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Final Project Agreement

Team# 12

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Table of Contents

1. Summary

At the conclusion of this design project, a three wheel repair vehicle was successfully designed to meet the required performance specifications as given by the customer including the ability to carry an operator and a 200 pound load, travel up a 20 degree incline at a minimum speed of 7.6 miles per hour, and possess necessary life for all non-maintenance components. The proposed design has absolute dimensions of an 8' 10" length, 4' 3" height, a width of 3' 9", and an unloaded weight of 550 pounds. Powering the vehicle is a 10 horsepower gas Briggs and Stratton 1150 Horizontal OHV engine which will be sufficient to meet the speed and slope requirements. The transmission system used supports forward and reverse motion through the tensioning of two separate double V-belt pulley systems with a linkage system. To achieve the appropriate angular velocity in the drive shaft, two 4:1 belt reductions were employed. The design features two 14 inch front wheels that will are connected to a rack and pinion steering system. The 16 inch rear wheel is attached to the drive shaft and serves as the primary source of forward and reverse motion for the vehicle. A mechanically activated go cart styled brake disk kit is used in order to stop the repair vehicle from its cruising speed of 11.45 miles per hour to rest in ten feet. In order to accommodate a variety of loading applications, a 9 square foot loading bed made of 12 gage sheet metal has been mounted to the rear end of the vehicle at a load height of roughly 3 feet. In addition to standard designing factors of safety, miscellaneous safety considerations were accounted for including belt guards around the transmission housing in case of belt slippage or failure and an upgraded seat with a seatbelt.

2. List of Assumptions

Throughout the course of this design project, a number of assumptions were made according to the various subsystem applications and design requirements. These assumptions are outlined below by subsystem along with a brief explanation of where uncertainty stemmed from or why the assumption was considered to be valid.

2.1 Belt Length and Tension Assumptions

- The largest assumption that was made in this portion of the project was that although double V-belts were used, the tension in the belts could be accurately modelled using the flat belt equations throughout Section 17-2 from the Tenth Edition of Shigley's Mechanical Engineering Design (unless otherwise specified, all table, figure, and equation references are taken from Shigley's Mechanical Engineering Design).
    - Originally the V-belt equations throughout Section 17-3 were attempted in order to model the double V-belts used in the transmission system. The pitch length equation was successfully used in order to determine the overall lengths of the double V-belts. Then, the allowable power per belt equation (H_a) yielded extremely low allowable horsepower per belt. This low value then trickled through other equations to result in very large N_b (number of belts) and ΔF (change in force between the tight and loose sides of the belt) values (for example, ΔF = 14,443 lb in a sample calculation between two 12 inch pulleys). As a culmination of the effect of the H_a term, the primary forces ($F_1$and F_2) were clearly in error through simple logical analysis. However, it was noted that tension equations that were not dependent on H_a such as the centrifugal tension (F_c) did result in values that were extremely similar to values obtained from corresponding flat belt equations. In addition, the length equations yielded

    almost identical results. Combining these last two observations, it was decided that modelling the double V-belts as flat belts would be an acceptably accurate representation of their tensions.

- The geometry for the transmission system belting was modelled off of Figure 17-1(a) and Figure 17-2(c). For the reverse operation system (17-2(c)) the geometric spacing of the proposed transmission system resulted in a positive angle off of the driving engine pulley as opposed to the negative angle shown in the figure. This angle direction discrepancy was assumed to be acceptable.
    - This assumption stemmed from an uncertainty due to lack of information in the text concerning this multi-pulley reverse operation layout. As neither the angle off of the driving pulley nor specific tension equations were listed, it was deemed acceptable that this assumption of similarity be made.
- In order to estimate the length of the reverse operation pulley system, the geometric distance was calculated using Inventor.
    - This assumption of similarity between the physical geometric length and the calculated lengths was established due to an observed pattern in the flat belt equations where this conclusion was seen to be a decent approximation.
- As seen in Figures 17-12 and 17-14, the tension in both flat and V-belts varies throughout the belt. However, in order to simplify the calculations for the belt tensions, it was assumed that other than at the tangent points on the pulleys (tight and loose) the tension in the belt was the same. This assumption was particularly useful in relating the tensions at similar tangent points in opposite pulleys, although, if this assumption is incorrect, it would have affected the moment diagrams on either the reduction or drive shafts.
- In order to estimate the tension in the double V-belt on the reverse operation pulley system, the system was modelled as a 12” to 12” pulley and a 3” to 6” pulley rather than as a 3” to 3” to 3” to 12” system. This assumption allowed the tensions in representative portions of the reverse operation pulley system to be calculated using the standard flat belt equations.
    - By modelling the reverse operation pulley system in this manner, the geometric lengths as calculated in Inventor were almost identical to the actual distances. This confirmed that the model was at least geometrically representative of the actual system. In addition, the calculated tensions from the model were of the same magnitude as tension calculations for some of the other double V-belts in the transmission system. This built confidence in the assumption that the model was a reasonable estimate of the tensions throughout the reverse operation pulley system (and thus of the forces acting on the various pulleys).
- It was assumed that 100% power transmission was achieved throughout the belt systems.
    - This assumption is clearly not an accurate model of a real world application although it was accepted in order to simplify the design process. In a future design, a higher horsepower engine would be selected in order to compensate for power lost in the belt transmission.

2.2 Braking Assumptions

- The largest assumption in the braking portion of the project was the desired distance that the vehicle would be able to stop in. This distance was set assuming constant contact throughout the braking process (no skidding) and then back checked for logic against the relative acceleration the driver would experience (in g’s).

- The added stresses due to heating up of the brakes during operation were assumed to be negligible for the scope of this design.
- The life of the brakes was assumed to be sufficient as the brake pads were considered to be replaceable parts.

2.3 Shaft Design Assumptions

- When calculating the endurance strength for both the reduction and drive shafts, the temperature modification factor and miscellaneous effects factor were assumed to be negligible and therefore equal to one. This assumption was commonly made throughout the course material as well as seemed reasonable within the defined application.
- After determining the first calculated diameter on both the reduction and drive shafts, the remaining diameters were calculated using the standard D/d ratio of 1.2 and then rounding to a standard size (page 364).
- As previous mentioned, there were a number of assumptions that went into the calculation of the belt tensions within the transmission system. It was assumed at the outset of the shaft design calculations that the belt tensions and forces on each pulley were accurate (meaning no additional safety factor was added to account for a previous uncertainty).
- As the reduction and drive shafts were designed before the entire design was completed, the weight used for the calculations was assumed to be the max supportable weight of the tires (1200 pounds) with an even weight distribution on each tire. Although the final unloaded design was well under this maximum value, this assumption allowed for the calculation of a worst case scenario design that was acceptable regardless of loading.
- When designing each shaft axial loading was considered to be negligible.

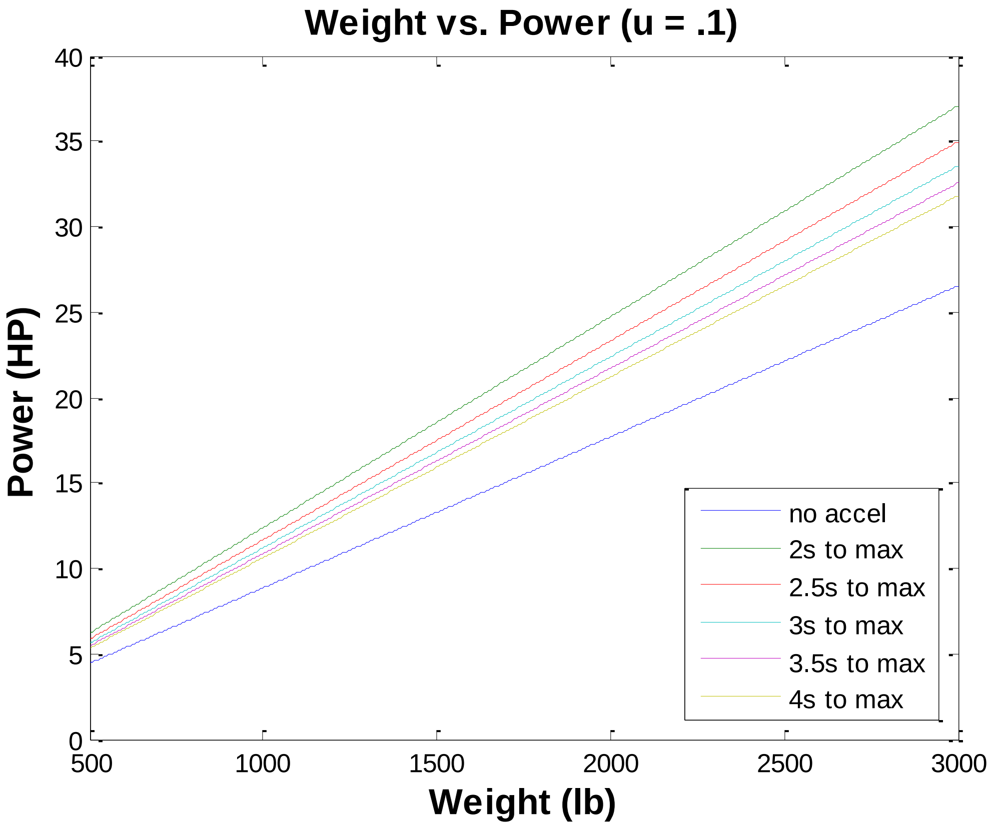
2.4 Steering Assumptions

- It was assumed that the bearings used to support the steering shaft as well as the steering components linking the rack to the tires will have an infinite life due to the small forces that they experience during routine operation.
- In order to simplify design considerations for the front end of the vehicle's frame it was assumed that the channel supporting the rack will be well greased and that the rack and pinion gear will be well lubricated and free of environmental contaminations.
- The torque that was applied to the steering wheel was assumed by considering a reasonable turning force to be 30lbf being applied on the 6.5" radius steering wheel.
- A maximum pinion speed was assumed to be 2 revolutions per second by attempting to visualize how quickly an operator could and would practically turn a steering wheel in routine operation.

3. Calculations and Analysis

3.1 General Motion Calculations

One of the first design features that had to be considered with this project was the required power needed to achieve the minimum 7.6 miles per hour motion up a slope of 20 degrees. This process was modelled with two sets of free body diagrams (one with no acceleration and one with acceleeration) showing the forces on a block (representing the repair vehicle) sliding up a 20 degree slope. The main forces that appeared in each free body diagram were the weight of the block, a normal force from the ground acting on the block, a friction force resisting the motion of the block, and an explicitly applied force representing the driving force from the engine. Each free body diagram was evaluated along the direction of the ramp using Newton's Second Law in order to solve for the explicitly applied driving force (in terms of weight and coefficient of friction). The power required was expressed as the function power equals the applied force times velocity. The power relationship combined with the expressions for the engine force were then plotted in MATLAB against a varying weight vector in order to provide a framework for the required horsepower now as a function solely based on the weight of the repair vehicle (note that the coefficient of friction was assumed to be 0.1). The initial estimate of the weight for the vehicle was 1500lbs, indicating a necessary power of between 12 and 17 horsepower. Reccomendations from the instructor of this course during update presentations hinted at a likely power of approximately 10 horsepower so the assumption for the expected weight was revised to 1000lbs. After completing the proposed design, the unloaded weight of 550lbs is well within the 10 horsepower engine's capabilities. In addition, even at the maximum possible loading of 1200lbs the 10 horsepower engine would just be beginning to strain to reach the required performance expectations.



Other general calculations that were completed at the beginning of this design project were the anticipated revolution speeds of the various pulleys in the transmission system. It was known that the 10 horsepower gas Briggs and Stratton 1150 Horizontal OHV engine had an output shaft that revolved at 3800 revolutions per minute. Estimating the rear drive wheel to be 14" in diameter, it was calculated using the kinematic concept of tangent velocities ($\omega_a r_a = \omega_b r_b$) that the drive shaft would need to rotate at 237.5 revolutions per minute. In order to reach this rotation, two 4:1 belt reductions were laid out (corresponding to revolutions per minute of 3800rpm → 950rpm and 950rpm → 237.5rpm). As the final selected rear wheel was 16" rather than 14", the final velocity of the vehicle was 11.4 miles per hour (slightly faster than calculated).

3.2 Belt Tension Calculations

Calculations were conducted on the various double V-belts within the transmission system to determine the overall forces acting on each of the pulleys. This information was later critical in the shaft designs for the transmission system. For the calculations, the double V-belts were modelled as flat belts and analyzed using the equation sets provided in Chapter 17 Section 2. As a general process, each belt tension calculation began by defining the horizontal distance between the centers of the pulleys (C), the larger diameter pulley (D), and the smaller diameter pulley (d). These values were then used to calculate the overall length of the belt (L from equation 17-2) and the angle of contact for the belt on each pulley (θ_d and θ_D from equation 17-1). When modelling the length of the double V-belts as V-belts rather than flat belts, equation 17-16a was used rather than equation 17-2 (note that even in these cases the belt tension was still modelled as a flat belt). Then, the weight of the belt was determined by multiplying the volume (for the double V-belts it was two equal trapezoids) by the specific weight of the manufacturer's specified belt material (typically urethane). Next, the belt speed (V from page 877) was determined using the smaller pulley diameter (d) and the rotational speed of the smaller pulley (n). Finally, the torque was calculated by dividing the power by the rotational speed of the pulley of interest. At this point, all of the defined values were applied into the F_i, F_c, F_1, and F_2 equations (equations 17-9, (e), and Figure 17-7 respectively) in order to determine the final tensions in the belts. The F_1 term always corresponded to the tight side of the belt whereas the F_2 represented the tension in the loose side of the belt. To provide a sense of magnitude, the largest calculated tension was 522.43lbf (occurring on the 3" single pulley on the reduction shaft) and the smallest tension was 30.95lbf (occurring on the 3" double pulley on the engine output shaft). When sourcing the belts for the transmission system, sizes were overestimated in order to allow the tensioner pulley flexibility to engage either the forward or reverse operation systems.

3.3 Braking Calculations

In this application, the preffered method of stoping the vehicle was to use a simple disk brake and caliper system. This disk brake was attached to the drive shaft so that its torque would be applied direcctly against the rotation of the wheel. To calculate this torque, a general constant acceleration equation was used. As seen on (Appendix B DSC Page 2a), the equation only requires the initial and final velocities and displacements. A reasonable stopping distance of 10 feet was set for when the vehicle was going its maximum speed of 11.4 mph. It was calculated that the car would accelerate at about 0.4 g's so the driver won't experience too much of a jerk when applying the brakes.

To find the force of friction on the brake disk, Newton's 2nd Law was utilized. Because the acceleration was now known, it could be multiplied by the mass of the vehicle to get a force. This force of about 525 pounds was the force of friction the caliper that was applied to the brake disk. With this information, the final drive shaft calculation could finally be computed using the friction force and the torque it applied.

3.4 Drive Shaft Design Calculations

When designing the drive shaft, the first step was to decide exactly what components were going to attach to the shaft. The shaft's purpose is to attach the back tire to the frame and to transfer power received from the reduction shaft to the tire. The vehicle also needs to be able to stop so the braking system was decided to be attached to the drive shaft. To attach the sourced tire to the shaft, it was concluded that the tire itself couldn't contact the shaft and would need a hub on either side that actually contacted the shaft. With this information, it could be seen that the shaft would need a bearing at the ends, two hubs for the tire, a brake disk, and a pulley to receive torque. Logically, the shaft was set up to be symetrical starting at the lowest diameter on the ends for the bearings. As seen in (Appendix B DSC Page 2), the next larger diameters for the pulley and brake disk and, finally, the largest diameter in the center for the tire hubs.

To transfer torque through the different points, the pulley, hubs, and brake disk needed keys. The keys for the pulley and brake disk were placed at the center of each but the hubs had a special case. On the hubs, the key grooves only go halfway through and are used to stop them from moving towards the center. This makes it so that when the hubs are bolted together through the tire rim, they are held in one spot and cannot slide on the shaft. To hold the brake disk and pulley in place, spacers were used in between them and the bearings to press them up against the shoulder. The overall lengths of the shaft were decided having the tire and center line up with the engine. From there, the distance to the pulley could be measured from the final transmission setup. To keep symmetry, the brake disk was placed like the pulley and the bearings were placed where the shaft would meet the frame.

When the spacing and goals of the shaft components were all decided, a total list of shaft components could be created:

- Shaft
- 2 Bearings
- 2 Spacers
- Pulley with Bushing and Key
- Brake Disk with Key
- 2 Hubs each with a Key
- Total of 14 Components

The next step of the drive shaft design was to find the diameters at each section. To accomplish this the forces on the shaft had to be calculated and set up in a free body diagram. Forces acting on the shaft were pulled from belt tension calculations, braking calculations, weight considerations, and were all brought together on (Appendix B DSC Page 3a) along with torques (T_m). From this point, all forces were split up into two free body diagrams in the xy and xz planes. The xy plane was analyzed first on (Appendix B DSC Page 3a) where the shear and moment diagrams were set up and the moment was calculated for all points A through K. On (Appendix B DSC Page 3b), again the shear and moment diagrams were set up but this time for the xz plane. Moments at all points were calculated and then combined with the xy moments to get the true alternating moments (M_a) on the shaft.

After all the forces were calculated and brought together on (Appendix B DSC Page 3c), it was time to solve material properties of the shaft. To solve for the endurance limit of the shaft, AISI 1020 CD steel was used as the starting material. Table 6-2 was used to calculate the surface dondition modification factor. Equation 6-20 was used to calculate the size modification factor. Equation 6-26 was used to calculate the load modificatioin factor. Temperature was assumed to be normal room temperature so the temperature modification factor was set to one. A reliability was set for the shaft to be 99.99% so the reliability factor was taken from Table 6-5.

Miscellaneous effects were assumed to be negligible so the miscellaneous effects modification factor was set to 1.

Initial stress concentration assumptions were taken from Table 7-1 and used to get values at the shoulders in bending and torsion. These values were then used in the *DE-Goodman* criteria to solve for the diameter at a few points on the shaft (see Appendix B DSC Page 3c). From the results, it could be concluded that, without keyway stresses, the shoulder at the brake disk was the weakest point on the shaft with a diameter of 1.5243". Since the values used were conservative, Table A-17 was used to get the next lowest standard size of 1.5". This value was then used to recalculate the endurance limit and to find notch sensitivities from Figures 6-20 and 6-21. Stress concentrations were read from Figures A-15-8 and A-15-9 and used with Equation 6-32 to find the regular and shear fatigre-stress concentration factor. As seen on (Appendix B DSC Page 4), these values were then used in the *DE-Goodman* criteria again to find the new diameter which turned out to be 1.4822". The first steps of another iteration were taken and the size modification factor was found to be about the same as before so the diameter won't change again. Therefore, the closest standard size of 1.5" was used for the diameter the brake disk sits on.

From Page 364 in the textbook it is stated that a general D/d ratio is 1.2. With this assumption the other two diameters for the shaft can be calculated as 1.25" and 1.8". Using the *DE-Goodman* criteria for fatigue factor of safety and combined loading for static yield, the shoulder at the brake disk was found to have $n_f = 2.07$ and $n_y = 4.56$.

Looking at Table 7-6, the key under the brake disk and pulley (because of symmetry) was set to be 3/8" wide and ¼" tall because of the 1.5" diameter. To solve for the length of the key a saftey factor had to be set for the key to meet. The chosen safety factor was 1.5 so that it would be lower than the shaft fatigue saftey factor because it is desired that the key would fail before the shaft. A key material of UNS G10180 was assumed and used for calculations. Force on the key was the only paramater left before the length could be calculated so it was calculated by dividing the torque by the radius. As seen on (Appendix B DSC Page 6) the length needed to resist failure by shear was 0.454" and the length needed to resist failure by crushing was 0.53". Thinking about safety first, the chosen length was 0.53" so it would be safer.

For the keys under the hubs, the diameter of 1.8" meant that Table 7-6 set the width and height to 0.5". Again, UNS G10180 and a saftey factor of 1.5 were used and the force was calculated on the key. With these values, the shear failure length was 0.28" and the crush failure length was 0.33". As before, the longer length of 0.33" was chosen as the final dimension for safety reasons.

The drive shaft was calculated to run for at least $5.24 * 10^7$ cycles. To calculate the life of the shaft, Equation 6-13 was used on (Appendix B DSC Page 5). In the end, the drive shaft was calculated for failure after $7.11 * 10^7$ cycles which means that it meets the final requirement for the design.

3.5 Reduction Shaft Design Calculations

The second and only other major shaft in the transmission system is the reduction shaft that provides a four to one drop in revolutions per minute from the engine and transfers power to the drive shaft. To accomplish this, the shaft needs a 12” pulley receiving power from the engine and a 3” pulley driving the drive shaft. Again, this shaft was designed to be symmetrical with three different diameters. With this information a list of components making up the reduction shaft could be made:

- Shaft
- 2 Bearings
- 2 Spacers
- 12” Pulley with Bushing and Key
- 3” Pulley with Bushing and Key
- Total of 11 Components

Once these components were decided for the reduction shaft, the length of the shaft and the distance between pulleys were set. The only distance that had to fit into the overall design was the distance from the bearing at point A and the 12” pulley at point F (see Appendix B RSC Page 1). This dimension was important because the shaft had to fit onto the frame and reach over far enough to attach to the 12” pulley coming from the engine. Other dimensions could be set arbitrarily to make the shaft symmetrical. As seen in (Appendix B RSC Page 4), the 12” pulley is subject to larger forces when the transmission is set to make the vehicle go forward as opposed to the forces in reverse. Therefore, the reduction shaft was completely analyzed with forward forces to have the most conservative saftey factors.

As previously done on the drive shaft, a free body diagram of forces was set up and it was analyzed in both xy and xz planes for moments (see Appendix B RSC Page 2). The resulting moments were then combined on (Appendix B RSC Page 2a) and used to calculate diameters with AISI 1020 CD steel. Table 7-1 initial assumptions and the *DE-Goodman* criteria were used again to get some initial diameter iterations and find the critical point. The critical point of the reduction shaft ended up being the shoulder at the 12” pulley (point E).

When the critical point was identified and an initial diameter estimate was made, more iterations of the *DE-Goodman* criteria could be done and the final diameter under the pulley was found to be 1.3283”. Because this diameter was conservative the standard diameter used was 1.25”. A D/d ratio of 1.2 and rounding to standard sizes were used again to get the final shaft diameters of 1” and 1.5”. The calculated saftey factors at this point against fatige were $n_f = 1.79$ and against yield $n_y = 2.46$. To match the initial shoulder saftey factors of the drive shaft ($n_f = 2.07$ and $n_y = 4.56$) and anticipate additional stress in the keyways, the material was upgraded to AISI 1050 CD steel. The new saftey factors became $n_f = 2.63$ and $n_y = 3.63$ (see Appendix B RSC Page 3).

After obtaining the saftey factors for the shoulder, the stress concentration due to the key at point F was then considered and calculated like the drive shaft keys. Because the diameter was 1.25”, a ¼” square key made of UNS G10180 was used to calculate the key length when set to a 1.5 saftey factor (see Appendix B RSC Page 5). The length ended up being 0.24” and this will be the same for the 3” pulley because the shaft is symmetrical. Finally, the fatigue factor of saftey at the keyway was calculated in (Appendix B RSC Page 3a) to be $n_f = 1.63$ which is less than the shaft saftey factor so the key is designed to fail before the shaft.

The last parameter to check on the reduction shaft was the life of the part. Using the same methods as the drive shaft, the required cycles for the reduction shaft to last was calculated to be $2.1375 * 10^8$. As seen on (Appendix B RSC Page 4a) the life of the reduction shaft was calculated to be $2.5016 * 10^9$. This means that the reduction shaft meets the final requirement for the design while being generally safe.

3.6 Steering Calculations

The steering system is a critical performance and safety feature of any vehicle. A rack and pinion system was chosen over a simple 1:1 go kart system for 2 reasons: 1. The gear reduction ofers more steering wheel revolutions from lock to lock. This allows the operator to make fine adjustments to the steering wheel without drastically changing the trajectory of the vechicle offering more precise vehicle control. 2. The gear reduction reduces the effort required by the opperator to turn the wheels. This is critical on a heavy vehicle with no power assisted steering. Due to this mechanical advantage, a smaller steering wheel of 13" in diameter was fitted. A smaller steering wheel also occupies less space giving the operator more leg room and a more comfortable driving position.

The steering system consists of a shaft supported by 2 bearings with the steering wheel bolted at one end of the shaft and the pinon gear welded on the other. The rack floats in a greased metal channel welded to the frame and retained by the pinion gear. The rack and pinion are spur gears with a ¾ inch face width . Due to the small number of infrequent revolutions (assumed 2 rev/s) and light loading, the life of the rack and pinion gear were assumed to be infinite; however, the calculations did not back up this assumption.

The size of the pinion gear determined the number of revolutions reqired to turn lock to lock. A 16 tooth pinion allows the steering wheel to travel 1.2 revolutions (432 degrees) from lock to lock or 0.6 (216 degrees) revolutions from straight to lock. Lock to lock offers 76 degrees of tire travel and straight to lock offers 38 degrees of tire travel. This offered a final turning radius of 9' 2". Cafeful consideration went into sizing the pinion and determining the steering ratio. A small pinion gear would result in a large number of steering wheel revolutions from lock to lock; not required for the slow operating speed and relatively light vehicle weight. It would also increase the time needed to turn the wheels from lock to lock, a common operation for the intended vehicle application. A large pinion gear would offer very heavy and tight steering; analogous to the simple go kart steering, completely defeating the purpose of the more complicated rack and pinion steering system.

The rack and pinion steering system was calculated using the wear factor of safety (equation 14-42). This calculated wear factor of safety was 0.32 which seems very low considering the small forces involved with relatively few revolutions compared to a normal gear application of several hundred revolutions per minute. High strength components were selected for the steering system that should have provided an infinite life. The low safety factor originated from the large denominator, specifically the σ_c term. The dominating factor in the σ_c expression is C_p which was calculated with a youngs modulus value of $30*10^6$ psi. All other dominating values were sourced from specific tables and each had similar ranges that would not have significantly affected the overall factor of safety. It was decided to include the incorrect calculations to show the attempted effort for overall completeness (sample values for the calculation shown below).

Given Data (Rack and Pinion):
Face width = ¾ in / Pressure angle 20 degrees / 16 teeth on pinion / Pitch diameter 1 in

Sample values:
S_c = 150,000 Nitrided
H_b = 83.5 HR 15N
H_{BP}/H_{BG} = 1 Assuming the same material for the gear and rack
K_B = 1 Temperature is less than 250 F / K_R = 1 for 99% reliability
$Z_n = 10^6$ cycles / Q_v = 5 / K_s = 1 / C_{mc} = 1 / C_{pm} = 1

3.7 Frame Consideration

Even though the frame was not a critical part of this project, a robust, flexible and well-designed frame was desired. A lot was taken into account when designing the frame and many versions were created. The frame was a “living” design in that it was initially created to be easily modified as other components were selected and analyzed. The frame and shafts were very much dependent on each other. The shafts needed the weight and dimensions of the frame so they could be analyzed, and the frame needed to be able to support the shafts and shaft components. For the design process, a layout was created. This layout got the general shape of the frame and the general frame was formed. From this, weights were estimated using Autodesk Inventor’s material properties to calculate the mass.

The initial frame was constructed of 1” diameter 12-gauge cold drawn steel tubing. This material was chosen because it is strong, light, relatively inexpensive, and had the overall appeal of rounded shapes. As the design progressed, it became more apparent that a flat surface would be needed for the shaft bearings to be mounted to. The previous frame was redesigned using the previous dimensions, but only changing the shape and size of the tube. The new frame was made out of 2”x2” 12-gauge cold drawn steel tubing for the long bars that run the entire length of the car and 1” x 1” 12-gauge steel for the load framework built up from it. The gauge of the tubing might be excessive, so given more time, further analysis could be done to choose a more appropriate gauge. The bars would be welded together which would result in a very strong frame. Materials other than the cold drawn steel were also considered. These materials were various types of aluminum alloys. These alloys were appealing because they were lighter, but they were more expensive and weren’t as strong. In addition, aluminum can be harder to weld than carbon steels, further increasing the overall costs for labor.

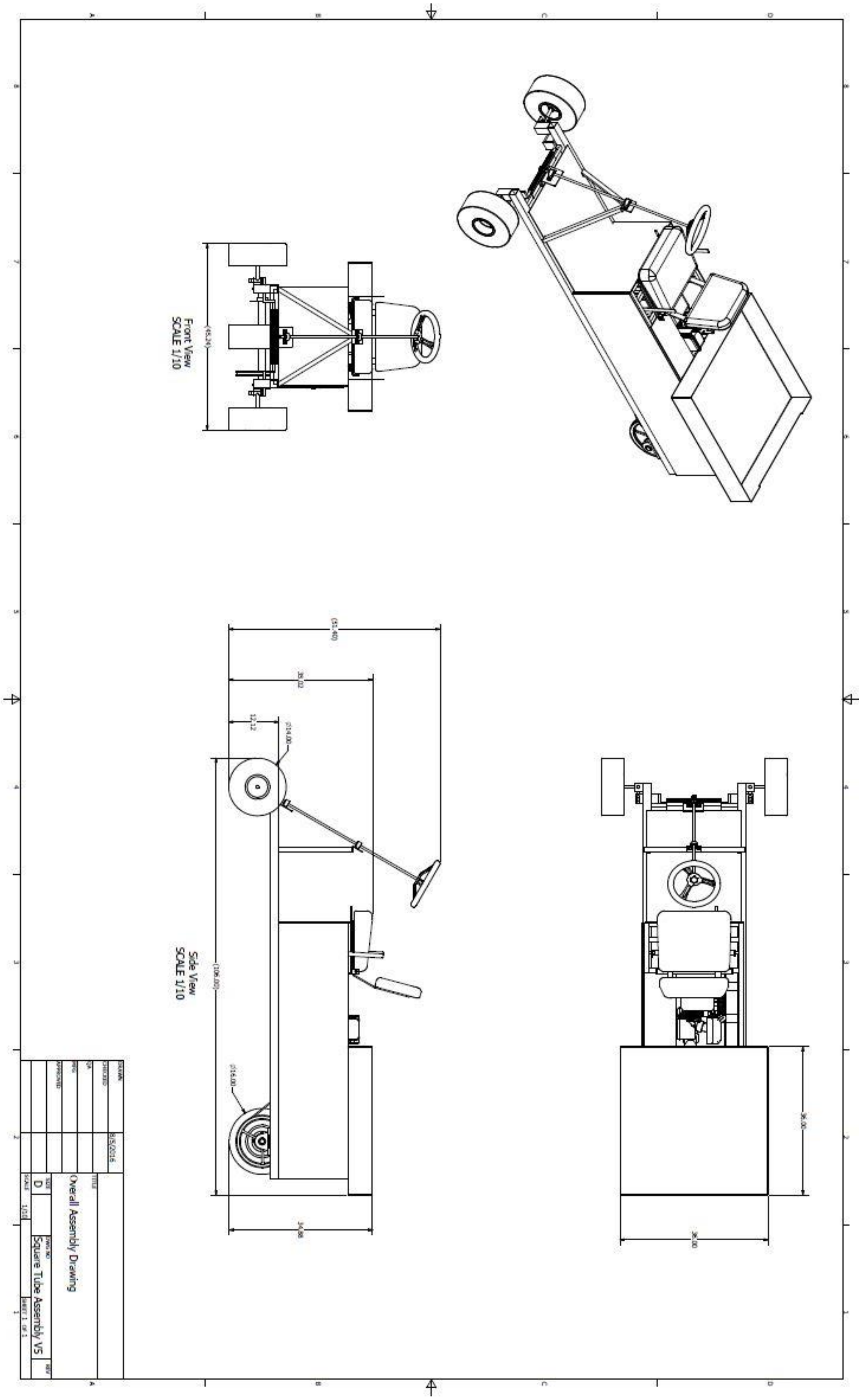
In addition to the frame changing the tubing shape as the design progressed, it became clear that the frame needed to be very open on the inside. Because the transmission system designed used belts, cross braces on the inside were limited to at the tops and bottoms. If cross braces were included at other points, it could interfere with the belt operation or make it impractical/impossible to change the belts for maintenance purposes because a steel tube would be permanently the center of the belt. In addition, the transmission and the brake also required clearance so the levers and connecting linkages could move freely without interfering with anything, but still needed to be supported at the point of rotations at the levers. The supports seen inside of the frame are used to mount the engine, reducing shaft, the brake lever and the transmission lever.

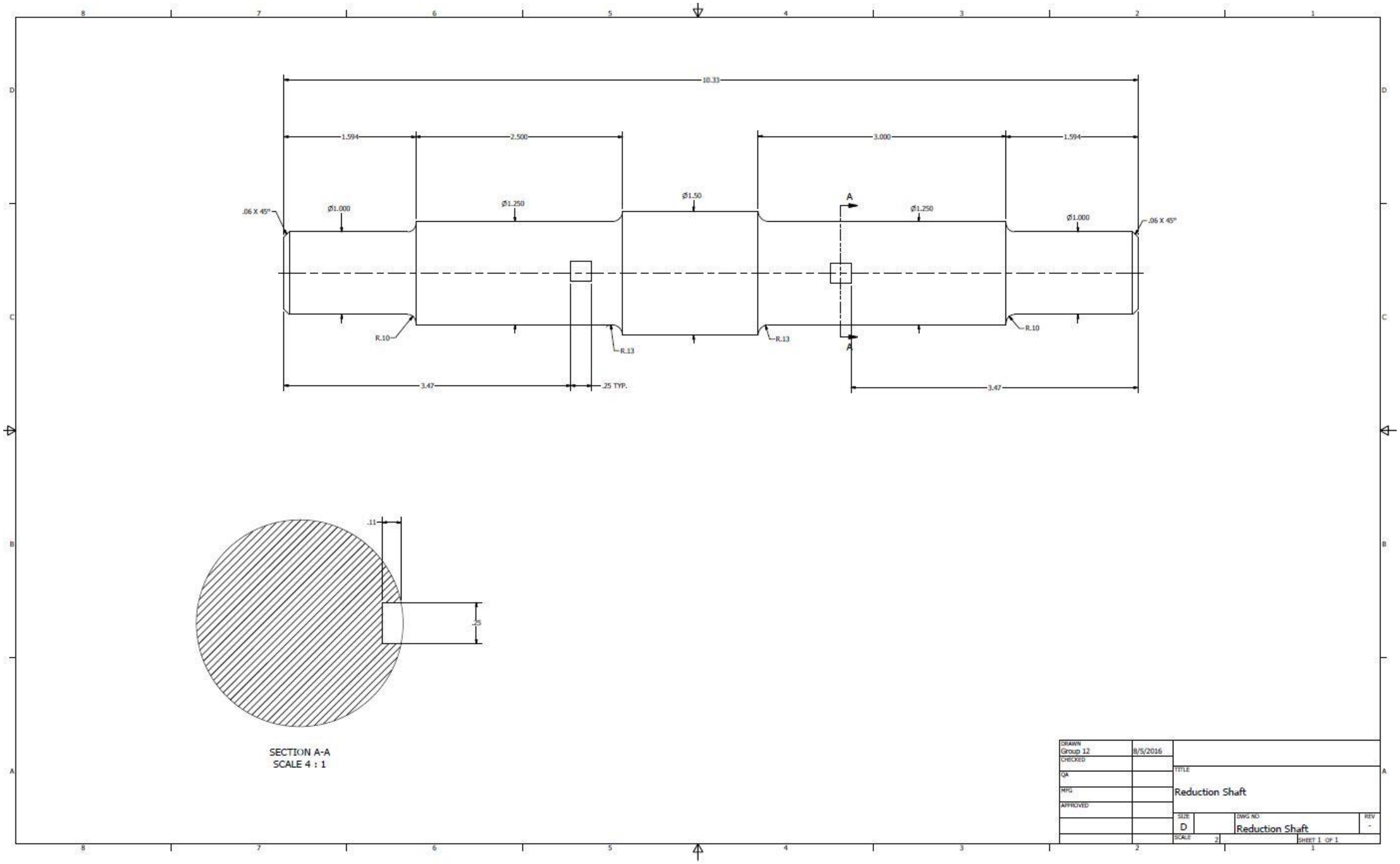
The overall dimensions of the final frame were: 96” long, 25” wide, and 23” tall. These dimensions were chosen for many reasons. The height was chosen so the total loading height for the bed when the tires were added wouldn’t be too high off the ground to make loading and unloading the repair vehicle difficult. Also this height allows the driver to sit at a comfortable distance above the ground. The width was chosen to allow the engine to be put in the center of the car and have enough clearance on either side for the transmission/braking levers and the transmission itself. The overall length was chosen to be 96” long so there would be plenty of room to mount a 3’ x 3’ cargo bed on the frame, have the engine easily accessible for everyday tasks such as filling up the gas tank and eventual maintenance of the engine, and have comfortable and safe space for the driver. This length allows the driver to be a little further away from the spinning engine and belts and allows there to be expanded metal guards placed around areas of concern.

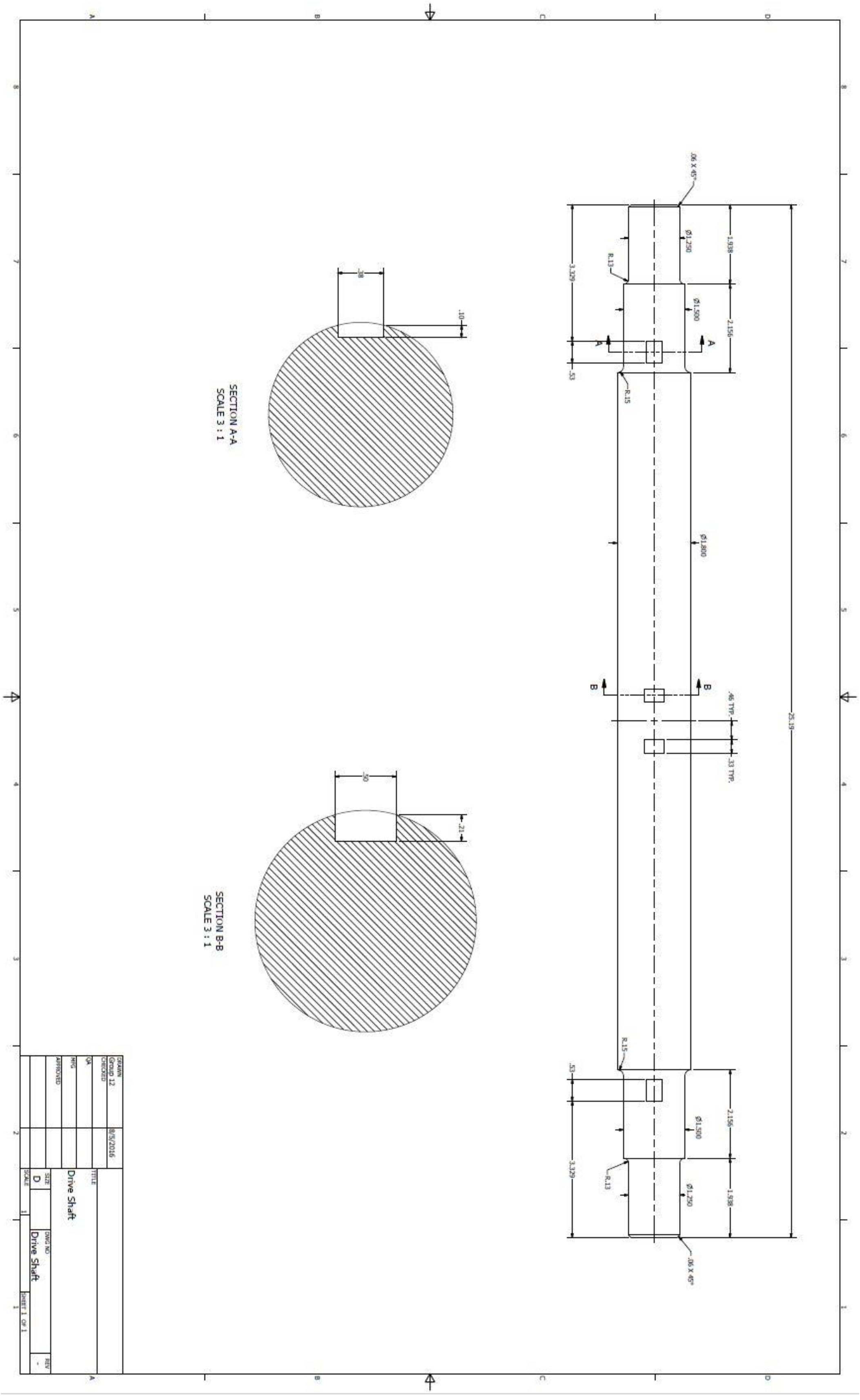
Safety guards needed to be attached to the frame to keep operators and bystanders safe during operation. As stated earlier, the guards were chosen to be made from expanded metal. Expanded metal was chosen as the material for the guards because it would be able to provide large amounts of airflow to the engine, while being ridged enough to keep objects out of unwanted areas. The guards are mounted to the frame using weld nuts which are welded to the frame, and bolts and washers so that way they can be taken off for maintenance purposes.

Another consideration was the driver's area. This area, as stated earlier had to be as safe and comfortable as possible. The floor in the driver's area is made of diamond plate for traction when getting in and out of the vehicle as well as for aesthetic. Part of the driver's area is where he or she is while operating the vehicle. The vehicle comes with a seat that includes a seatbelt. The seat can be adjusted for driver's preferences. Because the driver is somewhat near moving belts for the reverse portion of the transmission, expanded metal guards were placed to keep the driver from getting injured.

4. Layout and Detail Drawings




10.33
1.594
2.500
3.000
1.594
.06 X 45°
Ø1.000
Ø1.250
Ø1.50
Ø1.250
Ø1.000
.06 X 45°
A
A
R.10
R.13
R.13
R.10
3.47
.25 TYP.
3.47
.11
.25
SECTION A-A
SCALE 4 : 1
DRAWN
Group 12
8/5/2016
CHECKED
QA
MFG
APPROVED
TITLE
Reduction Shaft
SIZE
D
DWG NO
Reduction Shaft
REV
-
SCALE
2
SHEET 1 OF 1


SECTION A-A
SCALE 3 : 1
SECTION B-B
SCALE 3 : 1
.38
.10
.50
.21
.06 X 45°
Ø1.250
Ø1.500
Ø1.800
1.938
2.156
3.329
.53
R.13
R.15
25.19
.46 TYP.
.33 TYP.
A
B
DRAWN
Group 12
8/5/2016
CHECKED
QA
MFG
APPROVED
TITLE
Drive Shaft
SIZE
D
DWG NO
Drive Shaft
REV
-
SCALE
1
SHEET 1 OF 1

5. Cost Analysis

After completing the design for the repair vehicle, a basic cost analysis was conducted to determine a reasonable selling price for this vehicle. Factors that were considered included the price of purchased components, the labor cost required to design and manufacture the vehicle, and a reasonably competitive markup for a final sale price.

5.1 Labor Cost Calculations

Type of Labor	Cost/Hr of Labor	Expected Hours of Labor	Total Cost of Labor
Shipping and Receiving	$20.00	1	$20.00
Initial Inspection	$20.00	1.5	$30.00
Standard Machining	$20.00	18	$360.00
General Assembly	$20.00	40	$800.00
Welding	$40.00	3	$120.00
Welding (via PRI Robotics)	$1.87 per weld	(2)	($84.15)
Final Inspection	$20.00	2	$40.00
Shipping	$20.00	2	$40.00

- Machining time estimates by part
  - Welding assumptions
    - 1/8” welds; 45 welds; Each weld 4” long (15’ total welding)
  - Reduction Shaft
    - 3” pulley spacer: 0.5 hour
    - 3” single reduction pulley bushing: 1.5 hour
    - 12” double reduction pulley bushing: 1.5 hour
    - 12” pulley spacer: 0.5 hour
    - (2) reduction shaft keys: 0.5 hour
  - Drive Shaft
    - 12” single drive pulley bushing: 0.5 hour
    - 12” single pulley and bearing spacer: 1.5 hour
    - Brake rotor spacer: 0.5 hour
    - (2) keys (disk and drive): 0.5 hour
    - (2) hub keys: 0.5 hour
  - Boring of brake disk and re key seating: 1 hour
  - Cutting of frame pieces: 3 hours
  - Press bed into shape: 1 hour
  - Downtime organizational cost (example: bringing part to next machine): 5 hours
    - All standard finishes (finish turn ±0.001”) therefore remain at 100% cost

5.2 Purchased Components and Final Sale Pricing

Master Purchased Parts List

Part Number	Name	Supplier	Unit Price	Quantity to Purchase	Total Cost
15T212-0160-F8	10hp Gas Engine	Northern Tool + Equipment	$269.99	1	$269.99
6209K198	12" Double Pulley	McMaster Carr	$95.38	1	$95.38
6209K236	12" Single Pulley	McMaster Carr	$86.77	1	$86.77
6209K111	3" Double Pulley	McMaster Carr	$27.13	2	$54.26
6209K201	3" Single Pulley	McMaster Carr	$18.56	3	$55.68
KDBRKIT3PC	Brakes	Gopowersports.com	$59.00	1	$59.00
9052-0062	Double V Belt (F E3-R12)	V Belt Supply	$16.74	1	$16.74
9002-2065	Single V Belt (F/R R3-D12)	V Belt Supply	$7.18	1	$7.18
--	Double V Belt (R E3-R12)	V-Belt Guys	$17.42	1	$17.42
50784	Front Tires	Northern Tool + Equipment	$25.49	2	$50.98
13545	Rear Tire	Northern Tool + Equipment	$51.99	1	$51.99
--	Miscellaneous	(Fasteners)	$200.00	1	$200.00
2680T16	Seat	McMaster Carr	$121.41	1	$121.41
--	Labor cost	--	$1,410.00	1	$1,410.00
6494K16	Drive Shaft Bearings	McMaster Carr	$61.83	2	$123.66
6494K14	Reduction Shaft Bearings	McMaster Carr	$39.74	2	$79.48
5972K293	Axle Bearings	McMaster Carr	$22.58	3	$67.74
497470	Quite muffler option	Sears	$84.99	1	$84.99
T11111	1" x 1" tubing	Metals Depot	$37.20	2	$74.40
T12211	2" x 2" tubing	Metals Depot	$79.44	1	$79.44
F1181	1" x 1/8" flat bar	Metals Depot	$9.20	1	$9.20
S112	12 ga sheet metal	Metals Depot	$88.00	1	$88.00
5913K61	Steering Bearing	McMaster Carr	$10.95	2	$21.90
5174T2	Steering Rack	McMaster Carr	$23.46	1	$23.46
5172T21	Steering Gear	McMaster Carr	$22.00	1	$22.00
410219	13" Steering Wheel	BMI Karts & Supplies	$14.95	1	$14.95
P218	2' x 4' Diamond plate	Metals Depot	$66.48	1	$66.48
E11418F	4' x 8' Expanded Metal	Metals Depot	$144.00	1	$144.00

- Total purchased parts material cost: $1,986.50
- Combined total cost: Material cost + Labor cost = $3,396.50
- Sale price: 1.5 * Total cost = $5,094.75 (rounded up to $5,099.99 for final sale price)

Throughout the course of this design project, the master purchased parts list seen above was used to track purchased components. At the conclusion of the project, the costs were totaled and a reasonable profit margin was applied to the final selling price. Considering similar styled vehicles, this price should provide a competitive edge while not limiting the functionality or quality of the design itself.

APPENDICES

A – TEXTBOOK EQUATIONS, TABLES, AND FIGURES

$$S_f = a\,N^b \tag{6-13}$$

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases} \tag{6-20}$$

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion} \end{cases} \tag{6-26}$$

$$K_f = 1 + q(K_t - 1) \qquad \text{or} \qquad K_{fs} = 1 + q_{\text{shear}}(K_{ts} - 1) \tag{6-32}$$

$$\begin{aligned} \theta_d &= \pi - 2\sin^{-1}\frac{D-d}{2C} \\ \theta_D &= \pi + 2\sin^{-1}\frac{D-d}{2C} \end{aligned} \tag{17-1}$$

$$L = [4C^2 - (D-d)^2]^{1/2} + \frac{1}{2}(D\theta_D + d\theta_d) \tag{17-2}$$

$$\frac{F_1 - mr^2\omega^2}{F_2 - mr^2\omega^2} = \frac{F_1 - F_c}{F_2 - F_c} = \exp(f\phi) \tag{17-7}$$

$$F_i = \frac{T}{d}\,\frac{\exp(f\phi) + 1}{\exp(f\phi) - 1} \tag{17-9}$$

$$L_p = 2C + \pi(D+d)/2 + (D-d)^2/(4C) \tag{17-16a}$$

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

DE-Goodman
Diameter

$$\frac{1}{n} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}}$$

DE-Goodman
Safety Factor

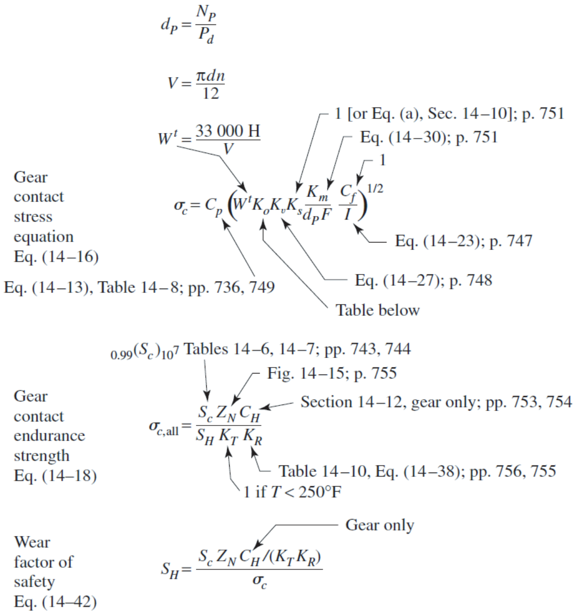
$$\sigma'_a = (\sigma_a^2 + 3\tau_a^2)^{1/2} = \left[\left(\frac{32K_f M_a}{\pi d^3} \right)^2 + 3 \left(\frac{16K_{fs} T_a}{\pi d^3} \right)^2 \right]^{1/2}$$

$$\sigma'_m = (\sigma_m^2 + 3\tau_m^2)^{1/2} = \left[\left(\frac{32K_f M_m}{\pi d^3} \right)^2 + 3 \left(\frac{16K_{fs} T_m}{\pi d^3} \right)^2 \right]^{1/2}$$

Combined
Loading

$$V = \pi\, dn/12 \qquad \text{ft/min}$$

Belt Speed (e)



$$d_P = \frac{N_P}{P_d}$$

$$V = \frac{\pi dn}{12}$$

$$W^t = \frac{33\,000\ H}{V}$$

$$\sigma_c = C_p \left(W^t K_o K_v K_s \frac{K_m}{d_P F} \frac{C_f}{I} \right)^{1/2}$$

$$\sigma_{c,\text{all}} = \frac{S_c Z_N C_H}{S_H K_T K_R}$$

$$S_H = \frac{S_c Z_N C_H/(K_T K_R)}{\sigma_c}$$

Gear
Equations

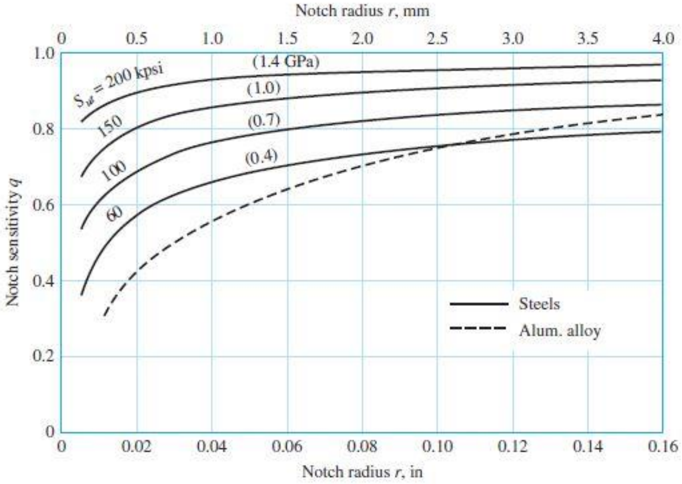


Figure 6-20

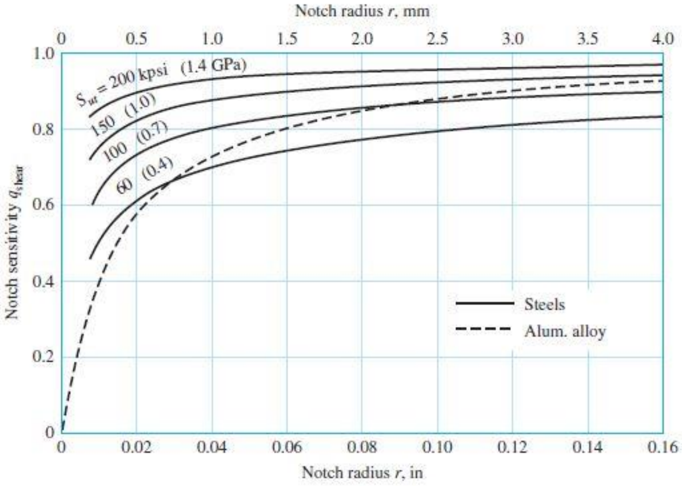


Figure 6-21

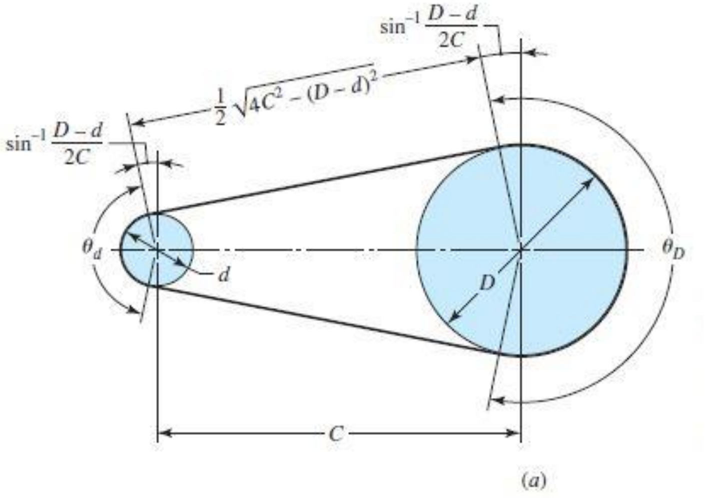


Figure 17-1a

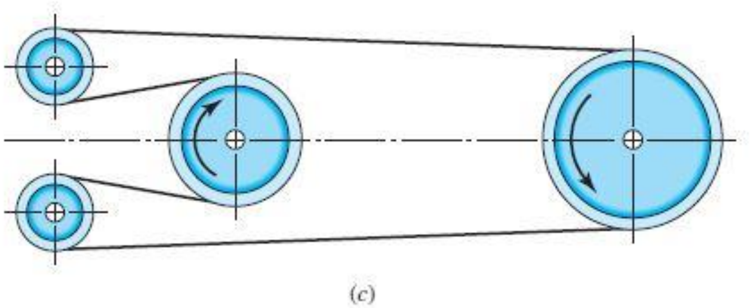


Figure 17-2c

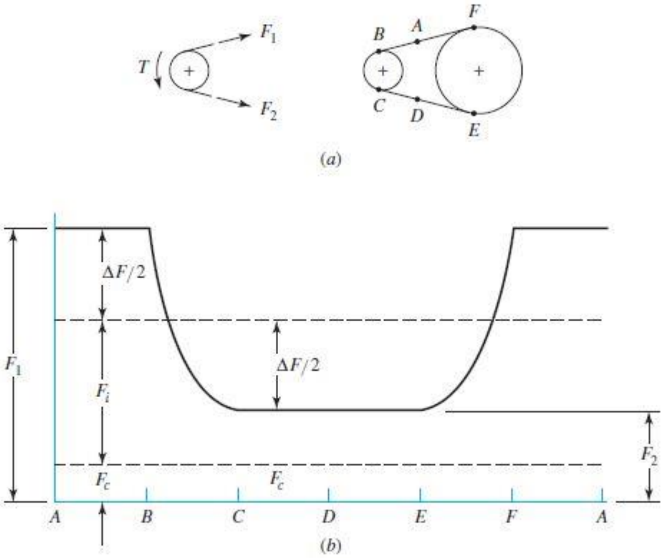


Figure 17-12

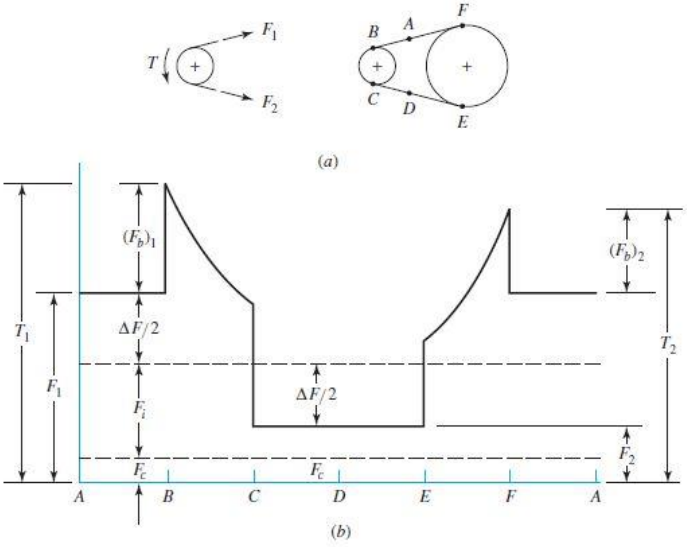


Figure 17-14

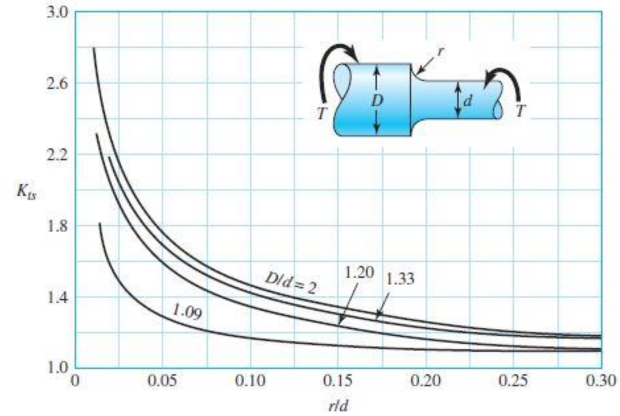


Figure A-15-8

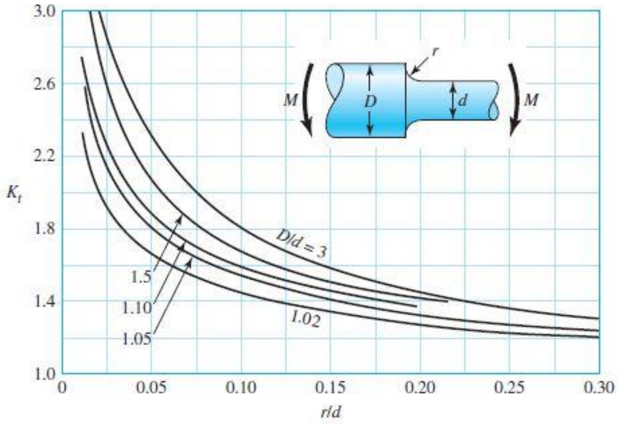
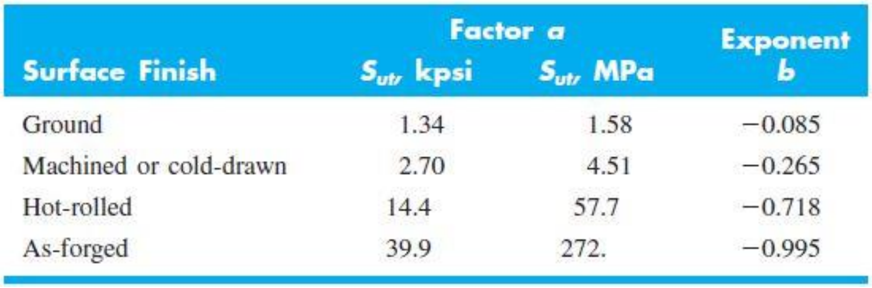
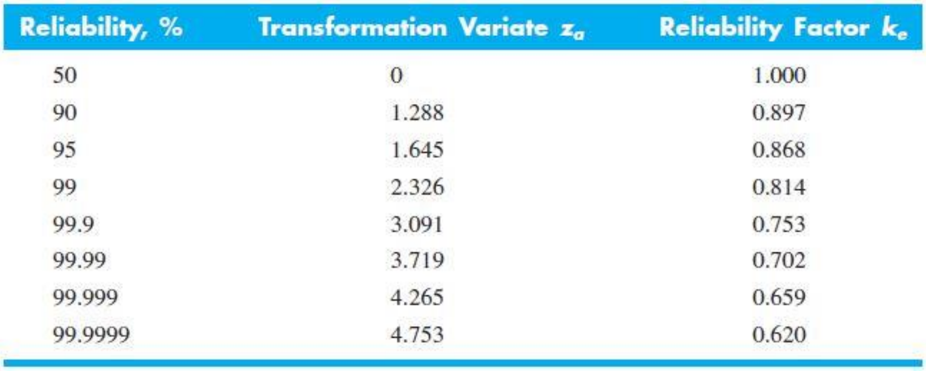


Figure A-15-9



Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Table 6-2



Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Table 6-5

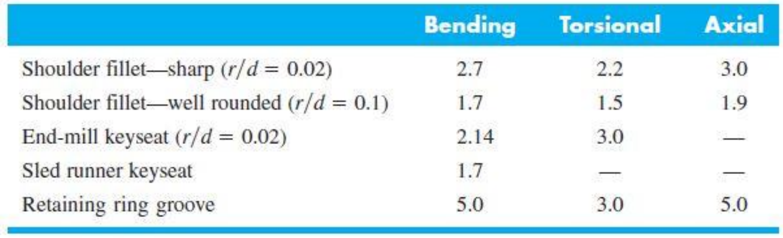


Table 7-1

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

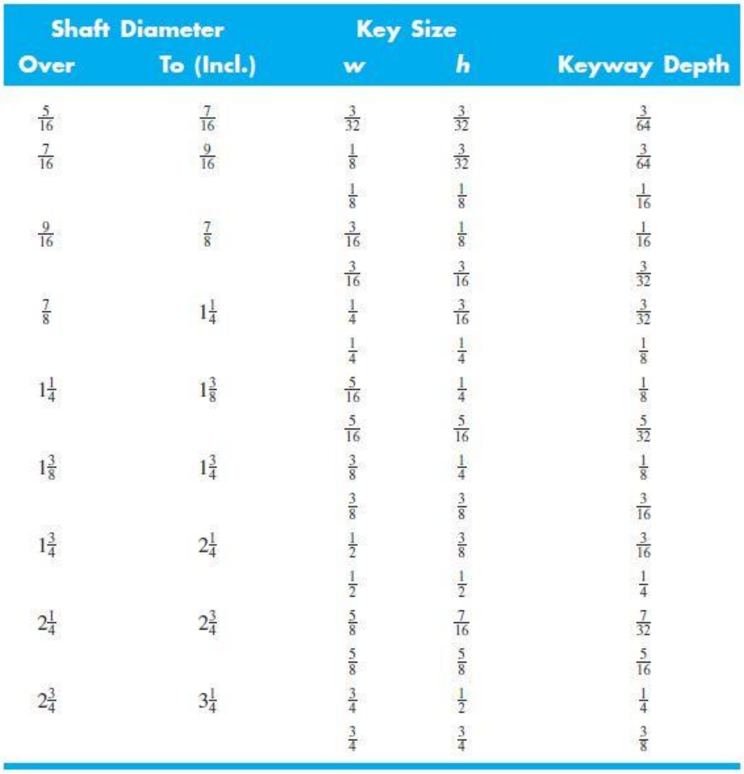


Table 7-6

Shaft Diameter		Key Size		
Over	To (Incl.)	*w*	*h*	Keyway Depth
$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{64}$
$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{3}{64}$
		$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{16}$
		$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$
		$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$
$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{32}$
$1\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$
		$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$
$2\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{32}$
		$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{16}$
$2\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$
		$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$

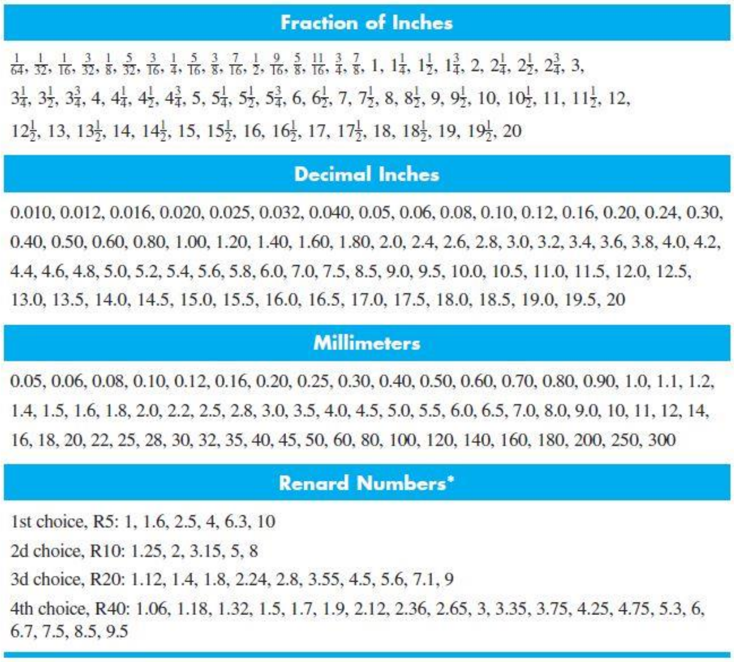


Table A-17

Fraction of Inches

$\frac{1}{64}, \frac{1}{32}, \frac{1}{16}, \frac{3}{32}, \frac{1}{8}, \frac{5}{32}, \frac{3}{16}, \frac{1}{4}, \frac{5}{16}, \frac{3}{8}, \frac{7}{16}, \frac{1}{2}, \frac{9}{16}, \frac{5}{8}, \frac{11}{16}, \frac{3}{4}, \frac{7}{8}, 1, 1\frac{1}{4}, 1\frac{1}{2}, 1\frac{3}{4}, 2, 2\frac{1}{4}, 2\frac{1}{2}, 2\frac{3}{4}, 3,$
$3\frac{1}{4}, 3\frac{1}{2}, 3\frac{3}{4}, 4, 4\frac{1}{4}, 4\frac{1}{2}, 4\frac{3}{4}, 5, 5\frac{1}{4}, 5\frac{1}{2}, 5\frac{3}{4}, 6, 6\frac{1}{2}, 7, 7\frac{1}{2}, 8, 8\frac{1}{2}, 9, 9\frac{1}{2}, 10, 10\frac{1}{2}, 11, 11\frac{1}{2}, 12,$
$12\frac{1}{2}, 13, 13\frac{1}{2}, 14, 14\frac{1}{2}, 15, 15\frac{1}{2}, 16, 16\frac{1}{2}, 17, 17\frac{1}{2}, 18, 18\frac{1}{2}, 19, 19\frac{1}{2}, 20$

Decimal Inches

0.010, 0.012, 0.016, 0.020, 0.025, 0.032, 0.040, 0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.24, 0.30, 0.40, 0.50, 0.60, 0.80, 1.00, 1.20, 1.40, 1.60, 1.80, 2.0, 2.4, 2.6, 2.8, 3.0, 3.2, 3.4, 3.6, 3.8, 4.0, 4.2, 4.4, 4.6, 4.8, 5.0, 5.2, 5.4, 5.6, 5.8, 6.0, 7.0, 7.5, 8.5, 9.0, 9.5, 10.0, 10.5, 11.0, 11.5, 12.0, 12.5, 13.0, 13.5, 14.0, 14.5, 15.0, 15.5, 16.0, 16.5, 17.0, 17.5, 18.0, 18.5, 19.0, 19.5, 20

Millimeters

0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.25, 0.30, 0.40, 0.50, 0.60, 0.70, 0.80, 0.90, 1.0, 1.1, 1.2, 1.4, 1.5, 1.6, 1.8, 2.0, 2.2, 2.5, 2.8, 3.0, 3.5, 4.0, 4.5, 5.0, 5.5, 6.0, 6.5, 7.0, 8.0, 9.0, 10, 11, 12, 14, 16, 18, 20, 22, 25, 28, 30, 32, 35, 40, 45, 50, 60, 80, 100, 120, 140, 160, 180, 200, 250, 300

Renard Numbers*

1st choice, R5: 1, 1.6, 2.5, 4, 6.3, 10

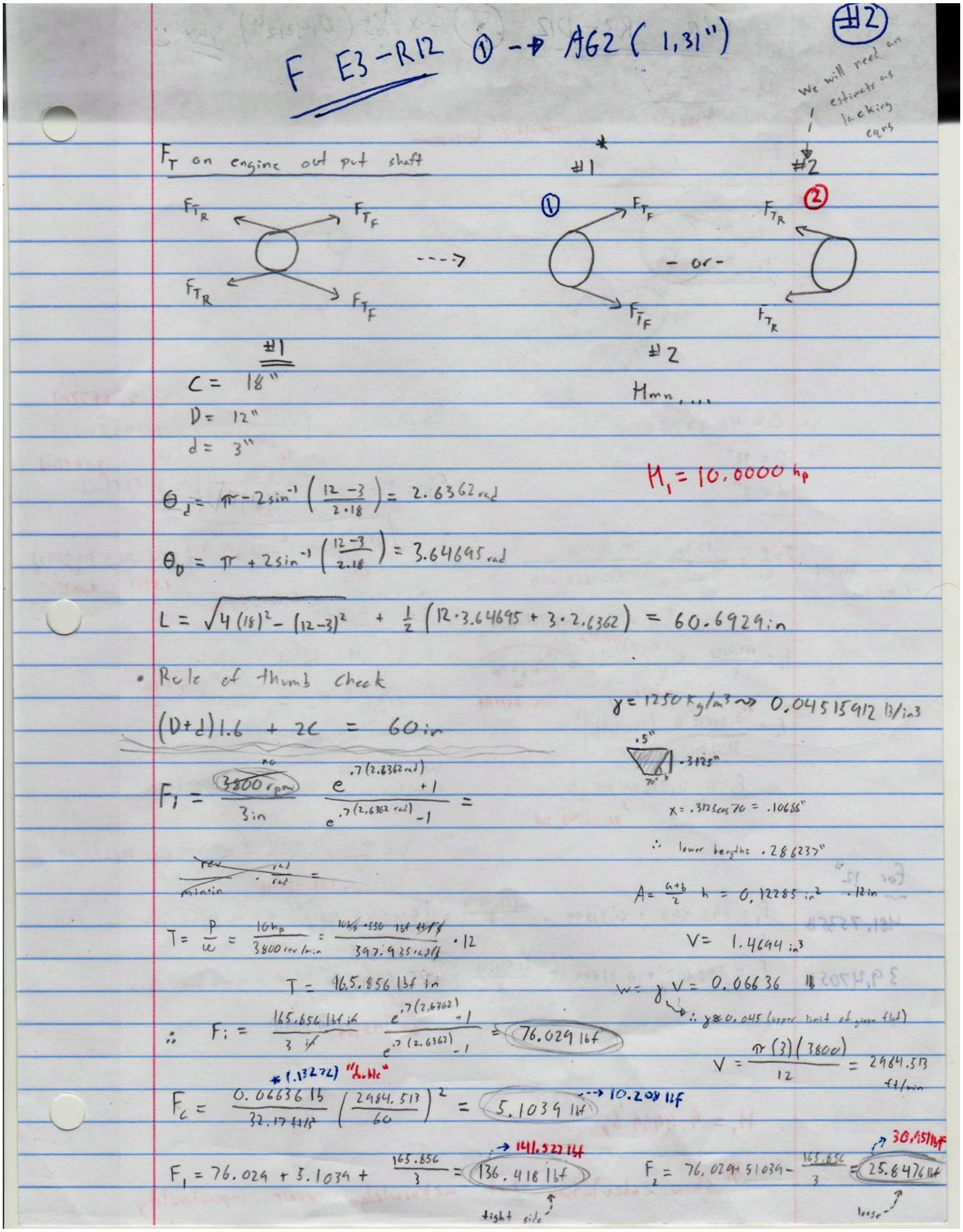
2d choice, R10: 1.25, 2, 3.15, 5, 8

3d choice, R20: 1.12, 1.4, 1.8, 2.24, 2.8, 3.55, 4.5, 5.6, 7.1, 9

4th choice, R40: 1.06, 1.18, 1.32, 1.5, 1.7, 1.9, 2.12, 2.36, 2.65, 3, 3.35, 3.75, 4.25, 4.75, 5.3, 6, 6.7, 7.5, 8.5, 9.5

B – SCANNED WORK SHEETS

“Belt Tension Calculations”



F E3-R12 ① → A62 (1,31")

#2

We will need an estimate as lacking ears

F_T on engine output shaft

#1

C = 18"

D = 12"

d = 3"

#2

Hmm, ...

$H_1 = 10.0000\ hp$

$\theta_d = \pi - 2\sin^{-1}\left(\frac{12-3}{2\cdot 18}\right) = 2.6362_{rad}$

$\theta_D = \pi + 2\sin^{-1}\left(\frac{12-3}{2\cdot 18}\right) = 3.64695_{rad}$

$L = \sqrt{4(18)^2-(12-3)^2} + \frac{1}{2}\left(12\cdot 3.64695 + 3\cdot 2.6362\right) = 60.6929in$

- Rule of thumb check

$(D+d)1.6 + 2C = 60in$

$F_1 = \frac{\overset{no}{\cancel{3800\ rpm}}}{3in}\ \frac{e^{.7(2.6362rad)}+1}{e^{.7(2.6362\ rad)}-1} =$

$\cancel{\frac{rev}{min\cdot in}\cdot\frac{rad}{rad}} =$

$T = \frac{P}{\omega} = \frac{10hp}{3800\ rev/min} = \frac{10hp\cdot 550\ lbf\ \cancel{ft/s}}{397.935\ rad/s}\cdot 12$

$T = 165.856\ lbf\ in$

$\therefore\ F_i = \frac{165.856\ lbf\ in}{3\ in}\ \frac{e^{.7(2.6362)}+1}{e^{.7(2.6362)}-1} = \boxed{76.029\ lbf}$

$\gamma = 1250\ kg/m^3 \rightsquigarrow 0.04515912\ lb/in^3$

$x = .3125\cos 70 = .10688"$

$\therefore$ lower lengths: .286237"

$A = \frac{a+b}{2}h = 0.12285\ in^2 \quad \cdot 12in$

$V = 1.4694\ in^3$

$w = \gamma V = 0.06636\ lb$

$\therefore \gamma \approx 0.045$ (upper limit of given flat)

$V = \frac{\pi(3)(3800)}{12} = 2984.513\ ft/min$

* (.13272) "double"

$F_c = \frac{0.06636\ lb}{32.17\ ft/s^2}\left(\frac{2984.513}{60}\right)^2 = \boxed{5.1039\ lbf} \dashrightarrow 10.208\ lbf$

$F_1 = 76.029 + 5.1039 + \frac{165.856}{3} = \boxed{136.418\ lbf} \rightarrow 141.522\ lbf$

tight side

$F_2 = 76,029 + 51039 - \frac{165.856}{3} = \boxed{25.8476\ lbf} \rightarrow 30.451\ lbf$

loose

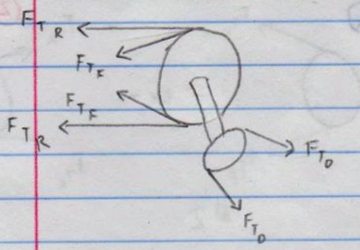
F/R R3 – D12 ③ --> A65 (0,4212") yay :)

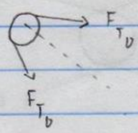
#12a

$$H = \frac{(F_1 - F_2)V}{33000}$$ -- "transmitted horsepower"

$\cancel{F_1 - F_2 = \frac{2T}{d}}$

F_t on reduction shaft





$C = 46.227697 \text{ in} \dashrightarrow 20"$

$D = 12"$

$d = 3"$

$\theta_d = \pi - 2\sin^{-1}\left(\frac{12-3}{2 \cdot 46.227697}\right) = 2.7497 \text{ rad}$ → 2.6877 rad (20")

$\theta_D = \pi + 2\sin^{-1}\left(\frac{12-3}{2 \cdot 46.227697}\right) = 3.5335 \text{ rad}$ → 3.5955 rad (20")

$L = \sqrt{4(46.227697)^2 - (12-3)^2} + \frac{1}{2}(12 \cdot 3.5335 + 3 \cdot 2.7497)$ (20") (3.5955) (2.6877)

$L = 112.3417 \text{ in}$ → 64.5788 in

f=0.7 from Table 17-2 w/ w=.5in

$T = \frac{P}{\omega} = \frac{10 \text{ hp}}{950 \text{ rev/min}} = \frac{10 \cdot 550 \cdot 12}{99.4838 \text{ rad/s}}$

$T = 663.4248 \text{ lbf in}$

- Rule of thumb check

$(D+d)1.6 + 2C = 116.4554 \text{ in}$ → 64 in

$F_i = \frac{663.4248}{3} \cdot \frac{e^{.7(2.7497)}+1}{e^{.7(2.7497)}-1} = 296.697 \text{ lbf}$ (2.6877) → 300.6515 lbf

$F_c = \frac{0.06636 \text{ lb}}{32.17 \text{ ft/s}^2}\left(\frac{746.1283}{60}\right)^2$ (2.)

$F_c = 0.31899 \text{ lbf}$ → 0.63798 lbf

W consistent throughout as same belt ∴ w = 0.06636 lb

$V = \frac{\pi d n}{12} = \frac{\pi(3)(950)}{12} = 746.1283 \text{ ft/min}$

For 12"

481.7535 lb

$F_1 = 296.697 + 0.31899 + \frac{663.4248}{3} = 518.157 \text{ lbf}$ → 522.431 lbf ←-- tight side

39.4705 lb

$F_2 = 296.697 + 0.31899 - \frac{663.4248}{3} = 75.8742 \text{ lbf}$ → 80.148 lbf ←--- loose side

$H_2 = 9.9999 \text{ hp}$

* show calculation but acknowledge error impracticality

#13

V-belt Length recalculation

$L_p = 2C + \pi(D+d)/2 + (D-d)^2/(4C)$

① $C = 18"$
$D = 12"$
$d = 3"$

$L_p = 2(18) + \pi(12+3)/2 + (12-3)^2/(4\cdot 18)$

$L_p = 60.6869444"$

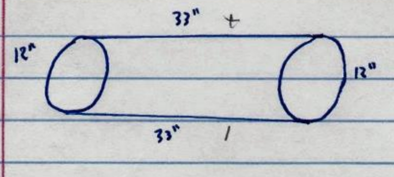
③ $C = 20"$
$D = 12"$
$d = 3"$

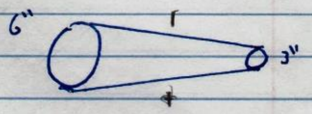
$L_p = 2(20) + \pi(12+3)/2 + (12-3)^2/(4\cdot 18)$

$L_p = 64.5744449"$

② Rule of thumb: geo + 1.3 = 129.1359586 + 1.3 = 130.4359586"

·Estimate parts





$L_{g_1} = 2(33) + 2\left(6\cdot\frac{\pi}{2}\right) = 84.84955592"$

$L_T = L_{g_1} + L_{g_2} = 129.1367955$

$T_1 = \frac{P}{\omega} = \frac{10\cdot 550\cdot 12}{99.4838\text{ rad/s}} = 663.4248\text{ lbf·in}$

$\theta_d = \pi$ rad $\theta_d = \pi$ rad $V = \frac{\pi d n}{12} = 2984.513$ ft/min

$f = .7$

$F_i = \frac{663.4248}{12}\;\frac{e^{.7(\pi)}+1}{e^{.7(\pi)}-1} = 69.0774\text{ lbf}$

$F_c = \frac{.13272}{32.17}\left(\frac{2984.513}{60}\right)^2 = 10.20771\text{ bf}$

① $F_1 = F_i + F_c + \frac{663.4248}{12} = \boxed{134.5705\text{ lbf}}$

② $F_2 = F_i + F_c - \frac{663.4248}{12} = \boxed{23.9997\text{ lbf}}$

$L_{g_2} = 2\left(\sqrt{1.492^2 + 14.850^2}\right) + 3(3.34195645) + 1.5(2.94122886)$

$L_{g_2} = 44.28723454"$

$T = \frac{P}{\omega} = \frac{10\cdot 550\cdot 12}{397.9351} = 165.8562\text{ lbf·in}$

$\theta_d = \pi - 2\sin^{-1}\left(\frac{6-3}{2\cdot 15}\right) = 2.9413\text{ rad}$

$\theta_D = \pi + 2\sin^{-1}\left(\frac{6-3}{2\cdot 15}\right) = 3.3419\text{ rad}$

$f = .7$ $V = \frac{\pi d n}{12} = \frac{\pi(3)(\overset{3800}{\cancel{397.9351}})}{12} = 2984.513$ ft/min

$F_i = \frac{165.8562}{3}\;\frac{e^{.7(2.9413)}+1}{e^{.7(2.9413)}-1} = 71.4568\text{ lb}$

$F_c = \frac{.13272}{32.17}\left(\frac{2984.513}{60}\right)^2 = 10.2077\text{ lbf}$

③ $F_1 = \boxed{136.9499\text{ lbf}}$

④ $F_2 = \boxed{26.3741\text{ lbf}}$

#4

$\frac{T}{d} = \frac{\text{torque}}{\text{diameter}}$

$F_c = \frac{w}{g}\left(\frac{V}{60}\right)^2$

w = weight of foot of belt
↳ density · volume

$V = \frac{\pi d n}{12}$

d = diameter (in)
n = rot speed (rev/min)

$F_i = \frac{T}{d} \frac{e^{f\phi} + 1}{e^{f\phi} - 1}$

f = coefficient of friction (table 17-2)

$\phi = \theta_d$

tight → $F_1 = F_i + F_c + \frac{T}{d}$

loose → $F_2 = F_i + F_c - \frac{T}{d}$

$$\theta_d = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\theta_D + d\theta_d)$$

$$(F_1)_a = b F_a C_p C_v$$

b = width

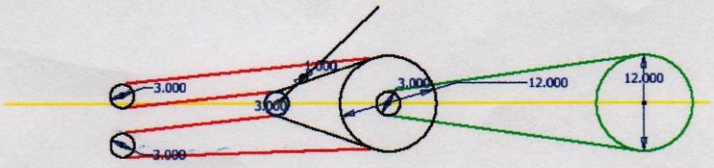
F_a = max tension

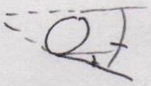
C_p = pulley correction

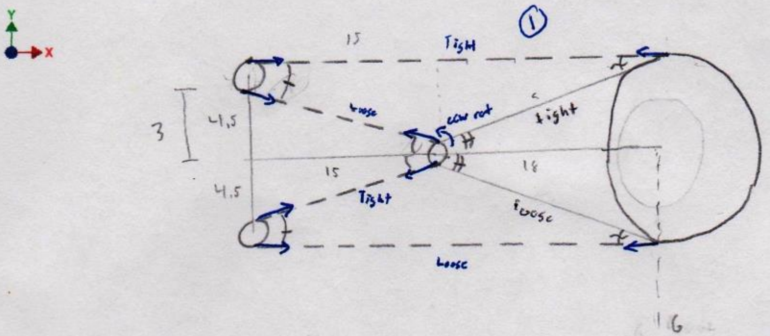
C_v = velocity correction

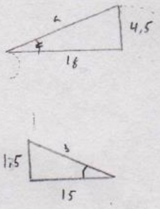
S=rθ

#5







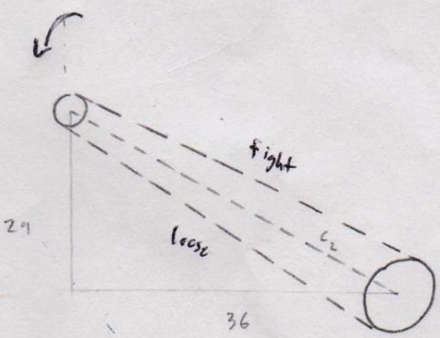


$\measuredangle$: $\tan^{-1}(1.5/15) = 5.7106°$

$\measuredangle = \measuredangle$

$\measuredangle = \tan^{-1}(4.5/18) = 14.0362°$ → belt angles

$\measuredangle = \measuredangle$



*

Pure L_1

$2\cdot\left(\sqrt{4.353^2+16.859^2}\right) + 6''(3.65262506\,\text{rad}) + 3''(2.63644453\,\text{rad})$

4.357 16.875 3.64704 1.5 2.6361453

2 · hyp big p arc engine arc

$= 64.64815''$

60.66227144"

$C_2 = \sqrt{29^2+36^2} = 46.22769733\,\text{in}$

Pure L_2

$2\cdot\left(\sqrt{4.515^2+15.296^2}\right) + 2\cdot 1.5''(3.042807402\,\text{rad}) + 2\cdot(15''+18'') + \frac{1}{2}\pi(12'') + 3''(2.94332325\,\text{rad}) = 135.5496345''$

2.944619542 1.5

2 · hyp 2 · little p arc hor big p arc engine arc

124.1359586"

Pure L_3

$2\cdot\sqrt{\cancel{32.212}^2+\cancel{32.646}^2} + 3''(2.44163938\,\text{rad}) + 12''(3.34213048\,\text{rad}) = 140.4586204''$

14.4 12.559 1.5 2.68763252 6 3.59555279

2 · hyp little p arc big p arc

→ 90.18329097" → 64.57852457"

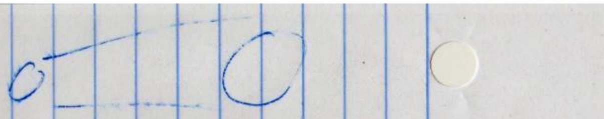
#6

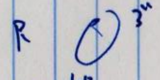
$\frac{rev}{min} \cdot \frac{\pi d \text{ in}}{1 \text{ rev}} \cdot \frac{ft}{12 \text{ in}} = ft/min$

Table 17-16

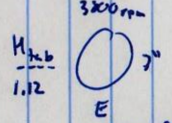
$K_b = 220$

$K_c = 0.561$



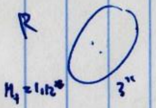


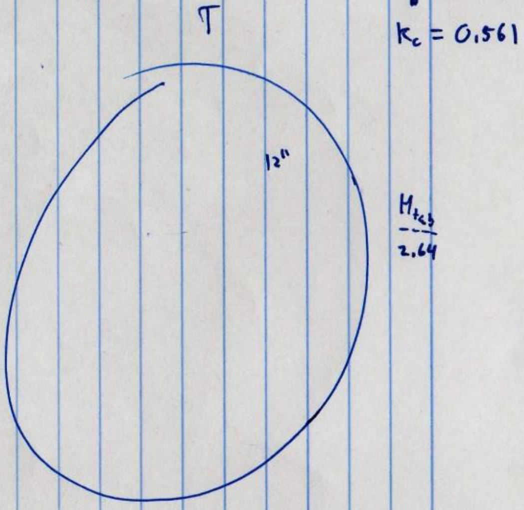
$M_t = 1.12$



~~[illegible] ft/min~~

2984.513 ft/min
(everywhere)





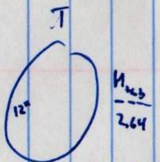
$H_a = K_1 K_2 M_t = 2.079$

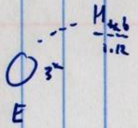
$\Delta F = 14443.22917$??

$F_c = 4.997$ lb

$H_a = K_1 K_2 M_t =$

$K_2 \Rightarrow 44.3" \therefore 0.9$



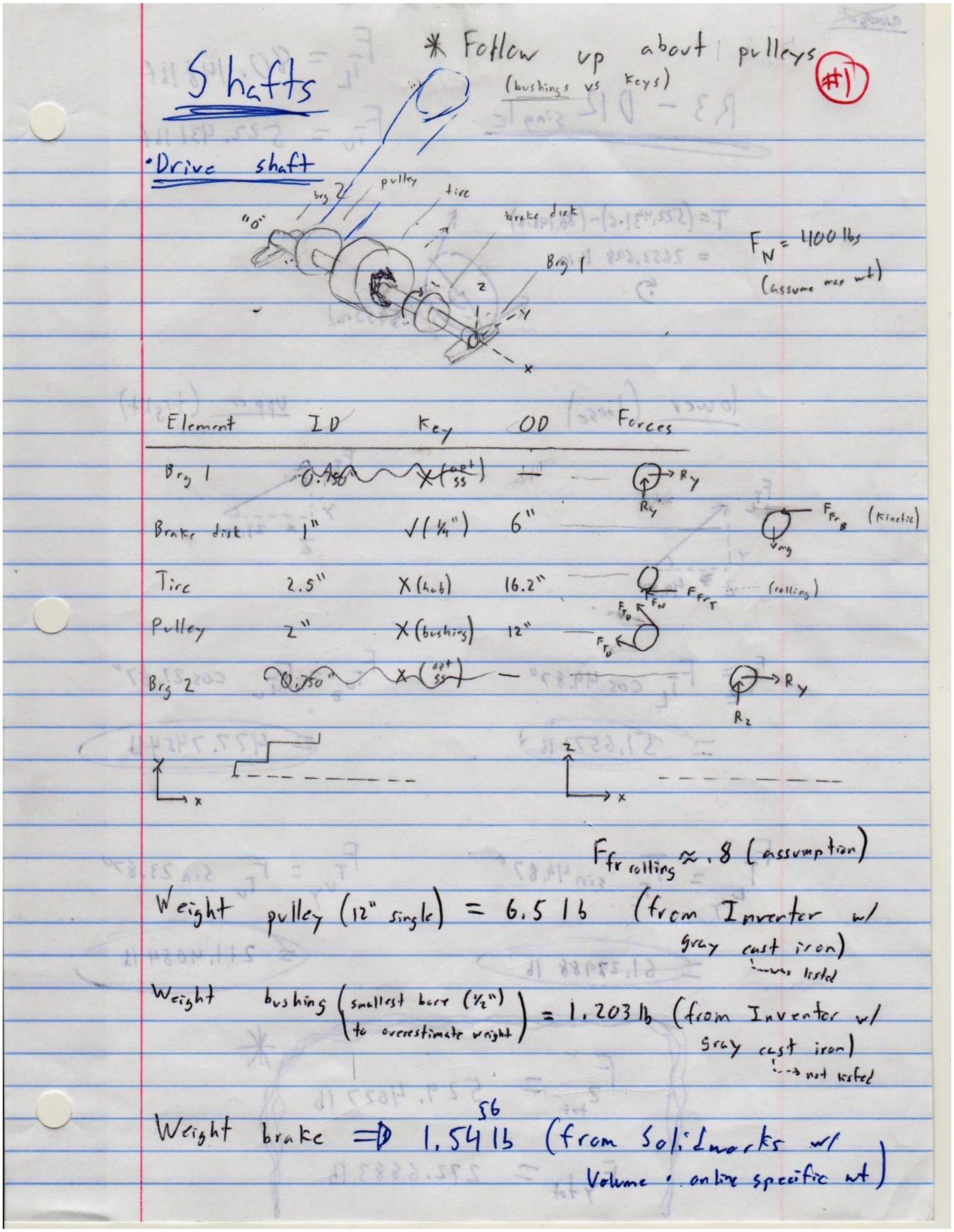


$K_1 \Rightarrow \frac{12-12}{33} = 0 \therefore .75$

$K_1 \Rightarrow \frac{6-3}{15} = .2 \therefore .78$

$K_2 \Rightarrow 84.8" \therefore 1.05$

"Drive Shaft Calculations"



Shafts

* Follow up about pulleys (bushings vs keys) #1

•Drive shaft

$F_N = 400$ lbs (assume max wt)

Element	ID	Key	OD	Forces
Brg 1	~~0.750~~	~~X (opt / ss)~~	—	R_y R_y
Brake disk	1"	✓ (¼")	6"	F_{Fr_B} (kinetic) mg
Tire	2.5"	X (hub)	16.2"	F_N F_{Fr_T} (rolling)
Pulley	2"	X (bushing)	12"	F_{T_0} F_{T_0}
Brg 2	~~0.750"~~	~~X (opt / ss)~~	—	R_y R_z

$F_{fr\ rolling} \approx .8$ (assumption)

Weight pulley (12" single) = 6.5 lb (from Inventor w/ gray cast iron) ↳ was listed

Weight bushing (smallest bore (½") to overestimate weight) = 1.203 lb (from Inventor w/ gray cast iron) ↳ not listed

Weight brake ⟹ $1.\overset{56}{54}$ lb (from SolidWorks w/ Volume · online specific wt)

~~escargot~~

R3 - D12 single

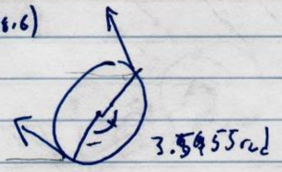
$F_{T_L} = 80.148\ \text{lbf}$

$F_{T_U} = 522.431\ \text{lbf}$

$T = (522,431 \cdot 6) - (80,148 \cdot 6)$

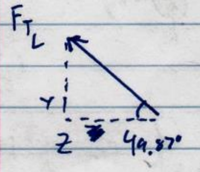
$= 2653.698\ \text{lb in}$

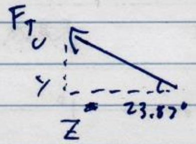
↺



lower (loose)

upper (tight)





$F_{T_{L_Z}} = F_{T_L} \cos 49.87°$

$= \boxed{51,6573\ \text{lb}}$

$F_{T_{U_Z}} = F_{T_U} \cos 23.87°$

$= \boxed{477.7454\ \text{lb}}$

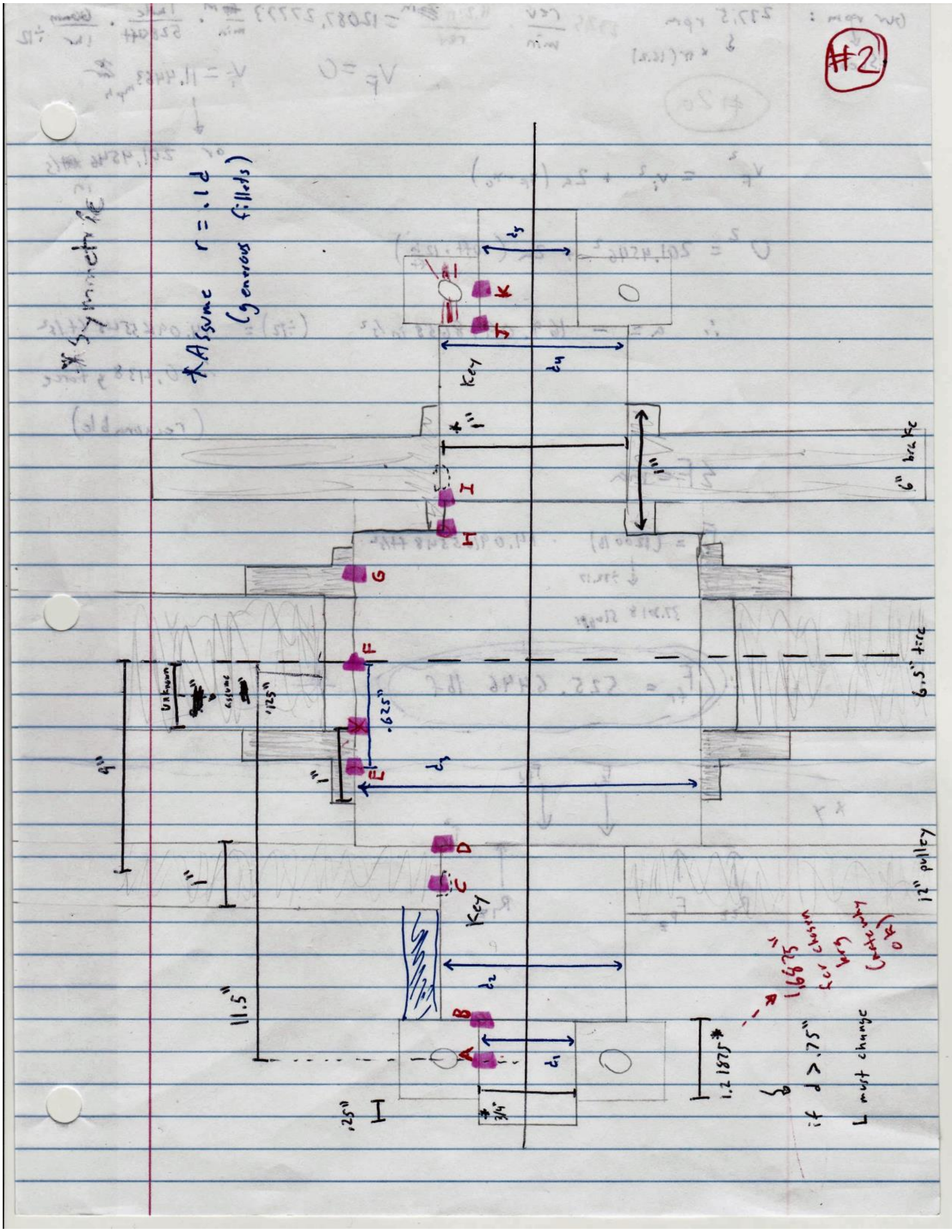
$F_{T_{L_Y}} = F_{T_L} \sin 49.87°$

$= \boxed{61,29988\ \text{lb}}$

$F_{T_{U_Y}} = F_{T_U} \sin 23.87°$

$= \boxed{211.4084\ \text{lb}}$

$$\boxed{\begin{aligned} F_{Z_{tot}} &= 529.4027\ \text{lb} \\ F_{Y\,tot} &= 272.6883\ \text{lb} \end{aligned}}$$ *


#2
*Symmetric
*Assume r = .1d
(generous fillets)
A
B
C
D
E
F
G
H
I
J
K
Key
Key
d_1
d_2
d_3
d_4
d_5
*1"
1"
6" brake
6.5" tire
12" pulley
4"
1"
11.5"
.25"
*3/4"
1.21875*
.625"
1"
.125"
Unknown
assume
if d > .75"
must change
1.6875" for chosen key (tentatively ok)

Our rpm : → Speed

237.5 rpm
$\times \pi(16.2)$

$237.5 \frac{rev}{min} \cdot \frac{16.2 \cdot \pi \text{ in}}{rev} = 12087.27773 \frac{in}{min} \cdot \frac{1 \text{ mile}}{5280 ft} \cdot \frac{60 min}{1 hr} \div 12$

#2a

$V_F = 0$ $\qquad V_i = 11.4463_{mph}$

or 201.4546 in/s

$$v_F^2 = v_i^2 + 2a(x_F - x_0)$$

$$0^2 = 201.4546^2 + 2a\left(10ft \cdot 12\frac{in}{ft}\right)$$

$\therefore a = -169.0998658 \text{ in/s}^2$ $\quad (\div 12) = 14.09165548 \text{ ft/s}^2$

$\therefore$ 0.438 g force

(reasonable)

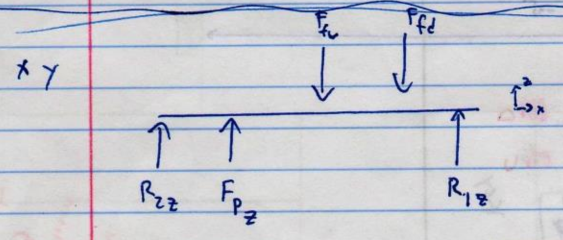
$$\Sigma F = ma$$

$F_{fr} = (1200 \text{ lb}) \cdot 14.09165548 \text{ ft/s}^2$

$\div 32.17$

37.3018 slugs

$$\therefore \boxed{F_{fr} = 525.6446 \text{ lbf}}$$ ☆

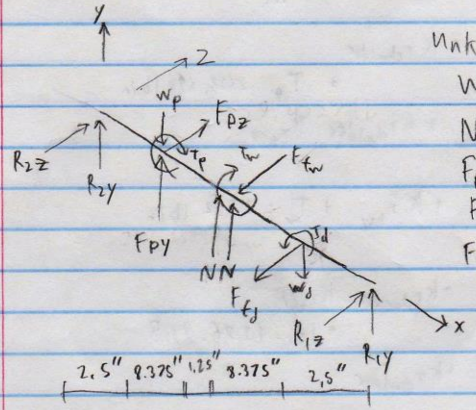


#3a

wheel friction: $f = 0.8$ $N = 400$

$F_{tw} = 320$ lb

$T_w = F_{tw} \cdot 8.1\text{ in} = 2592$ lb·in



Unknowns: R_{1y}, R_{1z}, R_{2y}, R_{2z}

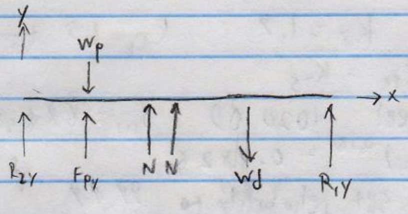
$W_p = 7.703$ lb $W_d = 1.5456$ lb

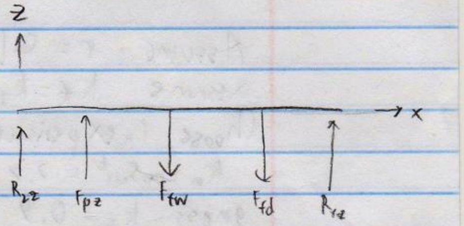
$N = 200$ lb $F_{py} = 272.6883$ lb

$F_{pz} = 529.4027$ lb $T_p = 2653.698$ lb in

$F_{tw} = 320$ lb $T_w = 2592$ lb in

$F_{fd} = 525.8446$ lb $T_d = 1576.9338$ lb in



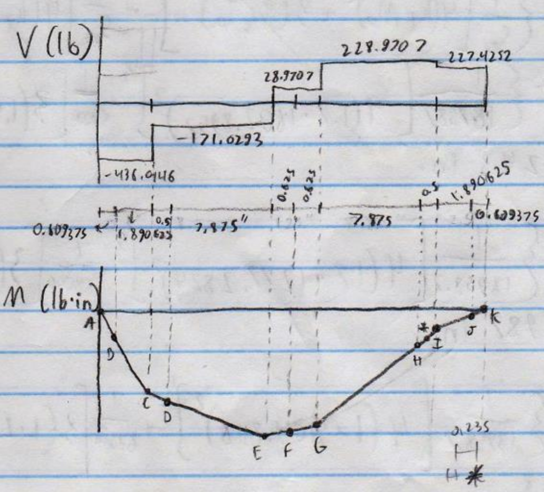


$\circlearrowleft \Sigma M_{R_{2y}} = 2.5(272.6883) - 2.5(7.703) + 10.875(200) + 12.125(200) - 20.5(1.5456) + 23 R_{1y} = 0$

$R_{1y} = -227.4252$ lb

$\uparrow + \Sigma F_y = -227.4252 + 272.6883 - 7.703 + 200 + 200 - 1.5456 + R_{2y} = 0$

$R_{2y} = -436.0146$



$A = 0$

$B = 0.609375 \cdot -436.0146 = -265.6964$

$C = 2.5 \cdot -436.0146 = -1090.0365$ lb in

$D = C + (0.5 \cdot -171.0293) = -1175.5512$ lb in

$E = C + (8.375 \cdot -171.0293) = -2522.4069$ lb in

$F = E + (0.625 \cdot 28.9707) = -2504.3002$ lb in

$G = F + (0.625 \cdot 28.9707) = -2486.1935$ lb in

$H = G + (7.875 \cdot 228.9707) = -683.0493$ lb in

$I = H + (0.5 \cdot 228.9707) = -568.5639$ lb in

$J = I + (1.890625 \cdot 227.4252) = -138.5881$ lb in

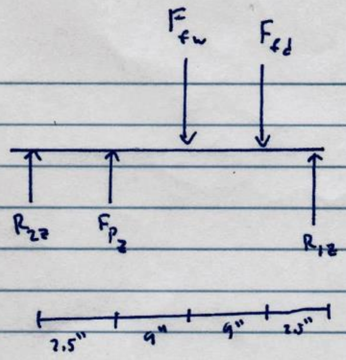
$K = 0$

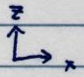
$* = -629.2412$ lb in

4 2

#36
4

XZ



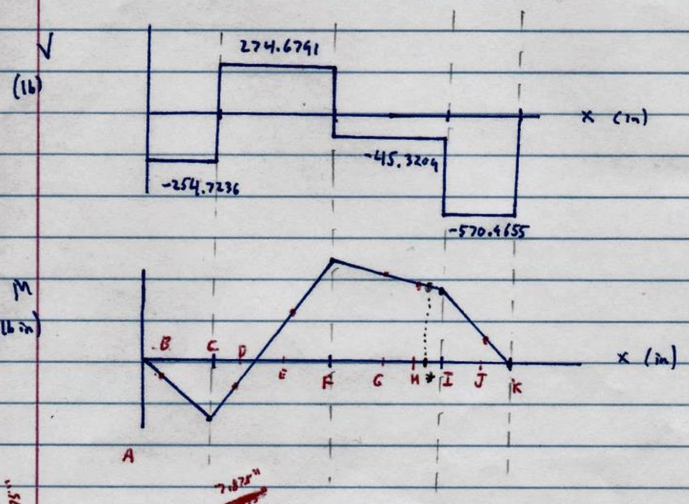


$$\circlearrowleft + \sum M_{R_2}: 2.5''(529.4027\text{ lb}) - 11.5''(320\text{ lb}) - 20.5''(525.6446\text{ lb}) + 23'' R_{1z} = 0$$

$$\therefore R_{1z} = 570.9655\text{ lb} \uparrow$$

$$+\uparrow \sum F_z: R_{2z} + 529.4027 - 320 - 525.6446 + 570.9655 = 0$$

$$\therefore R_{2z} = -254.7236\text{ lb} \downarrow$$



$A_1 = 2.5(-254.7236) = -636.869$

$A_2 = A_1 + 9 \cdot 274.6791 = 1835.3029$

$A_3 = A_2 + 9 \cdot -45.3209 = 1427.4148$

$A_4 = A_3 + -570.9655 \cdot 2.5 = 0$

.664375"

A = 0

B = -155.2222

C = -636.869

D = -444.5245

7.875"

E = 1663.5695

F = 1835.3029

.625"

G = 1806.4773

7.875"

H = 1450.0753

I = 1427.4148

1.890625

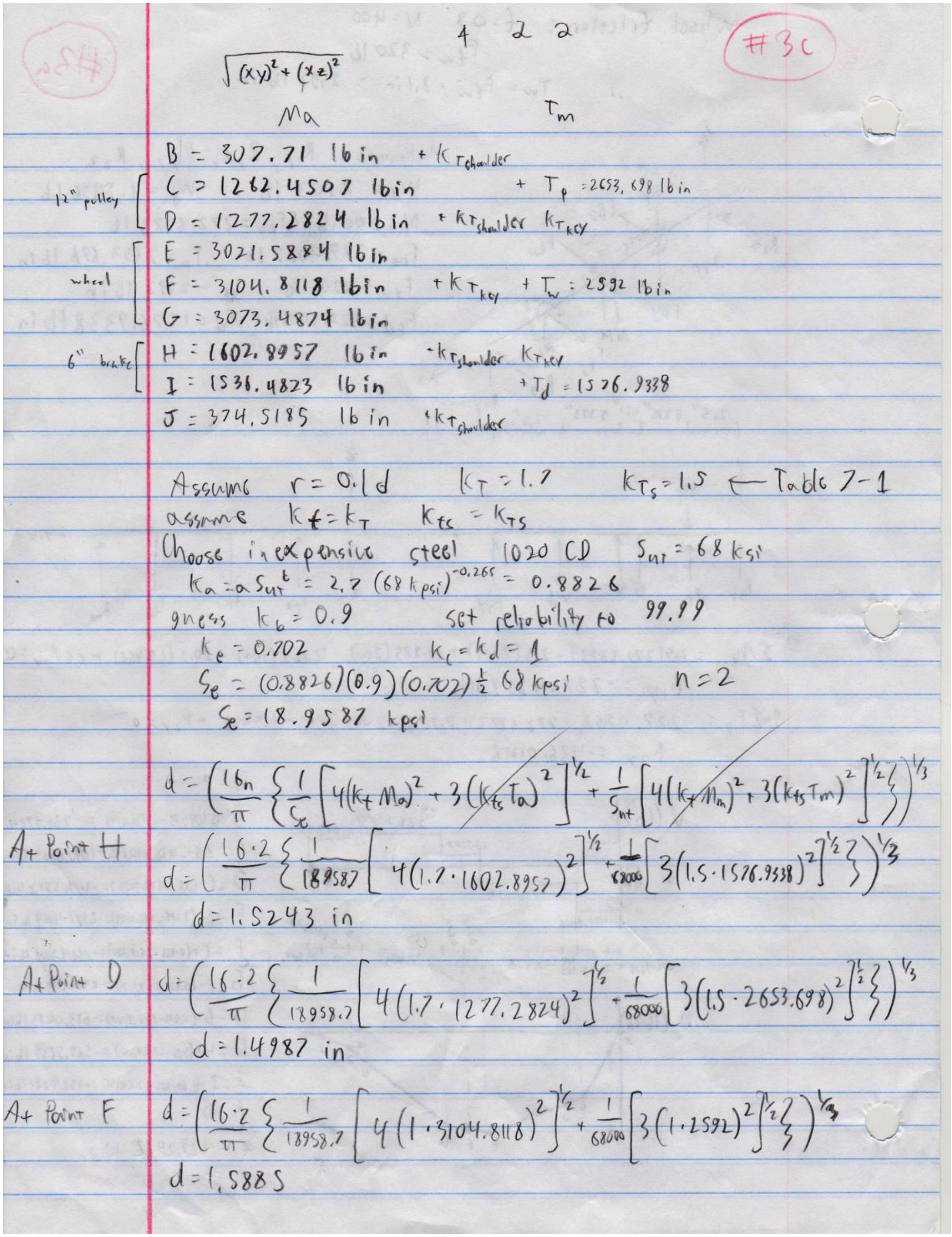
J = 347.4832

K = 0

brake Key

$* = H + .235(-45.3204) = \text{~~1444.8222~~}$

$= 1440.0249$



4 2 2

#3c

$\sqrt{(xy)^2+(xz)^2}$

M_a T_m

B = 307.71 lb in + $K_{T_{shoulder}}$

12" pulley
C = 1282.4507 lbin + T_p = 2653.698 lb in
D = 1277.2824 lb in + $K_{T_{shoulder}}$ $K_{T_{key}}$

wheel
E = 3021.5884 lbin
F = 3104.8118 lbin + $K_{T_{key}}$ + T_w = 2592 lbin
G = 3073.4874 lbin

6" brake
H = 1602.8957 lb in + $K_{T_{shoulder}}$ $K_{T_{key}}$
I = 1536.4823 lb in + T_d = 1576.9338
J = 374.5185 lb in + $K_{T_{shoulder}}$

Assume $r = 0.1d$ $K_T = 1.7$ $K_{T_s} = 1.5$ ← Table 7-1

assume $K_f = K_T$ $K_{fs} = K_{TS}$

Choose inexpensive steel 1020 CD $S_{ut} = 68$ ksi

$K_a = a S_{ut}^b = 2.7 (68 \text{ kpsi})^{-0.265} = 0.8826$

guess $K_b = 0.9$ set reliability to 99.99

$K_e = 0.702$ $K_c = K_d = 1$

$S_e = (0.8826)(0.9)(0.702)\frac{1}{2} 68$ kpsi $n = 2$

$S_e = 18.9587$ kpsi

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

At Point H

$$d = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{189587}\left[4(1.7 \cdot 1602.8957)^2\right]^{1/2} + \frac{1}{68000}\left[3(1.5 \cdot 1576.9338)^2\right]^{1/2}\right\}\right)^{1/3}$$

$d = 1.5243$ in

At Point D

$$d = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{18958.7}\left[4(1.7 \cdot 1277.2824)^2\right]^{1/2} + \frac{1}{68000}\left[3(1.5 \cdot 2653.698)^2\right]^{1/2}\right\}\right)^{1/3}$$

$d = 1.4987$ in

At Point F

$$d = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{18958.7}\left[4(1 \cdot 3104.8118)^2\right]^{1/2} + \frac{1}{68000}\left[3(1 \cdot 2592)^2\right]^{1/2}\right\}\right)^{1/3}$$

$d = 1.5885$

#4

r/d ratio

weaker of two symmetric diameters, expect larger d at F (and L2 will satisfy)

From point H calculations

Table A-17 next lower standard size $d = 1.5$ in

$k_b = \left(\frac{d}{0.3}\right)^{-0.107} = \left(\frac{1.5}{0.3}\right)^{-0.107} = 0.8418$

$S_e = (0.8826)(0.8418)(0.702)\frac{1}{2}68\text{kpsi}$

$S_e = 17.2333$ kpsi

From Figure 6-20

$q = 0.8$

From Figure 6-21

$q_s = 0.85$

Figure A-15-8 $\rightarrow$ $K_{ts} = 1.35$

Figure A-15-9 $\rightarrow$ $K_t = 1.6$

$K_f = 1 + q(K_T - 1) \rightarrow K_f = 1.48$

$K_{fs} = 1 + q_s(K_{ts} - 1) \rightarrow K_{fs} = 1.2975$

$d = 1.4822 \dashrightarrow K_{b_{new}} = 0.8428 \approx 0.8418 \therefore$ use 1.5'' standard

$\sigma_a' = \frac{32 K_f M_a}{\pi d^3} = \frac{32 \cdot 1.48 \cdot 1602.8957}{\pi \cdot 1.5^3} = 7159.6750$ psi

$\sigma_m' = \left[3\left(\frac{16 K_{fs} T_m}{\pi d^3}\right)^2\right]^{1/2} = \left[3\left(\frac{16 \cdot 1.2975 \cdot 1576.9338}{\pi \cdot 1.5^3}\right)^2\right]^{1/2} = 5347.8324$ psi

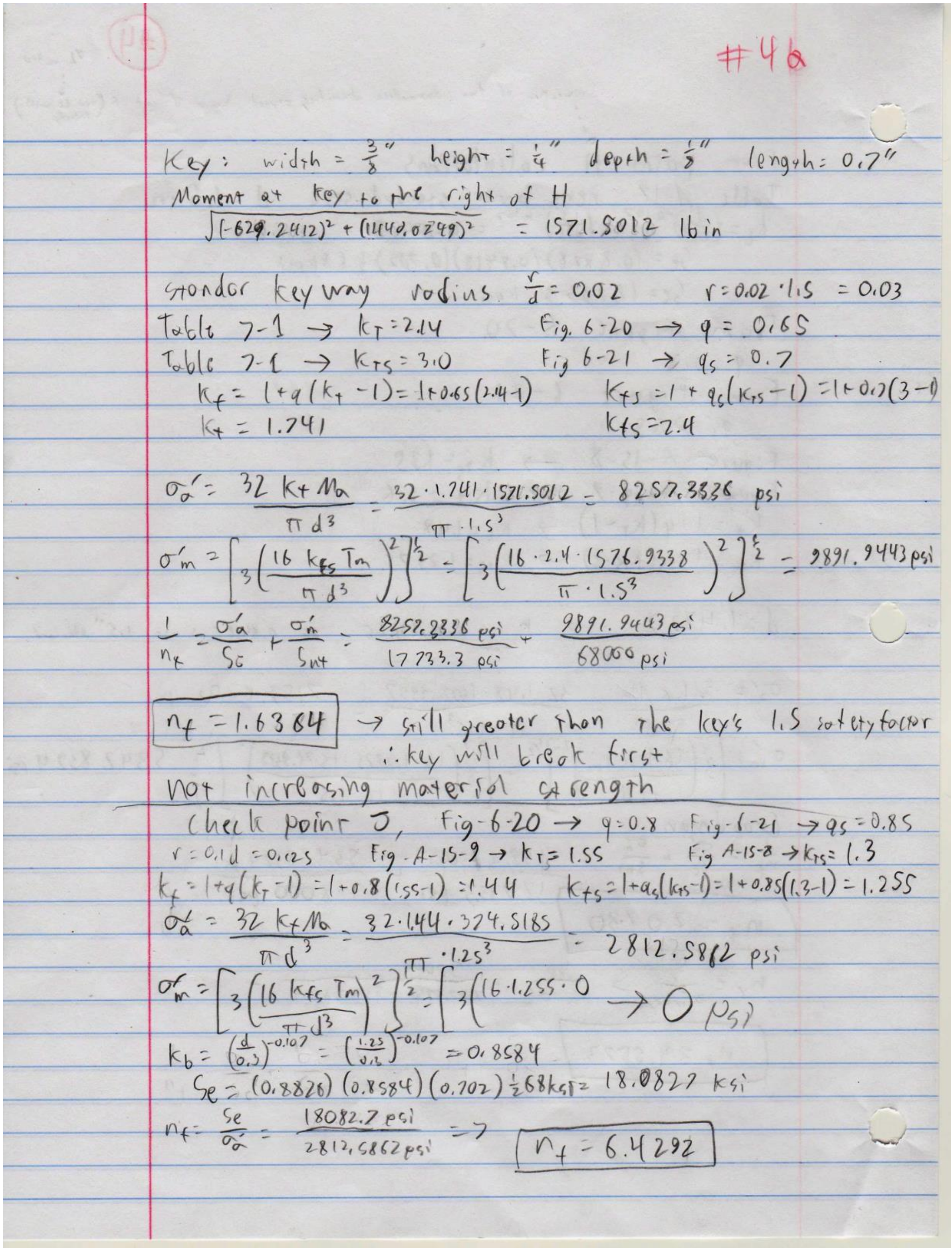
Goodman

$\frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{7159,6750\text{psi}}{17733,3\text{ psi}} + \frac{5347,8324\text{ psi}}{68000\text{ psi}}$

$\boxed{n_f = 2.0730}$

$n_y = \frac{S_y}{\sigma_{max}'} > \frac{S_y}{\sigma_a' + \sigma_m'} = \frac{57000}{7159,6750 + 5347,8324} =$

$\boxed{n_y = 4.5573}$



#4a

Key: width = $\frac{3}{8}$" height = $\frac{1}{4}$" depth = $\frac{1}{8}$" length = 0.7"

Moment at key to the right of H

$\sqrt{(-629.2412)^2 + (1440.0249)^2} = 1571.5012$ lbin

standar keyway radius $\frac{r}{d} = 0.02$ $r = 0.02 \cdot 1.5 = 0.03$

Table 7-1 → $k_T = 2.14$ Fig. 6-20 → $q = 0.65$

Table 7-1 → $k_{TS} = 3.0$ Fig 6-21 → $q_s = 0.7$

$k_f = 1 + q(k_t - 1) = 1 + 0.65(2.14 - 1)$ $k_{fs} = 1 + q_s(k_{ts} - 1) = 1 + 0.7(3 - 1)$

$k_f = 1.741$ $k_{fs} = 2.4$

$$\sigma_a' = \frac{32 k_f M_a}{\pi d^3} = \frac{32 \cdot 1.741 \cdot 1571.5012}{\pi \cdot 1.5^3} = 8257.3336 \text{ psi}$$

$$\sigma_m' = \left[3\left(\frac{16 k_{fs} T_m}{\pi d^3}\right)^2\right]^{\frac{1}{2}} = \left[3\left(\frac{16 \cdot 2.4 \cdot 1576.9338}{\pi \cdot 1.5^3}\right)^2\right]^{\frac{1}{2}} = 9891.9443 \text{ psi}$$

$$\frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{8257.3336 \text{ psi}}{17733.3 \text{ psi}} + \frac{9891.9443 \text{ psi}}{68000 \text{ psi}}$$

$\boxed{n_f = 1.6364}$ → still greater than the key's 1.5 safety factor

∴ key will break first

not increasing material strength

Check point J, Fig-6-20 → $q = 0.8$ Fig-6-21 → $q_s = 0.85$

$r = 0.1d = 0.125$ Fig-A-15-9 → $k_T = 1.55$ Fig A-15-8 → $k_{Ts} = 1.3$

$k_f = 1 + q(k_T - 1) = 1 + 0.8(1.55 - 1) = 1.44$ $k_{fs} = 1 + q_s(k_{ts} - 1) = 1 + 0.85(1.3 - 1) = 1.255$

$$\sigma_a' = \frac{32 k_f M_a}{\pi d^3} = \frac{32 \cdot 1.44 \cdot 374.5185}{\pi \cdot 1.25^3} = 2812.5862 \text{ psi}$$

$$\sigma_m' = \left[3\left(\frac{16 k_{fs} T_m}{\pi d^3}\right)^2\right]^{\frac{1}{2}} = \left[3\left(16 \cdot 1.255 \cdot 0 \rightarrow 0 \text{ psi}\right.\right.$$

$$k_b = \left(\frac{d}{0.3}\right)^{-0.107} = \left(\frac{1.25}{0.3}\right)^{-0.107} = 0.8584$$

$S_e = (0.8826)(0.8584)(0.702)\frac{1}{2}68\text{ksi} = 18.0827$ ksi

$$n_f = \frac{S_e}{\sigma_a'} = \frac{18082.7 \text{ psi}}{2812.5862 \text{ psi}} \Rightarrow \boxed{n_f = 6.4292}$$

#5

Point F check

k_T and $k_{TS} = 1$ so $k_f = k_{fs} = 1$

$k_b = \left(\frac{1.8}{0.3}\right)^{-0.107} = 0.8255$

$S_e = (0.8826)(0.8256)(0.702)\frac{1}{2}68\,\text{ksi} = 17.3907$ ksi

$\sigma_a' = \frac{32\ k_f\ M_a}{\pi d^3} = \frac{32 \cdot 1 \cdot 3104.8118}{\pi \cdot 1.8^3} = 5422.7286$ psi

$\sigma_m' = \left[3\left(\frac{16 \cdot 1 \cdot 2592}{\pi \cdot 1.8^3}\right)^2\right]^{\frac{1}{2}} = 3920.5610$ psi

$\frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{UT}} = \frac{5422.7286\,\text{psi}}{17390.7\ \text{psi}} + \frac{3920.5610\ \text{psi}}{68000\ \text{psi}} = \boxed{n_f = 2.7066}$

Life calculation   Fig. 6-18 → $f = 0.9$

$a = \frac{(f S_{ut})^2}{S_e} = \frac{(0.9 \cdot 68000)^2}{17733.3\ \text{psi}} = 211209.4196$ psi

$b = -\frac{1}{3}\log\left(\frac{f\ S_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{0.9 \cdot 68}{17.7333}\right) = -0.1793$

$N = \left(\frac{\sigma_{rev}}{a}\right)^{1/b}$ set $\sigma_{rev} = \sigma_a'$

$= \left(\frac{8257.3336}{211209.4196}\right)^{1/-0.1793}$

$N = 7.1130 \cdot 10^7$

cycles we need

5 years · 50 weeks · 5 days · 3 hours · 60 min

225000 minutes · 237.5 rpm

$N_{required} = 5.3438 \cdot 10^7$ cycles

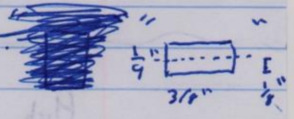
Brake Key

#6

*assume key material = UNS G10180 -- $S_y = 54$ ksi

D = 1.5" between $1\frac{3}{8}$" and $1\frac{3}{4}$"

From Table 7-6 choose $\frac{3}{8}$" x $\frac{1}{4}$" Key



$$\tau_{max} = \frac{\text{Force}}{\text{Area}_{\text{key}}} = \frac{\text{Force}}{\underset{\text{(width)}}{.375"} \cdot \text{Length}}$$

$$\text{Force} = \frac{T}{r} = \frac{\frac{63000\,(10\text{hp})}{237.5\text{rpm}}}{\frac{1.5\text{in}}{2}} = 3536.8421 \text{ lb}$$

$$\tau_{max} = \frac{\text{Key material property}}{\text{safety factor}} = \frac{\overset{\text{(distortion-energy theory)}}{0.577 S_y}}{n} = \frac{0.577(54)}{1.5} = 20.772 \text{ ksi}$$

(1.5 ← designing to fail before shaft)

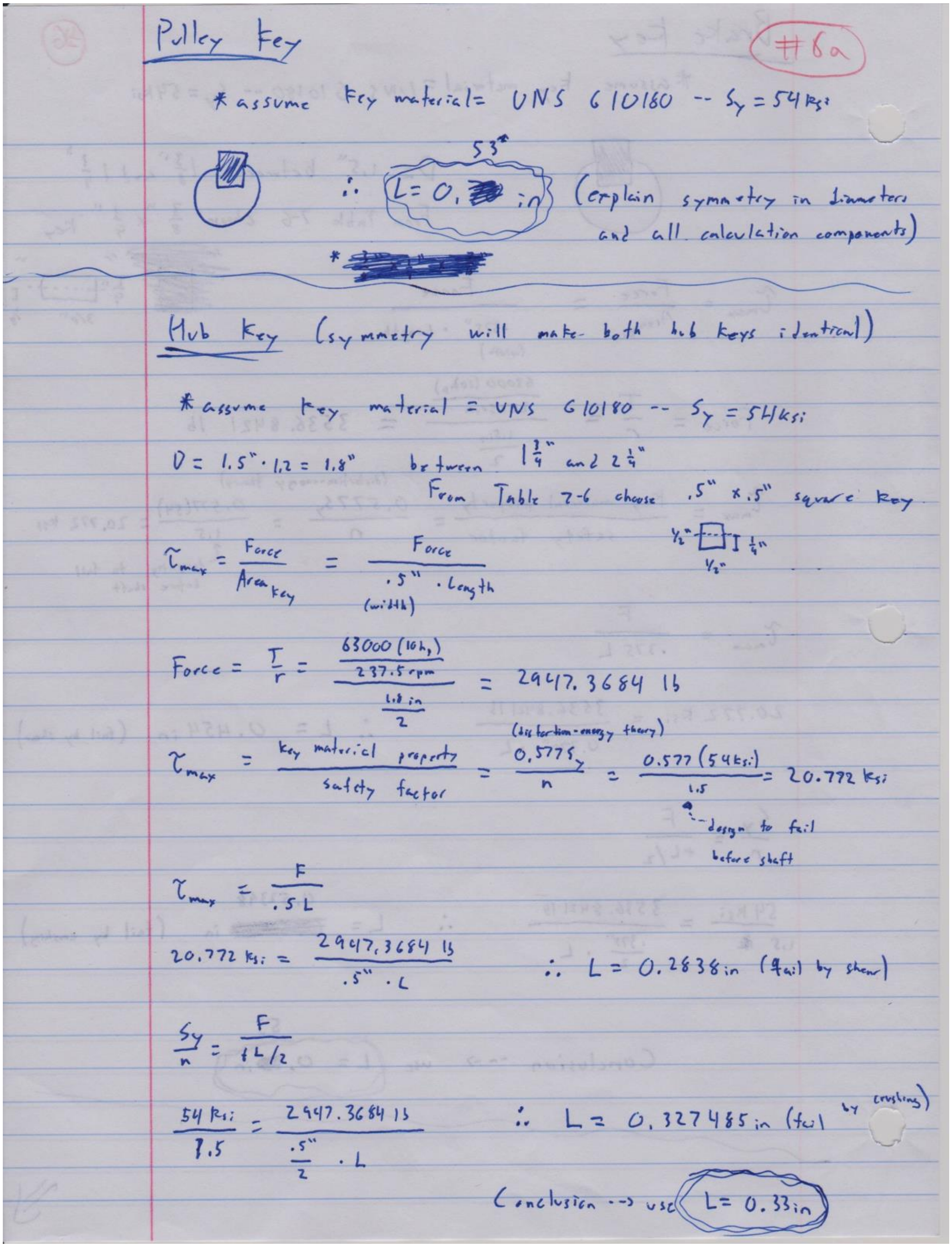
$$\tau_{max} = \frac{F}{.375\,L}$$

$$20.772 \text{ ksi} = \frac{3536.8421 \text{ lb}}{0.375" \cdot L} \qquad \therefore L = 0.454 \text{ in (fail by shear)}$$

$$\frac{S_y}{n} = \frac{F}{tL/2}$$

$$\frac{54 \text{ ksi}}{1.5} = \frac{3536.8421 \text{ lb}}{\frac{.375"}{2} \cdot L} \qquad \therefore L = \overset{0.52348}{\text{[illegible]}} \text{ in (fail by crushing)}$$

Conclusion --> use $\boxed{L = 0.\overset{53}{\text{[illegible]}} \text{ in}}$



Pulley Key

#6a

* assume key material = UNS G10180 -- $S_y = 54$ ksi

53*

∴ $\boxed{L = 0.[redacted]\text{ in}}$ (explain symmetry in diameters and all calculation components)

* [redacted]

Hub Key (symmetry will make both hub keys identical)

* assume key material = UNS G10180 -- $S_y = 54$ ksi

$D = 1.5" \cdot 1.2 = 1.8"$ between $1\frac{3}{4}"$ and $2\frac{1}{4}"$

From Table 7-6 choose .5" x .5" square key.

½" ¼"
½"

$$\tau_{max} = \frac{\text{Force}}{\text{Area}_{key}} = \frac{\text{Force}}{\underset{(width)}{.5"} \cdot \text{Length}}$$

$$\text{Force} = \frac{T}{r} = \frac{\frac{63000\,(10hp)}{237.5\text{ rpm}}}{\frac{1.8\text{ in}}{2}} = 2947.3684\text{ lb}$$

(distortion-energy theory)

$$\tau_{max} = \frac{\text{key material property}}{\text{safety factor}} = \frac{0.577 S_y}{n} = \frac{0.577(54\text{ksi})}{1.5} = 20.772\text{ ksi}$$

design to fail before shaft

$$\tau_{max} = \frac{F}{.5L}$$

$$20.772\text{ ksi} = \frac{2947.3684\text{ lb}}{.5" \cdot L}$$

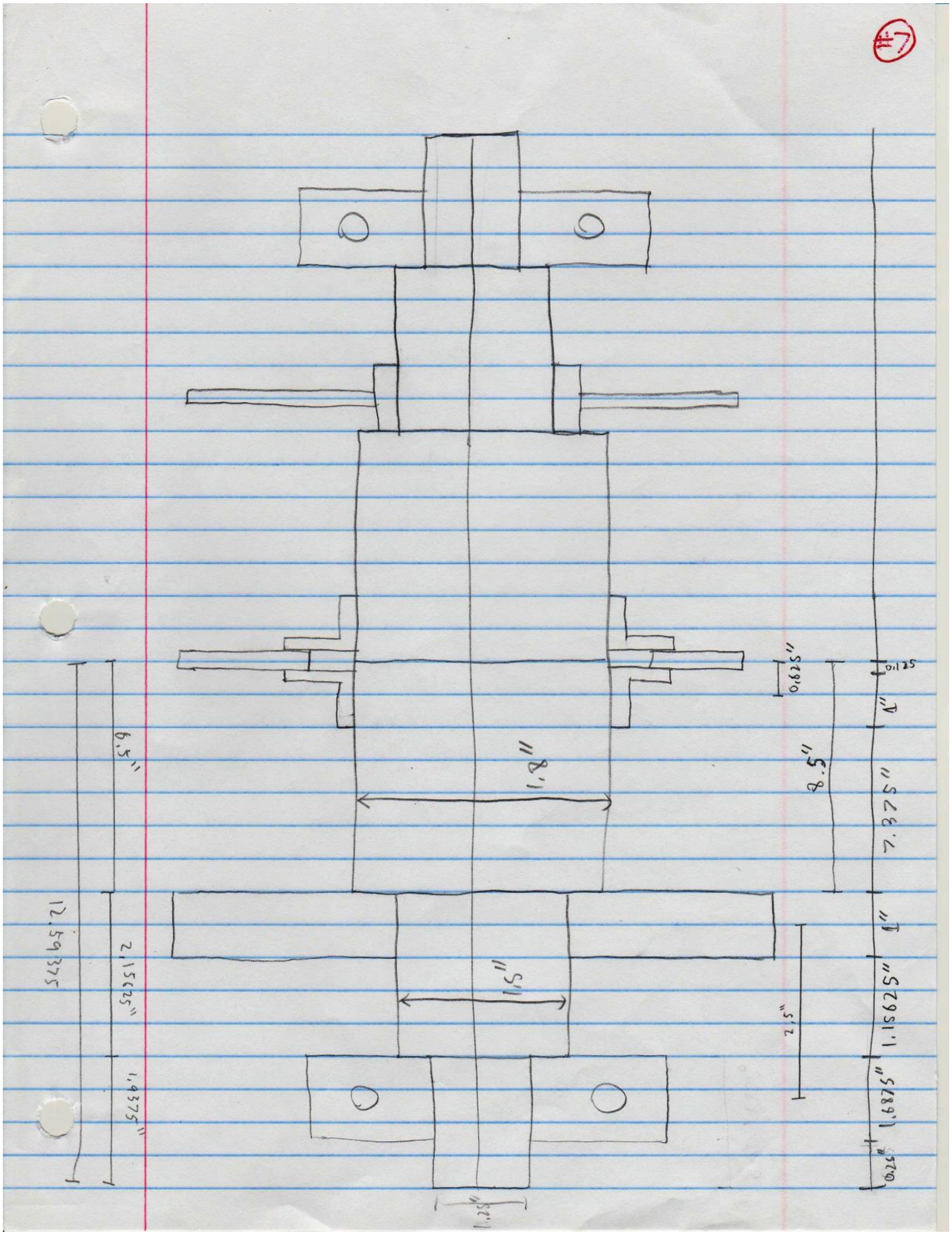
∴ L = 0.2838 in (fail by shear)

$$\frac{S_y}{n} = \frac{F}{tL/2}$$

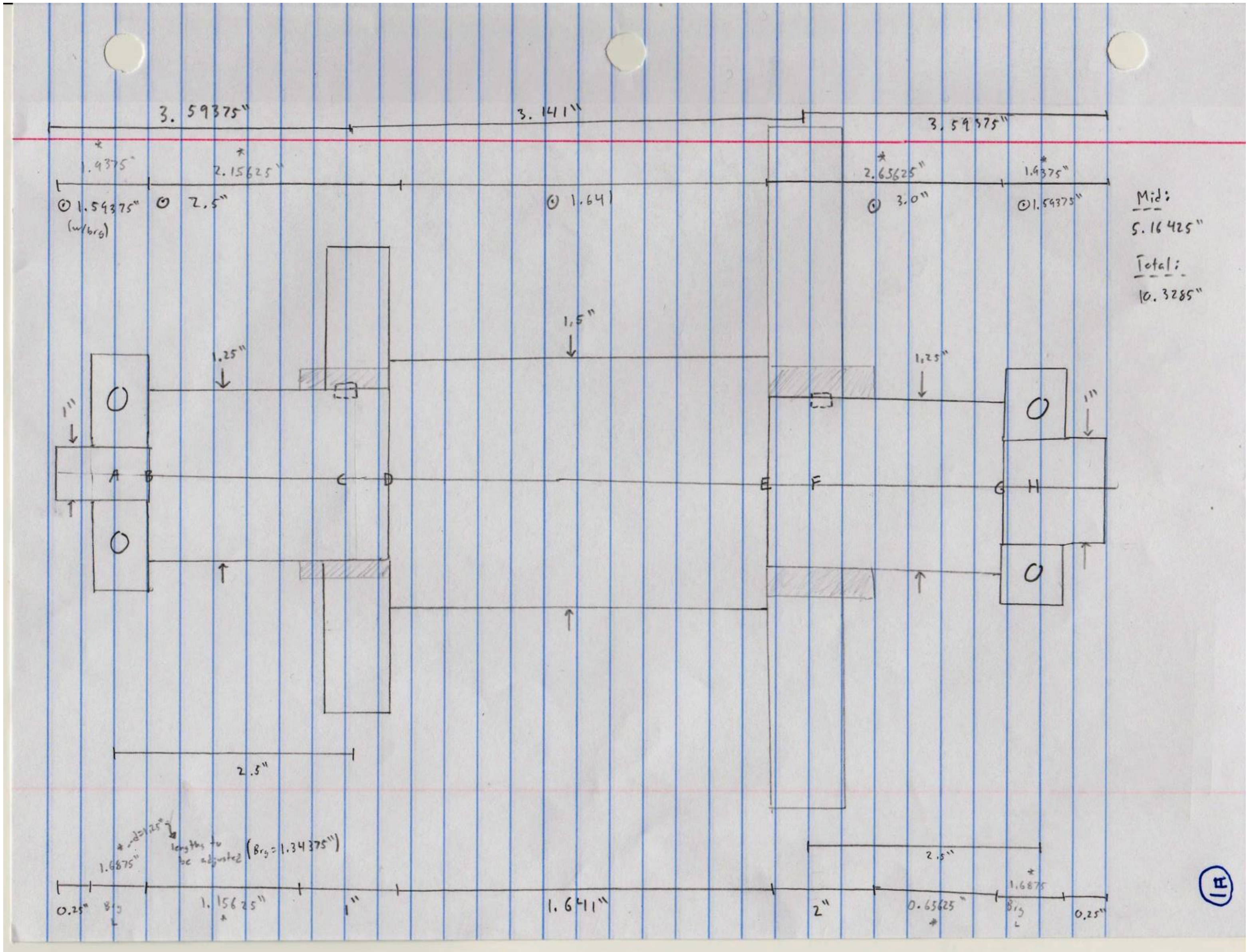
$$\frac{54\text{ ksi}}{1.5} = \frac{2947.3684\text{ lb}}{\frac{.5"}{2} \cdot L}$$

∴ L = 0.327485 in (fail by crushing)

Conclusion --> use $\boxed{L = 0.33\text{in}}$

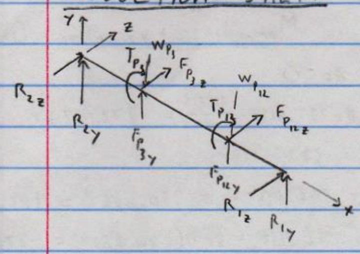

#7
12.59375
6.5"
2.15625"
1.9375"
18"
15"
1.25"
0.625"
8.5"
2.5"
0.125
1"
7.375"
1"
1.15625"
1.6875"
0.25"

“Reduction Shaft Calculations”



(12)

Reduction shaft



Unknowns : R_{1y} , R_{1z} , R_{2y} , R_{2z}

$W_{P_{12}} = 10.266 + 1.203\text{ lb} = 11.469\text{ lb}$

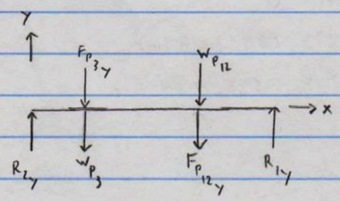
$W_{P_3} = 1.297 + 1.203\text{ lb} = 2.500\text{ lb}$

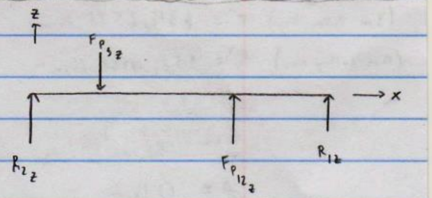
$F_{P_{3y}} = -431.8753\text{ lb}$ $F_{P_{3z}} = -410.0120\text{ lb}$

$F_{P_{12y}} = -110.5893\text{ lb}$ $F_{P_{12z}} = 504.6673\text{ lb}$

$T_{P_3} = -410.0120\text{ lb} \cdot \frac{3\text{ in}}{2}$ $T_{P_{12}} = 504.6673\text{ lb} \cdot \frac{12\text{ in}}{2}$

$= -615.0179\text{ lbin}$ $= 3028.0039\text{ lbin}$





$\circlearrowleft + \Sigma M_{R_2} : -2.5''(431.8753) - 2.5''(2.500) - 5.641(110.5893) - 5.641(11.469) + 8.141 R_{1y} = 0$

$\therefore R_{1y} = 217.9670\text{ lb} \uparrow$

$\circlearrowleft + \Sigma M_{R_2} : -2.5''(410.0120) + 5.641(504.6673) + 8.141 R_{1z} = 0$

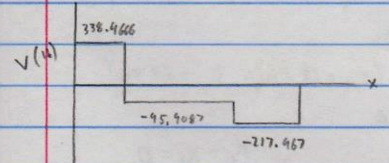
$\therefore R_{1z} = -223.7806\text{ lb} \downarrow$

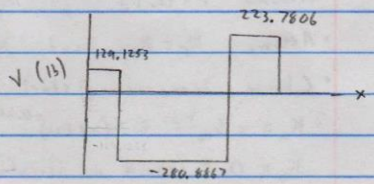
$+\uparrow \Sigma F_y : R_{2y} - 431.8753 - 2.5 - 110.5893 - 11.469 + 217.9670 = 0$

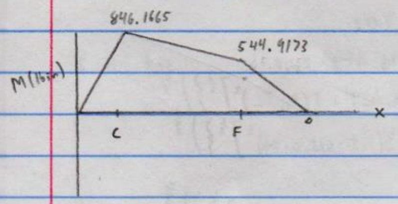
$\therefore R_{2y} = 338.4666\text{ lb} \uparrow$

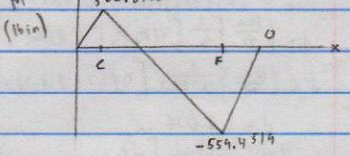
$+\uparrow \Sigma F_z : R_{2z} - 410.0120 + 504.6673 - 223.7806 = 0$

$\therefore R_{2z} = 129.1253\text{ lb} \uparrow$









#2a

Moments in xy plane (lbin)

$A = 0$

$B = 338.4666 \cdot \frac{1.6875}{2} = 285.5812$

$C = 846.1665$

$D = 846.1665 - .5(95.9087) = 798.2122$

$E = 846.1665 - 2.141(95.9087) = 640.8260$

$F = 544.9173$

$G = 544.9173 - 1.65625(217.467) = 183.9095$

$H = 0$

*see later
(3 in Pulley key) $C' = 834.6575$ lbin

(12 in pulley key) $F' = 556.4263$ lbin

Moments in xz plane (lbin)

$A = 0$

$B = 322.8133 \cdot \frac{1.6875}{2} = 272.3737$

$C = 322.8133$

$D = 322.8133 - .5(280.8867) = 182.3700$

$E = 322.8133 - 2.141(280.8867) = -278.5651$

$F = -559.4519$

$G = -559.4519 + 1.65625(223.7806) = -188.8153$

$H = 0$

$C' = 289.1069$ lbin

$F' = -525.7454$ lbin

Combined loading $\left(\sqrt{xy^2 + xz^2}\right)$

$A = 0$ lbin

$B = 394.6442$ lbin $+K_{t_{shoulder}}$

3" pulley [$C = 905.6523$ lbin $+T_{p_3}$ $+K_{t_{key}}$

$D = 818.7805$ lbin $+T_{p_3} + K_{t_{shoulder}}$

12" pulley [$E = 698.7535$ lb in $+T_{p_{12}}$ $+K_{t_{shoulder}}$

$F = 780.9746$ lbin $+T_{p_{12}}$ $+K_{t_{key}}$

$G = 263.5791$ lbin

$H = 0$ lbin

$C' = 883.3096$ lbin

$F' = 765.5185$ lbin

- Assume $r = 0.1d$, $K_T = 1.7$ and $K_{Ts} = 1.5$ (Table 7-1)
- Assume $K_f = K_T$ and $K_{fs} = K_{Ts}$
- Choose inexpensive steel 1020 CD ($S_{ut} = 68$ ksi, $S_y = 57$ ksi)

$k_a = a S_{ut}^{\,b} = 2.7(68\,\text{ksi})^{-0.265} = 0.8826$

$k_b = 0.9$ (check w/ iterate)

$k_c = k_d = 0$

reliability $= 99.99\%$ $\therefore$ $k_e = 0.702$

$S_e = (0.8826)(0.9)(0.702)(\frac{1}{2} \cdot 68\,\text{kpsi}) = 18.9587$ kpsi

- set $n = 2$

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

At point D: $$d = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{18958.7}\left[4(1.7 \cdot 818.7805)^2\right]^{1/2} + \frac{1}{68000}\left[3(1.5 \cdot 615.0179)^2\right]^{1/2}\right\}\right)^{\frac{1}{3}}$$

$d = 1.2016$ in

At point E: $$d = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{18958.7}\left[4(1.7 \cdot 698.7535)^2\right]^{1/2} + \frac{1}{68000}\left[3(1.5 \cdot 3028.0039)^2\right]^{1/2}\right\}\right)^{1/3}$$

$d = 1.3490$ in

#3

From point E calculations (weaker of two symmetric diameters)

$d_{est} = 1.25 in$ (Table A-17 for next lower standard size)

Update assumptions from first iteration:

$$K_b = \left(\frac{d}{0.3}\right)^{-0.107} = \left(\frac{1.25}{0.3}\right)^{-0.107} = 0.8584$$

$$S_e = (0.8826)(0.8584)(0.702)(.5 \cdot 68 \text{ kpsi}) = 18.0827 \text{ kpsi}$$

$q \approx 0.8$ (Figure 6-20 w/ $r = 0.125''$ and $S_{ut} = 68 k_{psi}$)

$q_s \approx 0.85$ (Figure 6-21 w/ $r = 0.125''$ and $S_{ut} = 68 k_{psi}$)

$K_T \approx 1.6$ (Figure A-15-9 w/ $r/d = .1$ and $D/d = 1.2$)

$K_{Ts} \approx 1.35$ (Figure A-15-8 w/ $r/d = .1$ and $D/d = 1.2$)

$K_f = 1 + q(K_T - 1) = 1 + 0.8(1.6 - 1) = 1.48$

$K_{fs} = 1 + q_s(K_{Ts} - 1) = 1 + 0.85(1.35 - 1) = 1.2975$

$$d_{new} = \left(\frac{16 \cdot 2}{\pi}\left\{\frac{1}{18082.7}\left[4(1.48 \cdot 698.7535)^2\right]^{\frac{1}{2}} + \frac{1}{68000}\left[3(1.5 \cdot 3028.0039)^2\right]^{\frac{1}{2}}\right\}\right)^{\frac{1}{3}}$$

$d_{new} = 1.3283$ in $\dashrightarrow$ $K_{b_{new}} = 0.8528 \approx 0.8584$ $\therefore$ use 1.25" standard

$$\sigma_A' = \frac{32 K_f M_a}{\pi d^3} = \frac{32(1.48)(698.7535)}{\pi(1.25)^3} = 5393.3149 \text{ psi}$$

$$\sigma_m' = \left[3\left(\frac{16 K_{fs} T_m}{\pi d^3}\right)^2\right]^{\frac{1}{2}} = \left[3\left(\frac{16(1.2975)(3028.0039)}{\pi(1.25)^3}\right)^2\right]^{\frac{1}{2}} = 17744.5298 \text{ psi}$$

Goodman criterion for safety against fatigue

$$\frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{5393.3149}{18082.7} + \frac{17744.5298}{68\,000} \quad \therefore \boxed{n_f = 1.7882}$$

Check against yielding

$$n_y = \frac{S_y}{\sigma_{max}'} > \frac{S_y}{\sigma_A' + \sigma_m'} = \frac{57000}{5393.3149 + 17744.5298} \quad \therefore \boxed{n_y = 2.4635}$$

- Safety factors lower than desired so increase material to 1050 CD steel ($S_{ut} = 100$ kpsi, $S_y = 84$ kpsi)

$$\frac{1}{n_{f_{new}}} = \frac{5393.3149 \text{ psi}}{(0.8826)(0.8584)(0.702)(.5 \cdot 100 \text{ kpsi})} + \frac{17744.5298 \text{ psi}}{100\,000 \text{ psi}} \quad \therefore \boxed{n_{f_{new}} = 2.6298}$$

$$n_y = \frac{84\,000}{5393.3149 + 17744.5298} \quad \therefore \boxed{n_{y_{new}} = 3.6304}$$

closer to drive shaft safety and should anticipate key stress concentration

↘

#13a

12" pulley key : width = 1/4" height = 1/4". depth = 1/8" length = 0.24"
(at point F)

Moment at key to the right of point E :
xy plane = 640.8260 − 95.9087(0.88") = 556.4263 lbin
xz plane = −278.5651 − 280.8867(0.88") = −525.7454 lbin

combined = $\sqrt{x_y^2 + x_z^2}$ = 765.5185 lbin

* Note → similar process gives combined at left of D as 883.3096 lbin ∴ Check both

- Standard keyway radius ($\frac{r}{d} = 0.02$ ∴ $r = .02 \cdot 1.25 = 0.025$ in)

$K_T = 2.14$ (Table 7-1) $q \approx 0.7$ (Figure 6-20 w/ r = 0.025" and $S_{ut} = 100$ kpsi)

$K_{T_s} = 3.0$ (Table 7-1) $q_s \approx 0.75$ (Figure 6-21 w/ r = 0.025" and $S_{ut} = 100$ kpsi)

$K_f = 1 + q(K_T - 1) = 1 + 0.7(2.14 - 1) = 1.798$

$K_{f_s} = 1 + q_s(K_{T_s} - 1) = 1 + 0.75(3.0 - 1) = 2.5$

Right of E:

$$\sigma_a' = \frac{32 K_f M_a}{\pi d^3} = \frac{32(1.798)(765.5185)}{\pi(1.25)^3} = 7178.1982 \text{ psi}$$

$$\sigma_m' = \left[3\left(\frac{16 K_{f_s} T_m}{\pi d^3}\right)^2\right]^{\frac{1}{2}} = \left[3\left(\frac{16(2.5)(3028.0034)}{\pi(1.25)^3}\right)^2\right]^{\frac{1}{2}} = 34189.8454 \text{ psi}$$

$$\frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{7178.1982 \text{ psi}}{26592.5968 \text{ psi}} + \frac{34189.8454 \text{ psi}}{100000 \text{ psi}} \quad \therefore \boxed{n_f = 1.6344}$$

* 1.6344 > 1.5 of key therefore key will break first still (also confirms decision to increase material strength as back check w/ 1020 CD would have given $n_f = 1.1345$)

Left of D:

$$\sigma_a' = \frac{32(1.798)(883.3096)}{\pi(1.25)^3} = 8282.7147 \text{ psi}$$

$$\sigma_m' = \left[3\left(\frac{16(2.5)(615.0174)}{\pi(1.25)^3}\right)^2\right]^{\frac{1}{2}} = 6944.2998 \text{ psi}$$

$$\frac{1}{n_f} = \frac{8282.7147 \text{ psi}}{26592.5968 \text{ psi}} + \frac{6944.2998 \text{ psi}}{100000 \text{ psi}} \quad \therefore \boxed{n_f = 2.6253}$$

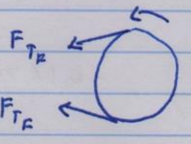
* key on 12in pulley proved to be the more critical as expected

⊙ Fill in other diameters w/ D/d = 1.2

Reduction Shaft Tension forces

#4

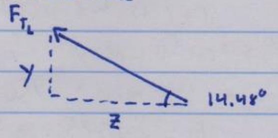
E3 - R12 double (forward)



$F_{T_L} = 39.4705$ lb

$F_{T_U} = 481.7535$ lb

lower (loose)



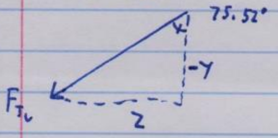
$F_{T_{L_z}} = F_{T_L} \cos 14.48°$

$\therefore F_{T_{L_z}} = 38.2167$ lb

$F_{T_{L_y}} = F_{T_L} \sin 14.48°$

$\therefore F_{T_{L_y}} = 9.8693$ lb

Upper (tight)



$F_{T_{U_z}} = F_{T_U} \sin 75.52°$

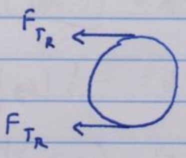
$\therefore F_{T_{U_z}} = 466.4506$ lb

$F_{T_{U_y}} = F_{T_U} \cos 75.52°$

$\therefore F_{T_{U_y}} = -120.4586$ lb

$$\left.\begin{aligned} \therefore F_{T_{z\,tot}} &= 504.6673 \text{ lb} \\ F_{T_{y\,tot}} &= -110.5843 \text{ lb} \end{aligned}\right] \text{- use}$$

M12 - R12 double (reverse)



lower (loose)

$F_{T_{R_L}} = 23.9997$ lb

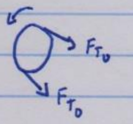
upper (tight)

$F_{T_{R_U}} = 134.5705$ lb

$\therefore F_{T_z} = 158.5702$ lb , $F_{T_y} = 0$ lb

* For double pulley, analyze in forward motion as significantly greater forces

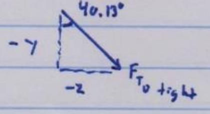
R3 - D12 (forward)



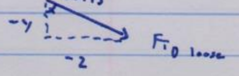
$F_{T_{D\,tight}} = 522.431$ lb

$F_{T_{D\,loose}} = 80.148$ lb

lower (tight)



upper (loose)



$-y_1 = 399.4424$ lb

$-z_1 = 336.7193$ lb

$-y_2 = 32.4329$ lb

$-z_2 = 73.2926$ lb

$\therefore F_{T_{z\,tot}} = -410.0120$ lb , $F_{T_{y\,tot}} = -431.8753$ lb

#4a

Life Calculation

$$a = \frac{(f\,S_{ut})^2}{S_e} = \frac{(0.845 \cdot 100)^2}{26.5926} = 268.5051$$

$$b = -\frac{1}{3}\log\left(\frac{f\,S_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{0.845 \cdot 100}{26.5926}\right) = -0.1674$$

$$N = \left(\frac{\sigma_{rev}}{a}\right)^{1/b} \quad \text{set } \sigma_{rev} = \sigma_a'$$

$$= \left(\frac{7.1782}{268.5051}\right)^{1/-0.1674}$$

$$N = 2.5016 \cdot 10^9$$

cycles we need

5 years · 50 weeks · 5 · days · 3 hours · 60 min

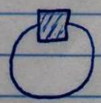
225000 min · 950 rpm

$N = 2.1325 \cdot 10^8$ cycles

Reduction Shaft Keys

#5

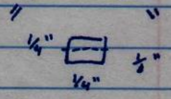
3" pulley key



- Assume Key material = UNS G 10180 ($S_y = 54$ Ksi)

D = 1,25" between $\frac{7}{8}$" and $1\frac{1}{4}$"

From Table 7-6 choose $\frac{1}{4}$" × $\frac{1}{4}$" Key



$$\tau_{max} = \frac{Force}{Area_{key}} = \frac{Force}{.25" \cdot L}$$

$$Force = \frac{T}{r} = \frac{\frac{63000\,(10hp)}{950\,rpm}}{\frac{1,25\,in}{2}} = 1061.0526 \text{ lb}$$

$$\tau_{max} = \frac{\text{Key material property}}{\text{safety factor}} = \frac{0,577\,S_y}{n} = \frac{0,577(54)}{1.5} = 20.772 \text{ ksi}$$

$$\tau_{max} = \frac{F}{.25L}$$

$$20.772 \text{ ksi} = \frac{1061.0526 \text{ lb}}{.25" \cdot L} \qquad \therefore \; L = 0.204\,in \text{ (fail by shear)}$$

$$\frac{S_y}{n} = \frac{F}{tL/2}$$

$$\frac{54\,ksi}{1.5} = \frac{1061.0526 \text{ lb}}{\frac{.25"}{2} \cdot L} \qquad \therefore \; L = 0.236\,in \text{ (fail by crushing)}$$

Conclusion → use $\boxed{L = 0.24\,in}$

* This will be the same key for the 12" pulley as same diameter and no variables change in key calculation

“Axle Calculations”

Axle Calculations

#1

ID Pulley = 1.816 in = 46.1264

Pulley width : 0.875 in

pulley weight = 1.217 lbf

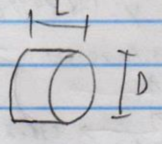
choose 47mm bearing

41.85039 in

bearing ID = 30 mm = 1.1811 in

width = 9 mm = 0.354331 in

Axle Design

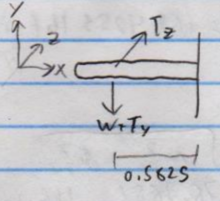


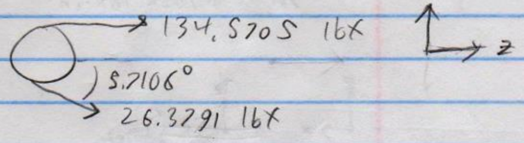
D = 30mm = 1.1811 in

L = 1 in

unscaled drawings below

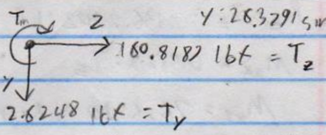
Top pulley





components $x: 26.3791 \cos(5.7106) = 26.2482$

$y: 26.3291 \sin(5.7106) = 2.6248$

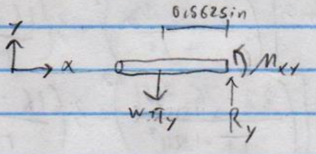


160.8187 lbf $= T_z$

2.6248 lbf $= T_y$

T = Torque = 1.5(134.5705 − 26.3791)

$T_m = 162.2871$ lbf·in



$R_y = W + T_y = 3.9218$ lbf

$\Sigma M = (-0.5625 \text{ in} \cdot -3.9218 \text{ lbf}) + M_{xy} = 0$

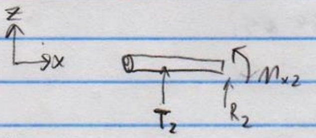
$M_{xy} = 2.2060$ lbf·in

$M_m = \sqrt{M_{xy}^2 + M_{xz}^2}$

$M_m = 90.4874$ lbf·in

$$\sigma'_m = \left[\left(\frac{32 K_f M_m}{\pi d^3}\right)^2 + 3\left(\frac{16 K_{fs} T_m}{\pi d^3}\right)\right]^{1/2}$$

$$\sigma'_m = \left[\left(\frac{32 \cdot 90.4874}{\pi \cdot 1.1811^3}\right)^2 + 3\left(\frac{16 \cdot 162.2871}{\pi \cdot 1.1811^3}\right)^2\right]^{1/2}$$



$R_z = T_z = 160.8187$ lbf

$\Sigma M = (-0.5625 \cdot 160.8187) + M_{xz} = 0$

$M_{xz} = 90.4605$ lbf·in

$\sigma'_m = 1033.3787$ psi $\quad \sigma'_a = 0$

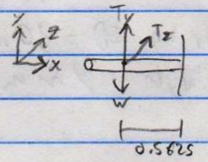
$n_y = \frac{S_y}{\sigma'_a + \sigma'_m} = 71020$ CD $\quad S_y = 57000$ psi

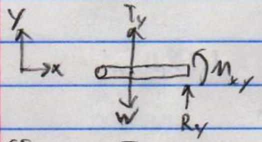
$n_y = \frac{57000 \text{ psi}}{1033.3787 \text{ psi}}$

$\boxed{n_y = 55.1589}$

#2

Bottom Pulley



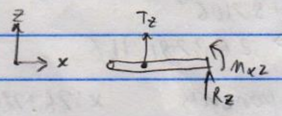


$\Sigma F_y = R_y + T_y - W = 0$

$R_y = -12.33$ lbf

$\Sigma M = M_{xy} - 0.5625 (T_y - w) = 0$

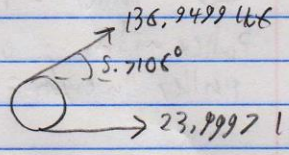
$M_{xy} = 6.9356$ lbf·in



$R_z = -T_z = -136.2702$ lbf

$M_{xz} = 0.5625 \cdot T_z$

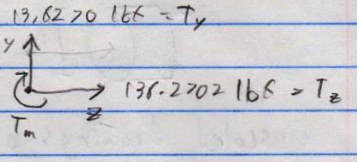
$M_{xz} = 76.6520$ lbf·in



components

$x: 136.9499 \cos(5.7106) = 136.2702$ lbf

$y: 136.9499 \sin(5.7106) = 13.6270$ lbf



$T_m = 1.5(136.9499 - 23.9997)$

$T_m = 169.4253$ lbf·in

$M_m = \sqrt{M_{xy}^2 + M_{xz}^2}$

$M_m = 76.9651$ lbf·in

$$\sigma'_m = \left[\left(\frac{32 k_f M_m}{\pi d^3}\right)^2 + 3\left(\frac{16 k_{fs} T_m}{\pi d^3}\right)^2\right]^{1/2}$$

$$\sigma'_m = \left[\left(\frac{32 \cdot 76.9651}{\pi \cdot 1.1811^3}\right)^2 + 3\left(\frac{16 \cdot 169.4253}{\pi \cdot 1.7811^3}\right)^2\right]^{1/2}$$

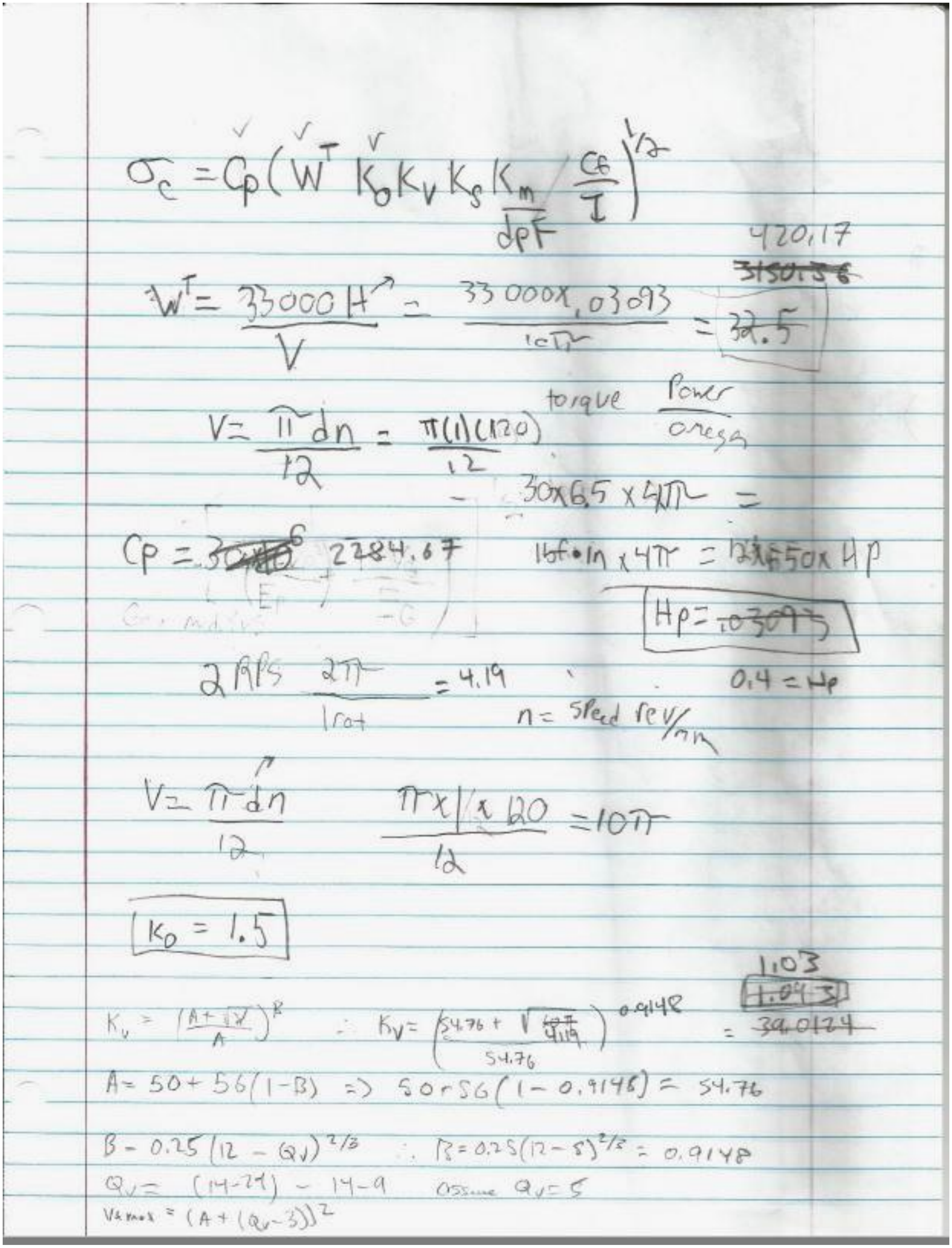
$\sigma'_m = 1024.3059$ psi

1020 CD steel → $S_y = 57000$ psi

$$n_y = \frac{S_y}{\sigma'_m + \sigma'_a} = \frac{S_y}{\sigma'_m} = \frac{57000 \text{ psi}}{1024.3059 \text{ psi}}$$

$$\boxed{n_y = 55.6474}$$

“Steering Gear Calculations”



$$\sigma_c = C_p\left(W^T K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}\right)^{1/2}$$

420.17

~~3150.56~~

$$W^T = \frac{33000H}{V} = \frac{33000\times .03093}{10\pi} = \boxed{32.5}$$

$$V = \frac{\pi d n}{12} = \frac{\pi(1)(120)}{12}$$

torque $\frac{\text{Power}}{\text{omega}}$

$30\times 6.5\times 4\pi =$

$C_p =$ ~~30~~ 2284.67

lbf·in × 4π = [illegible]50 × Hp

$\boxed{Hp = \cancel{.03093}}$

$2\text{ RPS}\ \frac{2\pi}{1\text{ rot}} = 4.19$

0.4 = Hp

n = speed rev/min

$$V = \frac{\pi d n}{12} \quad \frac{\pi\times 1\times 120}{12} = 10\pi$$

$\boxed{K_o = 1.5}$

1.03

$\boxed{\cancel{1.043}}$

$= \cancel{39.0124}$

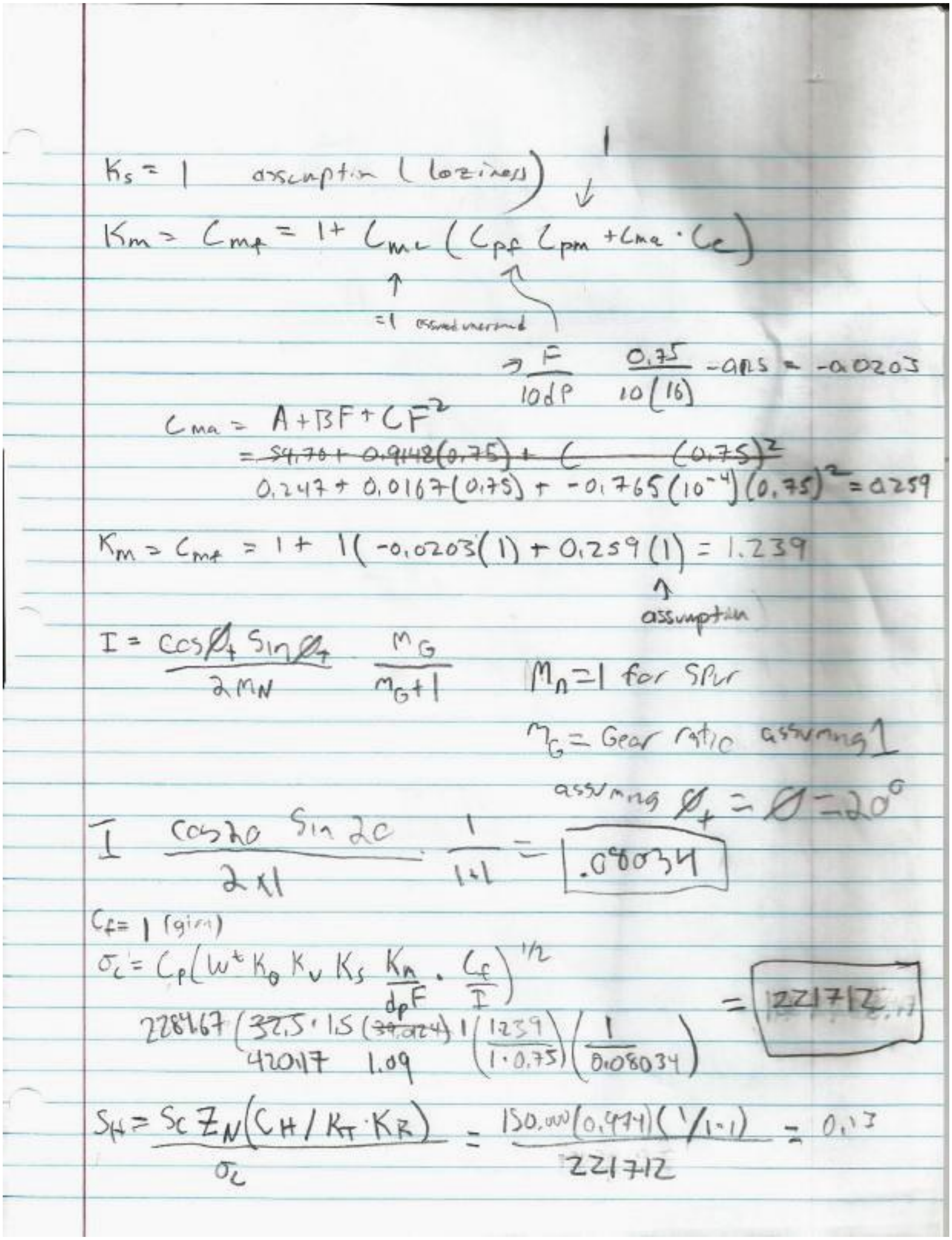
$$K_v = \left(\frac{A+\sqrt{V}}{A}\right)^B \therefore K_v = \left(\frac{54.76+\sqrt{\frac{10\pi}{4.19}}}{54.76}\right)^{0.9148}$$

$$A = 50 + 56(1-B) \Rightarrow 50+56(1-0.9148) = 54.76$$

$$B = 0.25(12-Q_v)^{2/3} \therefore B = 0.25(12-5)^{2/3} = 0.9148$$

$Q_v = (14-21) - 14-9$ assume $Q_v = 5$

$V_{t\,max} = (A+(Q_v-3))^2$



$K_s = 1$ assumption (laziness) $\rightarrow 1$

$K_m = C_{mf} = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma} \cdot C_e)$

C_{mc} =1 assumed uncrowned

$C_{pf} \rightarrow \frac{F}{10dP}$ $\frac{0.75}{10(16)} - 0.025 = -0.0203$

$$C_{ma} = A + BF + CF^2$$

$$\cancel{= 54.70 + 0.9148(0.75) + C \quad (0.75)^2}$$

$$0.247 + 0.0167(0.75) + -0.1765(10^{-4})(0.75)^2 = 0.259$$

$$K_m = C_{mf} = 1 + 1(-0.0203(1) + 0.259(1)) = 1.239$$

C_e (1) assumption

$$I = \frac{\cos\phi_t \sin\phi_t}{2m_N} \cdot \frac{m_G}{m_G + 1}$$

$m_n = 1$ for Spur

m_G = Gear ratio assuming 1

assuming $\phi_t = \phi = 20^\circ$

$$I \quad \frac{\cos 20 \quad \sin 20}{2 \times 1} \cdot \frac{1}{1+1} = \boxed{.08034}$$

$C_f = 1$ (given)

$$\sigma_c = C_p\left(W^t K_o K_v K_s \frac{K_m}{d_pF} \cdot \frac{C_f}{I}\right)^{1/2}$$

$$2284.67\left(\frac{32.5}{420.17} \cdot 1.5\left(\frac{\cancel{34.024}}{1.09}\right) 1 \left(\frac{1.239}{1 \cdot 0.75}\right)\left(\frac{1}{0.08034}\right)\right) = \boxed{122171.2}$$

$$S_H = \frac{S_c Z_N(C_H / K_T \cdot K_R)}{\sigma_c} = \frac{150{,}000(0.974)(1/1 \cdot 1)}{221712} = 0.13$$

$C_p =$

$E = 30\cdot10^6$ psi

$V = 0{,}292$

$$\checkmark\ C_p = \left(\frac{1}{\pi\left(\frac{1-0{,}292^2}{30\cdot10^6}+\frac{1-0{,}292^2}{30\cdot10^6}\right)}\right)^{1/2} = 2284{,}67$$

$3{,}049e^{-8}$

C – MATLAB CODE

```
%Weight vs. Power Theoretical Graph
clear all;
clc;

%%Varying Weight Solving for Power
W = 500:0.1:3000;
uk = .1;
v = 7.6*5280*12*(1/3600);
F1 = W*sind(20) + uk*W*cosd(20);
P1 = F1*v*(1/12)*(1/550);
F2 = W*sind(20) + uk*W*cosd(20) + (W/386.4)*(v/2);
P2 = F2*v*(1/12)*(1/550);
F3 = W*sind(20) + uk*W*cosd(20) + (W/386.4)*(v/2.5);
P3 = F3*v*(1/12)*(1/550);
F4 = W*sind(20) + uk*W*cosd(20) + (W/386.4)*(v/3);
P4 = F4*v*(1/12)*(1/550);
F5 = W*sind(20) + uk*W*cosd(20) + (W/386.4)*(v/3.5);
P5 = F5*v*(1/12)*(1/550);
F6 = W*sind(20) + uk*W*cosd(20) + (W/386.4)*(v/4);
P6 = F6*v*(1/12)*(1/550);
plot(W,P1,W,P2,W,P3,W,P4,W,P5,W,P6)
xlabel('\bfWeight (lb)','fontsize',14)
ylabel('\bfPower (HP)','fontsize',14)
title('\bfWeight vs. Power (u = .1)','fontsize',14)
legend('no accel','2s to max','2.5s to max','3s to max','3.5s to max','4s to
max')
```

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http://www.mcmaster.com/#6494k14/=13jjf01
http://www.mcmaster.com/#5972k293/=13jxxme
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