07

Table 3.3: These were the primary values that were used in the failure analysis of the rear sprocket. All the intermediate values can be found in the appendix.

The brake pad friction coefficient was found using the DOT coding on the back of the brake pads. The FE DOT code rating resulted in a minimum friction coefficient of 0.35 at temperatures between 250 F and 600 F.

3.3 Swingarm Stress Analysis

The force on the hinged portion of the swingarm was calculated using a free body diagram and a moment balance. The force of rear suspension was estimated to be 50 lbf and through a simple force balance the force at the axle end of the swingarm was calculated to be 342.7 lbf.

Table 3.4: Material properties of the alloy steel used for the von-mises stress simulation.

Name	Value
Elastic Modulus (Pa)	2.10e+11
Possion's Ratio	0.28
Shear Modulus (Pa)	7.90e+10
Yield Strength (Pa)	6.20e+8

Figure 3.1: The free body diagram of the ninja 250 weight. The centroid of the bike was estimated to be right at the engine which was 10 inches from the swingarm pivot bolt and 25 inches from the front axle.

Using the solidworks von mises stress simulation yielded a maximum stress of 6.58e+7 Pa for the loading shown. Compared to the alloy steel's yield strength of 6.20e+8 Pa resulted in a theoretical factor of safety of 9.42.

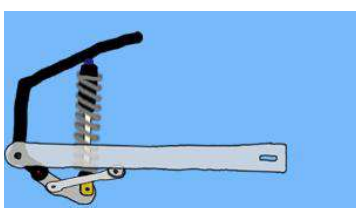

Figure 3.3: The swingarm range of motion [2]

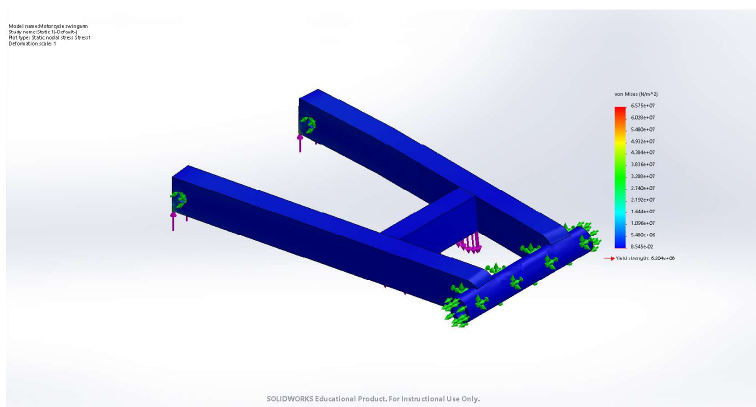

Figure 3.2: The result of the swingarm von mises stress calculation

3.4 Cam Chain Tensioner

Since the cam chain tensioner is based on a system of spring forces, those spring forces needed to be calculated before any von mises stress analysis could occur. The large diameter bearing spring force was modeled as if the spring was compressed to solid length. The large diameter bearing spring rate (K_1) was calculated to be 2.39 lbf/in. The spring force (F_1) was calculated to be 1.63 lbf at a max deflection (γ_1) ($L_0 - L_s$) of 0.68 in. This resulted in a factor of safety (η_1) of 3.20 for the large diameter bearing spring. The small diameter push rod spring rate (K_2) was calculated to be 12.24 lbf/in. At a factor of safety (η_1) of 1, the deflection in the spring (γ_2) was found to be 0.97 in using equation 2.7. This resulted in a spring force (F_2) of 11.82 lbf. The F_1 is

Name	Value	Name	Value
K_1 ($\frac{lbf}{in}$)	2.39	K_2 ($\frac{lbf}{in}$)	12.24
γ_1 (in)	0.68	γ_2 (in)	0.97
F_1 (lbf)	1.63	F_2 (lbf)	11.82
η_1	3.20	η_2	1

Table 3.5: These were the primary values that were used in the failure analysis of the cam chain tensioner push rod.

the force that holds the bearing in place and prevents it from falling into the engine. The F_2 is the force that is placed on the cam chain tensioner push rod and was modeled using a SolidWorks von mises stress analysis. The highest stress in the push rod was found to be 1.49e+6 Pa at the stress concentration. The material was assumed to be the same as the swingarm and resulted in a factor of safety of 416.1.

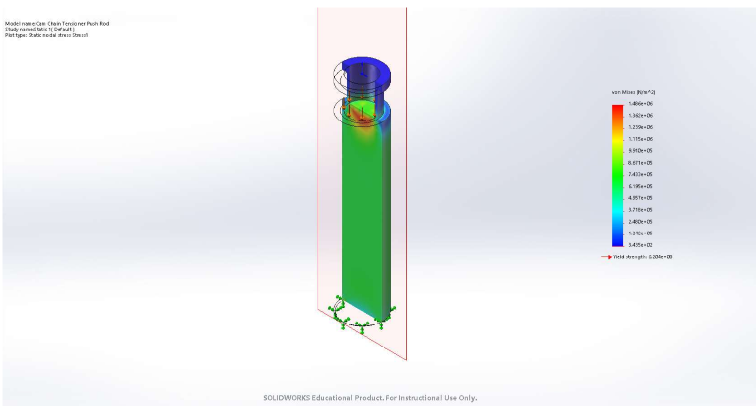

Figure 3.4: The Von Mises stress analysis of the cam chain tensioner push rod. A section cut was made down the middle of the push rod in order to reveal the area of highest stress.

Discussion 4

4.1 Horsepower and Torque Calculations

The horsepower curve of each of the engines was graphed at RPM intervals of 1 for each of the motorcycles. The resulting curves were linear for all the bikes with respect to RPM. The slope of the curve was determined by the engine configuration (Bore, Stroke, Compression ratio, etc.). Visually, real world dynamometer (a power measurement tool) power curves for each of the motorcycles appear to show a linear relationship between power and RPM. However, without discrete dynamometer data points, a percent error between theoretical and actual power curves is impossible.

Table 4.1: Max theoretical power figures for each motorcycle.

Motorcycle	Max Power (hp)
Ninja 250	38.69
R6	128.88
R1	197.27

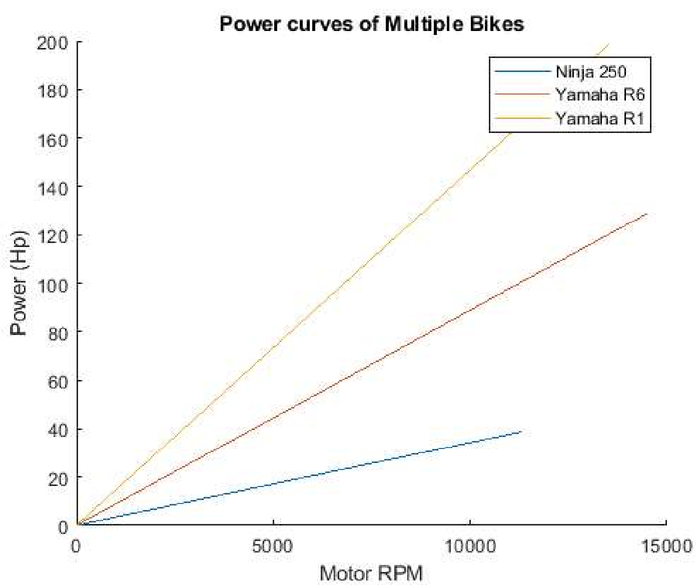

Figure 4.1: The horsepower curve of each of the three bikes. Notice that the engines will achieve higher RPMs then shown but the linear relationship breaks-down once max power is reached.

This is fairly intuitive as, in theory, none of the other variables change once the engine is made. However, over time the compression ratio of engines is known to decrease which would result in a loss of power over time.

4.2 Brake System

Using the brake force calculated on the rotor, the negative acceleration due to braking at full force can be calculated. The friction force as a result of the weight of the bike plus rider being moved was added together with the brake force.

Table 4.2: Max theoretical power figures for each motorcycle.

Force	Magnitude (lbf)
Braking Force	86.07
Weight Force	373.86
Total Force	459.93

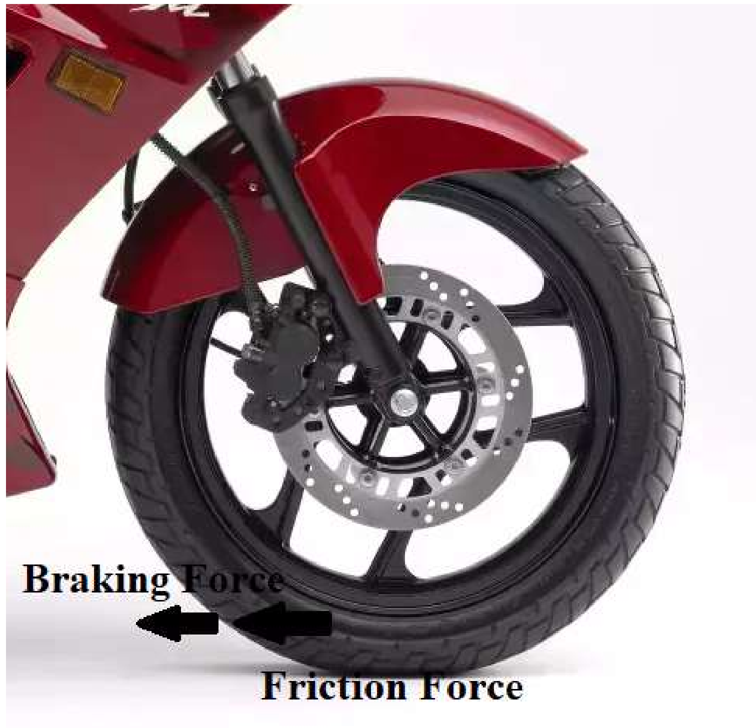

Figure 4.2: The free body diagram of the front wheel. The friction force and braking force can be added together as they act in the same direction on the same line.

Using Newton's second law of motion, the total force can be divided by the total mass of the motorcycle with the rider (549.8 lbf) to get the net deceleration. This net deceleration was found to be -26.93 $\frac{ft}{s}$.

Note

This brake calculation mathematically assumes that about $\frac{3}{4}$ of the braking force comes from the weight of the motorcycle. However, this is not true because when the clutch is pulled in and the brake is not applied, the motorcycle is not slowing down at 21.86 $\frac{ft}{s}$. This error is most likely caused by the assumption of a static system (Ignoring dynamic forces).

4.3 Swingarm Stress Analysis

The swingarm had a factor of safety estimated to be 9.42. However, the swingarm was only simulated under a static loading condition when the true purpose of the swingarm rigidty is to absorb impacts from the road. This would cause the swingarm to have a factor of safety much lower than what was calculated.

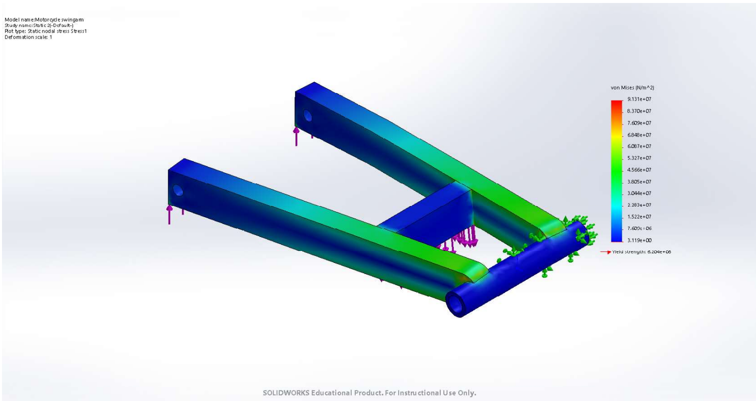

Figure 4.3: The Von Mises Stress Simulation w/o Rear Axle Support. Notice the general increase in stress everywhere in the swingarm.

Furthermore, the swingarm was clearly designed very optimally as removing any of the fixtures from the simulation caused a noteworthy change in the internal stress of the swingarm. For example, removing the rear axle support from the simulation increased the max stress to 9.131e+7 Pa. The average stress was also substantially increased as shown in the simulation results.

4.4 Cam Chain Tensioner

The cam chain tensioner push rods had an extremely high factor of safety of 416.1. This could either mean that the material that was selected was wrong, the dynamic forces and fatigue play a much larger role then anticipated (The simulation only modeled static forces.), or that the factor of safety is abnormally high since it is an engine part.

Appendix A

A.1 Horsepower and Torque Calculations

Ninja 250

Figure A.1: A picture of the 2004 Ninja 250 that was used in this report.

Transmission Gear Ratios

Table A.1: Gear ratios of all gears in transmission along with primary and final drive

Gear	Drive Gear N	Driven Gear N	Train Value
1	15	39	0.38
2	19	34	0.56
3	22	32	0.69
4	25	29	0.86
5	27	27	1.00
6	28	25	1.12
Primary Drive	25	71	0.35
Final Drive	14	45	0.31

Tire Specifications

Table A.2: Diameter, circumference, and circumference in miles of stock Ninja 250 Tire

Name	Value
Diameter (in)	26.2
Circumference (in)	82.31
Circumference ($miles$)	0.001299

Power Estimation

Table A.3: Intermediate Values of the Power Estimation of Ninja 250

Name	Value
Bore (m)	0.062
Stoke (in)	0.0412
# of Cylinders	2
Pressure	1231706.7
RPM	11300
Power ($Watts$)	28853.9
Power (hp)	38.69

Wheel Speed and Torque Calculations

Table A.4: Intermediate Values of the Wheel Speed and Torque of Ninja 250

Gear	Tire RPM	Speed (mph)	Torque ($lbf * ft$)	Force (lbf)
1	476.1	37.11	426.8	465.9
2	691.8	53.92	293.78	320.7
3	851.04	66.33	239.0	260.7
4	1067.1	83.18	190.4	207.9
5	1237.9	96.49	164.2	179.2
6	1386.4	108.1	146.6	160.0

A.2 Cam Chain Tensioner

Large Diameter Bearing Spring

Table A.5: Intermediate Values of the Large Diameter Bearing Spring

Name	Value	Name	Value
N_a	4.00	A	140
N_t	6.00	m	0.19
G	1.16e+7	k ($\frac{lbf}{in}$)	2.39
D (in)	0.98	C	19.64
d (in)	0.05	L_0 (in)	1.03
L_s (in)	0.35	F_{max} (lbf)	1.63
K_b	1.07	τ_{max} ($Mpsi$)	34.8
S_{ult} ($Mpsi$)	247.36	S_y ($Mpsi$)	111.3
η_1		3.20	

Small Diameter Push Rod Spring

Name	Value	Name	Value
N_a	46.00	A	140
N_t	48.00	m	0.19
G	1.16e+7	k $(\frac{lbf}{in})$	12.24
D (in)	0.188	C	19.64
d (in)	0.04	L_0 (in)	3.35
L_s (in)	1.96	F_{max} (lbf)	17.02
K_b	1.32	τ_{max} $(Mpsi)$	167.2
S_{ult} $(Mpsi)$	258.1	S_y $(Mpsi)$	116.1
η_1		0.694	

Table A.6: Intermediate Values of the Large Diameter Bearing Spring. Note that this table assumes the spring is compressed to solid length. The factor of safety was set to one and the force needed was solved for after these calculations were done.

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