Project Number:
ME-HXA-1001

Heel-Operated Bass Drum Pedal

A Major Qualifying Project Report
submitted to the Faculty of
WORCESTER POLYTECHNIC INSTITUTE
in partial fulfillment of the requirements for the
Degree of Bachelor of Science
by

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Date: April 28, 2011

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Abstract

Standard, toe-operated, bass drum pedals require the user to have both leg coordination and strength, both of which can be troublesome attributes for the average drummer. Ergonomic studies, however, revealed that the toe is the most natural balance point and the heel the most natural striker for foot-actuated devices. To improve the ergonomics of the bass drum pedal, with the focus specifically on reducing repetitive muscle fatigue and improving the consistency of the drummer's tempo, the goal of this project was to develop a heel-operated pedal design. Design specifications for this pedal were developed based on benchmarking of existing toe-operated pedals. The final heel-operated design utilizes a four-bar linkage as the working mechanism, and has an input-output angular velocity ratio comparable to that of a standard toe-operated bass drum pedal. With the use of SolidWorks, Esprit, and Fourbar software packages, the team successfully designed, manufactured, and tested a prototype of the envisioned pedal.

Executive Summary

Standard, toe-operated, bass drum pedals require the user to have both leg coordination and strength, both of which can be troublesome for the average drummer. Through ergonomic studies it was proposed that a heel-operated bass pedal could alleviate these issues. These studies revealed that the toe is the most natural balance point and the heel the most natural striker for foot actuated devices. A heel-operated pedal is configured so that it uses the toe for balance and the heel for striking, reducing leg coordination issues.

Ergonomic research also focused on the muscle groups used while playing either a toe-operated or heel-operated pedal. This information is significant because when playing with a toe-operated pedal it is not uncommon for a drummer to experience shin splints or other leg injuries. These injuries occur because of the relatively weak muscle groups that are used to activate the toe-operated pedal. Alternatively, the muscles groups used to operate the heel-operated pedal are naturally much stronger and will be able to withstand the forces put on them while playing.

In addition to ergonomic research a kinematic and dynamic analysis were done on a standard toe-operated pedal in order to create performance specifications for the heel-operated pedal. An analysis of the toe-operated pedal revealed that an input-to-output of 1:3 was desirable along with a maximum playable rate of 330 beats per minute. A final performance specification was developed by a pressure indicating film test in which the pressure of a beater hitting a drumhead with a single stroke was measured and converted to force. The force for the toe-operated pedal was converted to be 8.16 lbf.

Design specifications were also created based upon the conducted background research. Using these design specifications seven initial concepts were developed and each was analyzed for advantages and disadvantages. A final design, a 4-bar linkage with a heel plate instead of a footboard, was then chosen to analyze and explore further.

Ergonomic research first helped to determine the starting angle of the foot, which is 10º from horizontal. Then, using this angle in Program Fourbar, the link lengths were optimized to give the best input-to-output ratio and desirable transmission angles. The input-to-output ratio was optimized to be 1:5.9 while the transmission angles ranged from an initial angle of 62° to a maximum angle of 90°. Using these link lengths a SolidWorks model of the design was created

and the forces acting on the pedal were analyzed to ensure that it would not fail while being played.

The model was modified until it passed the stress analysis with a minimum safety factor of 2.9 and was then made into a prototype. Several parts were able to be salvaged from an older toe-operated pedal and only a few parts needed to be machined. These parts were machined in Worcester Polytechnic Institute's Washburn Machine Shops and then assembled into a working prototype. This prototype includes two special features: an adjustable foot rest to make the pedal more universal, and a self-adjusting heel plate to adjust to the angle of the player's foot throughout the stroke.

Testing of the heel-operated prototype consisted of three elements. The first two, a pressure indicating film test and a tempo test, were repeats of those done on the toe-operated pedal. These tests revealed that the beater was capable of hitting the drum head with 7.25 lbf and the maximum playable rate of the pedal was 270 beats per minute. The third test was a feedback test from drummers who have tried the heel-operated prototype. All of the drummers were impressed with how comfortable the pedal was to play, but also offered recommendations of how it could be improved. Taking these recommendations into consideration it was decided that future designs will have a longer and wider area for the foot, a spring underneath the heel plate to it return to its original position quicker, and pins and bushings instead of shoulder bolts for a quicker, smoother stroke.

Acknowledgements

We would like to thank Professor Holly Ault for all of her support and guidance throughout this past year. Without her we would have not learned as much as we did about all of the different considerations that go into designing and building a product from scratch. We would also like to thank Benjamin Hawkins for all of his help with the Esprit files and machining our parts.

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Introduction

When it comes to playing drums, the bass drum plays a significant role in creating a rhythm and sound that both keeps time and is appealing to the audience. The bass drum is played with one's feet via stepping on a specially designed pedal. Throughout its history, the standard bass pedal has always been toe-operated, meaning that the pedal is struck with the distal portion of the foot and the heel is used for balance. This method, however, has been known to not always be effective as many people are unable to play the bass drum due to foot coordination problems. Also, this method of playing tends to put a lot of strain on the ankle and shin and can sometimes cause injuries to these relatively weaker parts of the leg. (Workman, 2006) For these two reasons it has been decided that an alternate approach to the standard bass pedal should be constructed in order to attempt to alleviate these issues.

Relative to the foot the heel is the most natural striker and the toe is the most natural for balance. Therefore, it is proposed that a heel-operated bass pedal, one in which the toe is down and the heel strikes, will help alleviate the coordination problems that people may have and make playing the drums more accessible to everyone.

Furthermore, ergonomic studies suggest that a heel-operated pedal will greatly reduce muscle fatigue due to the fact that the drummer will be using different muscle groups to operate the pedal. With the standard pedal, as previously mentioned, the majority of the strain is placed on the muscles in the ankle and shin. In comparison, with the heel-operated pedal the majority of the strain will be placed on the calf, quadriceps, and hamstring, which are naturally stronger muscles than those in the ankle and shin. (Martini, 2000) By transferring the strain of playing to these stronger muscles the team hopes to reduce leg related injuries and increase the endurance of the drummer.

Background

Before delving into the design of a heel-operated bass drum pedal some background research was done in order to better understand the ergonomics, kinematics, and dynamics of toe-operated and heel-operated pedals. This chapter first explores different types of pedals and playing techniques, along with the ergonomics of those playing techniques. Subsequently a kinematic and dynamic analysis of a standard toe-operated pedal is discussed, along with an in-depth examination of a patent for an existing heel-operated pedal.

Toe-Operated Pedals

How They Work

In order to better understand how the heel-operated pedal should work, a foundation of knowledge must first be built by examining the operation of a standard toe-operated pedal. The standard pedal that was chosen for analysis was the Pearl Eliminator 2002B. A diagram of a similar pedal, the 2000C, and its components can be seen below in Figure 1. The 2000C is chain driven, whereas the 2002B is belt driven; otherwise these models are identical.

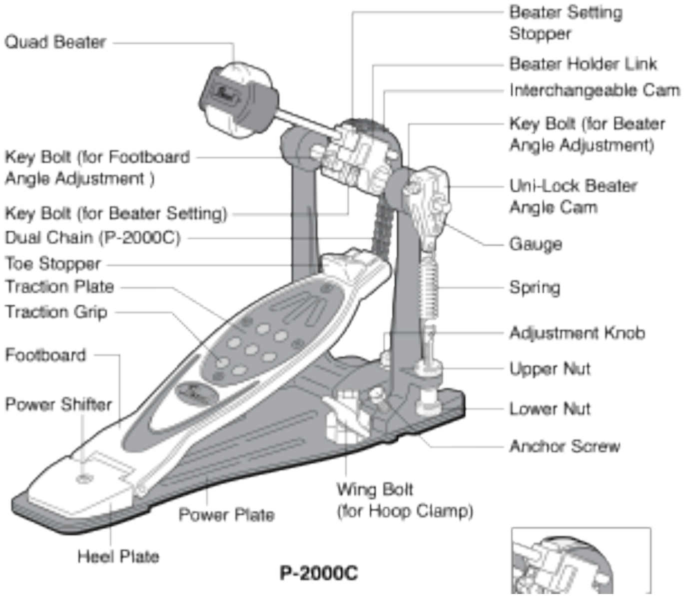


Figure 1: Pearl Toe-Operated Bass Drum Pedal (Pearl Drum Pedal P-2000C/P-2000B Instruction Manual)

Simple analysis from visually observing the pedal while someone is playing it tells one that the pedal operates by pushing the footboard down, which simultaneously rotates the beater shaft, and causes the beater to strike the drum. Then, when the pressure from one's foot is taken off of the footboard, a spring is used to return the beater to its original position.

Direct Drive vs. Chain or Belt Drive

The main difference between direct drives and chain or belt drives is the fact that direct drives use a rigid link, whereas chain and belt drives use a non-rigid link to connect the footboard to the rotation shaft. All three of these can be seen in Figure 2.

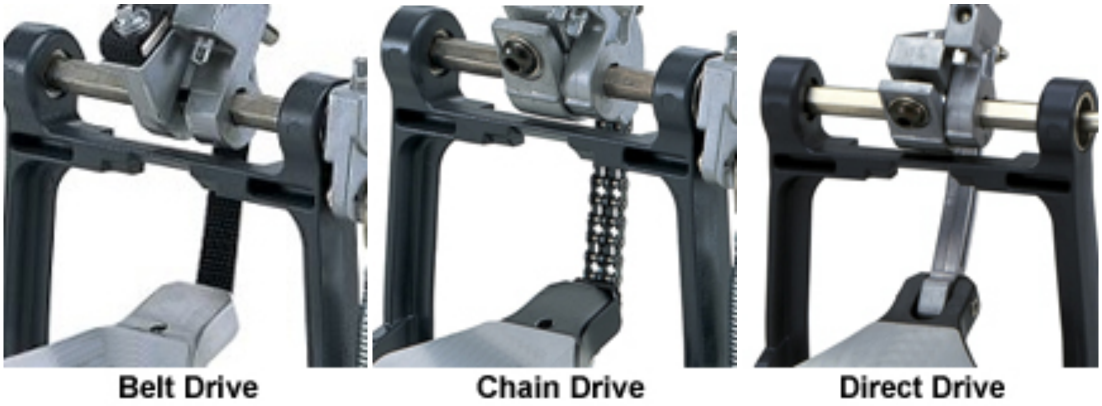


Figure 2: Belt Drive, Chain Drive, and Direct Drive Pedals (Double Bass Pedal Buyers Guide, 2010)

Both of these options have advantages as well as disadvantages. A non-rigid link allows for a lighter feel on the foot, is more adjustable, and also tends to last longer than the rigid link system. (Double Bass Pedal Buyers Guide, 2007) The chain drive is a little more durable than the belt drive, and has very little lag, but also needs to be cleaned regularly and makes unwanted noise. The belt drive, on the other hand, is faster, lighter, and doesn't require much maintenance, but is less durable and sometimes has unwanted lag between when one's pushes down the footboard and when the beater begins to move.

The rigid link direct drive system, however, allows for more speed and control of the pedal. Having no chain or belt in the system eliminates any lag that can be caused by a loose link and thus allows the pedal to be extremely responsive. The ability to have a rigid line of force transmission directly from the pedal to the beater makes these systems capable of more easily producing precise and intricate rhythms. The rigid link systems are not quite as durable as their non-rigid counterparts but they do require little to no maintenance. The disadvantages of the rigid link pedals are that they have little adjustability and can feel more mechanical and less smooth to the player. Both of these methods, rigid and non-rigid, will be explored when designing the heel-operated pedal. (Double Bass Pedal Buyers Guide, 2010)

Playing Styles: Heel Up vs. Heel Down

There is no right or wrong style to play the bass drum. Each method, heel up or heel down, see Figure 3, just provides a different aspect to drumming.

There are however advantages and disadvantages of each style. When playing heel up, the drummer is able to generate more power and speed while exerting less stress and strain on the body. When playing heel down, most of the relatively weaker muscles in the leg are being used to drive the beater head, and for beginning drummers this is the main problem. Until a drummer's leg is strengthened through time and experience, most drummers tend to play heel down. When playing heel down, the drummer is primarily using only the muscles in the shin and ankle. The problem with using those muscles is that in order for this style to work effectively one must have strong shins. Playing heel down allows the drummer to make controlled soft beats against the drumhead. When playing heel down, most drummers position themselves farther away from the drum, because it creates a more natural position of one's foot to strike the foot pedal. When playing heel up most drummers position themselves so that their knee is positioned directly over the foot pedal which also makes it easier to play the drum set because one is positioned closer. Heel up is the more commonly used style for experienced drummers.

Ergonomics

During our ergonomic research we learned that a seated operator can operate foot controls more easily than a standing operator (Kroemer, 2001). This is because the operator's seat largely supports his/her body, in relation to a standing operator, who has to shift his/her weight over to one leg, causing the operator more fatigue. Thus the reduced weight on the operator's feet allows him/her to move more freely and, given suitable conditions, allows the operator to exert larger forces with less stress on the rest of the body. From this inquiry we can make the assumption that designing foot controls for a seated operator is ideal. Figure 4 shows a seated operator's proper knee angle range. This angle range will produce maximum performance and is considered the best ergonomically.

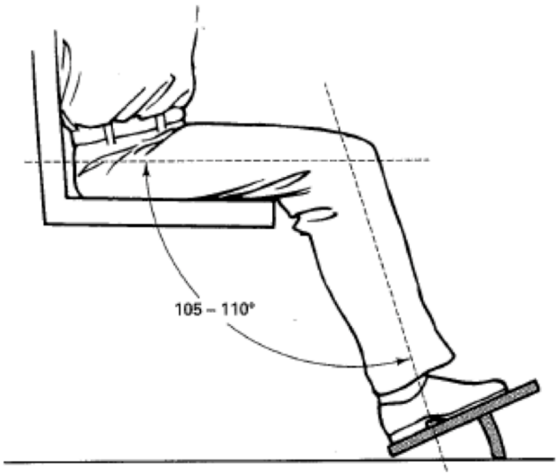


Figure 4: Ideal Knee Angles For Operation (Kroemer, 2001)

The largest forces can be generated with extended or nearly extended legs in the downward direction, limited by body inertia, and in the forward direction, limited by both inertia and the back support. (Kroemer, 2001)

Another factor that decides how much force can be generated by the foot is the pedal (ankle) angle. See Figure 5. In this figure, D represents the horizontal distance to the foot pedal from the front of the seat, and H represents the vertical distance to the pedal below the seat. We must also consider in our design that the mechanism must fit within the preferred and regular spaces for the feet, assuming a seated operator, see Figure 6.

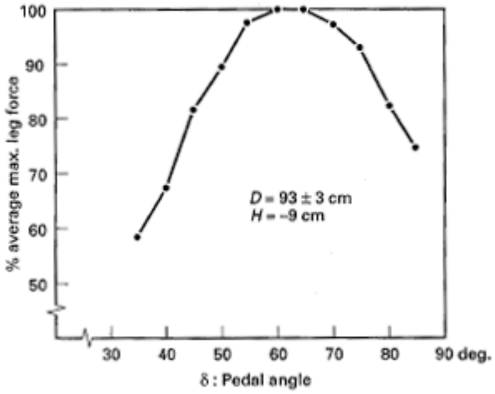


Figure 5: Force Output Relative to Pedal Angle (Kroemer, 2001)

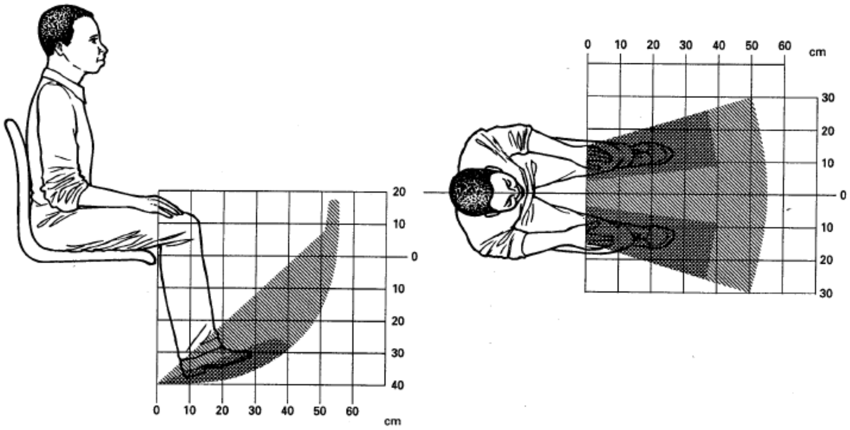


Figure 6: Ideal Foot Workspace (Kroemer, 2001)

Good design practices recommend that motions be limited to fewer than 400 repetitions. When repetitive motions are unavoidable, the design should minimize the loads on the user and position the joints in the most natural or neutral position where the muscles have maximum strength. Repetitive motion injury is what can occur when a body part is repeatedly overused. For our design repetitive motions are unavoidable since the use of a bass pedal requires repetition to keep a rhythm. A key reason for redesigning for heel-operation is to avoid shin splints, caused by tightening muscles and increased friction, which causes a pull on the tendons over time(Workman. 2006). The tight muscles then lose their elasticity and when under activity they do not allow enough stretch. This pulls the tendon from the bone it attaches to causing the pain felt on the shins.

The injury that is most common at the ankle is tenosynovitis, an inflammation of tendons and tendon sheaths in particular where these tendons cross tight ligaments. To avoid these it is best to create a design in which the operator is using a more natural range of angles, in particular for the ankle between 15 to -35 degrees on a horizontal axis. See Figure 7.

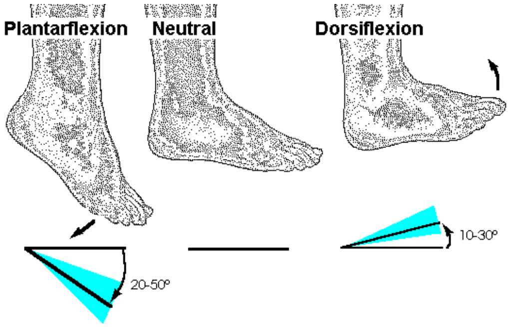


Figure 7: Natural Plantarflexion and Dorsiflexion Angle of Ankle Ranges (gla.ac.uk)

Muscles That Move the Foot

The extricate muscles that move the foot are shown in Figure 8 and are listed in Table 1, found in Appendix A. The muscles that move the foot at the ankle can be broken into two groups: the plantar flexors and the dorsiflexors. Plantar flexion is the movement that increases the approximate 90-degree angle between the front part of the foot and the shin, as when depressing an automobile pedal. It occurs at the ankle, and the range of motion for plantar flexion is usually indicated to be 30° to 40°, but can sometimes be up to 50°. Dorsiflexion is the movement that decreases the angle between the dorsum (superior surface) of the foot and the leg, so that the toes are brought closer to the shin. Put more simply, it applies to the upward movement of the foot at the ankle joint. The range of motion for dorsiflexion is indicated to be a maximum of 15 degrees in the majority of subjects tested.

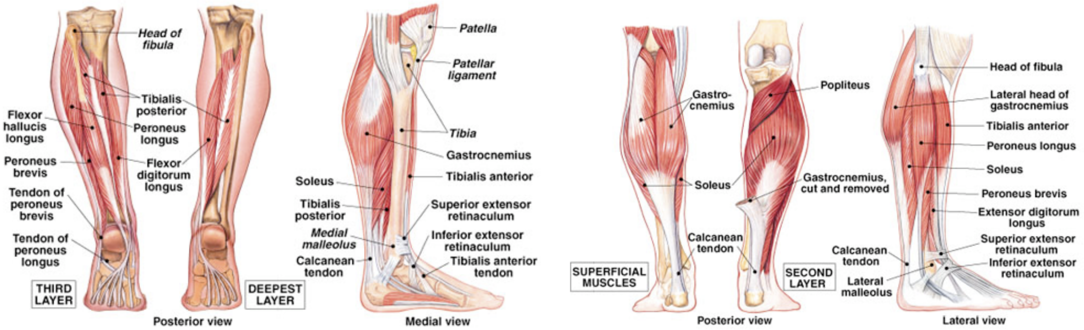


Figure 8: Leg Muscles 2nd and 3rd Layers (Martini, 2000)

Most of the muscles that move the ankle produce the plantar flexion involved with walking and running movements, and, for the purpose of this project, are in part responsible for the heel movement necessary for a heel-operated bass pedal. The gastrocnemius muscle of the calf is an important plantar flexor but the muscle fibers of the underlying soleus muscle are more powerful[4], due to the soleus' being in constant tension. This fact is what prevents the body from falling forward. These muscles are best seen in the posterior and lateral views shown in Figure 8. This makes the heel strike a powerful and precise strike. The gastrocnemius and soleus muscles share a common tendon, the calcanean tendon, commonly known as the Achilles tendon. Of these muscles the soleus was associated in a study with the flexor digitorum longus and deep crural fascia as being one of the most likely causes of shin splints.[6]

Toe-Operated/Heel Up

For the operator that has a toe-operated bass drum pedal, and plays with his heel up, the muscles that he uses are all the same as those of the heel-operated pedal. However, this is accounting for the fact that all of the flexor muscles previously mentioned are in constant flexion and are therefore more prone to fatigue. As for thigh muscles, the only two that are being used are the iliopsoas muscles, in the same fashion as for the heel-operated bass pedal, to lift the leg.

Toe-Operated/Heel Down

For this operator there is no need of the thigh muscles to be used and so the operator solely relies on a combination of dorsiflexion and plantar flexion foot muscles. Despite the previously mentioned plantar flexion muscles that were used in the heel-operated bass pedal this type of operator mostly uses his dorsiflexion muscle. The only muscle the operator uses for this motion is the tibialis anterior, which is the same muscle that gives the operator the sensation of shin splints.

Heel-Operated Pedal

The muscles that the drummer uses are all the same as those of the toe-operated/heel up style of play, without the flexor muscles being in constant flexion.This allows for the operator to balance his foot with his toes using a minimal amount of strength to do so.

Muscles That Move the Thighs

When looking at the movement for the heel-operated bass pedal the muscles of the thigh must be considered because this is where much of the lift force of the lower leg comes from, which is considered to be a flexion. Table 2 in Appendix A lists the muscles that move the thigh. From this list the muscles that produce the flexion action are the tensor faciate latae, adductor brevis, adductor longus, pectineus, gracilius, iliacus, and the psoas major. Yet the only muscles that produce the flexion that is desired for lifting the operator's leg are the iliopsoas group muscles of the pelvis, as seen in Figure 9. This is due to the iliopsoas group being the only muscles that cause rotational lift at the joint where the pelvis meets the femur. When striking with the heel the operator is using their extension muscles, in particular the gluteus maximus. The gluteus maximus for this particular case is able to use gravity to its advantage, requiring the muscles to exert less downward force. The iliopsoas muscles are a pair of powerful hip flexor muscles that dominate the medial surface of the pelvis. The large psoas major muscle originates alongside the inferior thoracic and lumbar vertebrae, and its insertion lies on the lesser trochanter of the femur. Before reaching this insertion, its tendon merges with that of the iliacus muscle, which nestles within the iliac fossa.

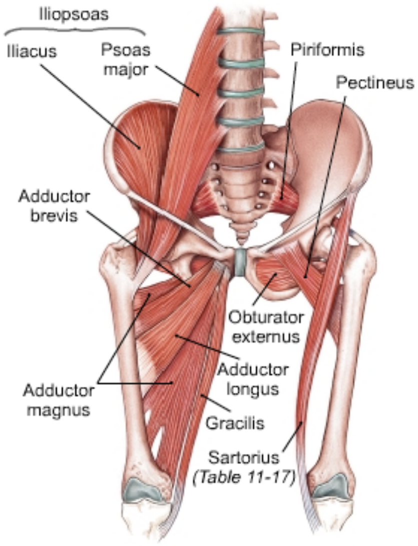


Figure 9: Iliopsoas Group Muscles of the Pelvis (Martini, 2000)

From this research we can assert that the heel-operated bass drum pedal requires far less effort to accomplish the same task and provides a more natural strike than the toe-operated pedals. This reduces tenosynovitis and shin splints which we associate with the more conventional toe-operated bass pedals.

Repetitive Motion Disorder

Repetitive motion disorder is a result of repeated motions that are performed in normal work or daily activity. These are uninterrupted repetitions of a movement performed in an unnatural or awkward way. These motions include the twisting of joints such as the ankle and shoulder, but can also occur in the legs, feet, knees, hips, back, and neck. The symptoms associated with repetitive motion disorder are sharp pain, tingling, numbness, visible swelling or redness of the affected area, and short periods of loss of flexibility and strength. Over time this can cause temporary, or even permanent, damage to the body's soft tissue; which includes the nerves, ligaments, muscles and tendons in addition to compression of the nerves or tissue. Individuals with this disorder usually perform repetitive tasks such as playing musical instruments, or doing computer and assembly line work, among other things. Individuals with a stint of RMD usually recover completely and can avoid re-injury by simply changing the way in which they perform the particular motion that causes RMD. This is yet another reason for designing a heel-operated bass pedal.

Toe-Operated Pedal Analysis

An analysis of a standard toe-operated pedal, the Pearl Eliminator 2000B, was completed in order to better understand how a bass drum pedal works, and to develop performance specifications for the heel-operated design.

Vector Loop Kinematic Analysis

A vector loop kinematic analysis was done to determine the input-output ratio of the standard toe-operated bass drum pedal. The first step in completing this analysis was to draw the vector loop. A sketch of this loop can be seen in Figure 10.

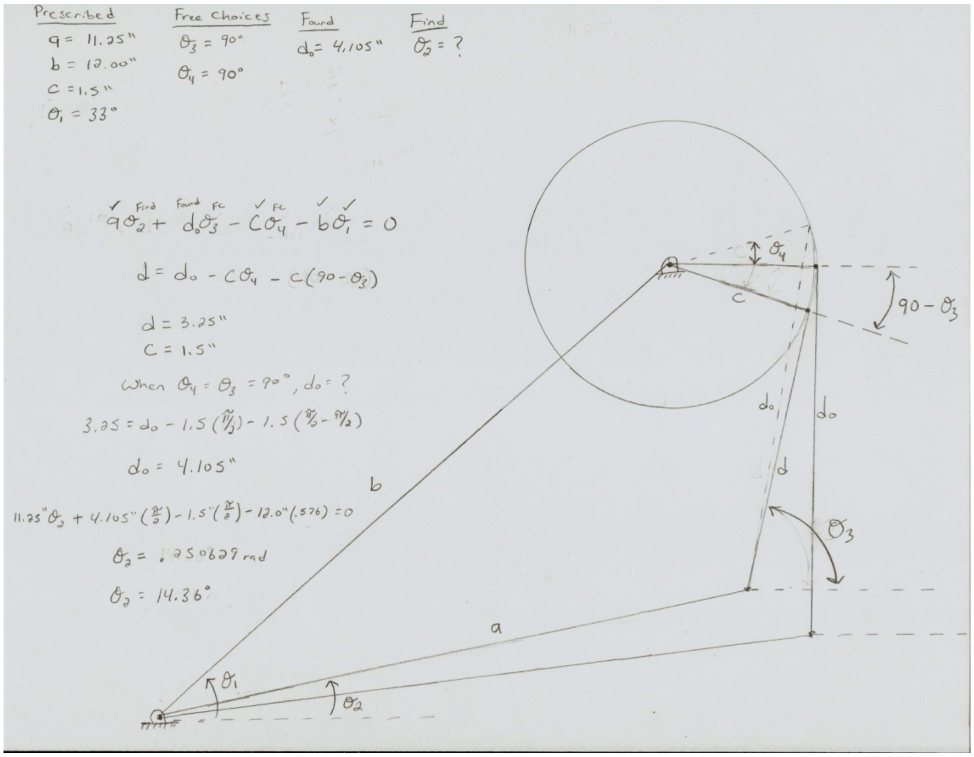


Figure 10: Vector Loop Analysis Sketch

The vector loop sketch has been labeled with letters and variables to make the equations easier to write. The known variables are:

- “a,” - Length of the footboard
- “b,” - Distance from the fixed pivot at the heel end of the footboard to the center of the cam;
- “c,” - Radius of the cam;
- “d_0,” - Length of the strap when the strap is in a vertical position
- “θ_1,” - Angle between the horizontal and link “b.”

The variable “θ_2,” will be used as the input variable, and the variables “d,” “θ_3,” and “θ_4” are unknown. From the sketch, the first set of equations for the position analysis was created.

Position Analysis

Using the **Design of Machinery** textbook as a reference (Norton, 2010), a vector loop equation for the position analysis of the standard pedal linkage was created. Euler’s identity was then substituted into the equation to give it real and imaginary components. The vector equation was then separated into two equations representing X (real components), and Y (imaginary components). These two equations along with a third equation, which represented the variable d, or length of the strap, were then used in MathCad to solve for variables θ_3, θ_4, and d in terms of the known variables a, b, c, θ_1, and θ_2. The derivation of the three equations can be seen in Appendix C.

With the equations for position now known, they were put into MatLab and the three unknown variables were solved for. The results can be seen in Table 1. From these results the team was able to plot the graph of θ_2 vs. θ_4, shown in Figure 11, to show the relationship between the input angle of the footboard and the output angle of the beater. The slope of the linear fit line, shown on the graph as $y = 3.0427x – 41.473$, is equal to the input-output ratio. The input-output ratio for the standard toe-operated pedal is approximately 1:3.

Table 1:Vector Loop Position Analysis MatLab Results

θ_2 (deg)	θ_3 (deg)	θ_4 (deg)	d (in)
13.0	89.4735	-2.0068	3.8388
13.5	89.1281	-0.4406	3.7887
14.0	88.7784	1.1171	3.7388
14.5	88.4239	2.6664	3.6889
15.0	88.0642	4.2076	3.6392
15.5	87.6989	5.7408	3.5895
16.0	87.3273	7.2660	3.5398
16.5	86.9489	8.7834	3.4902
17.0	86.5631	10.2929	3.4406
17.5	86.1693	11.7945	3.3909
18.0	85.7655	13.2879	3.3413
18.5	85.3542	14.7730	3.2916
19.0	84.9312	16.2494	3.2419

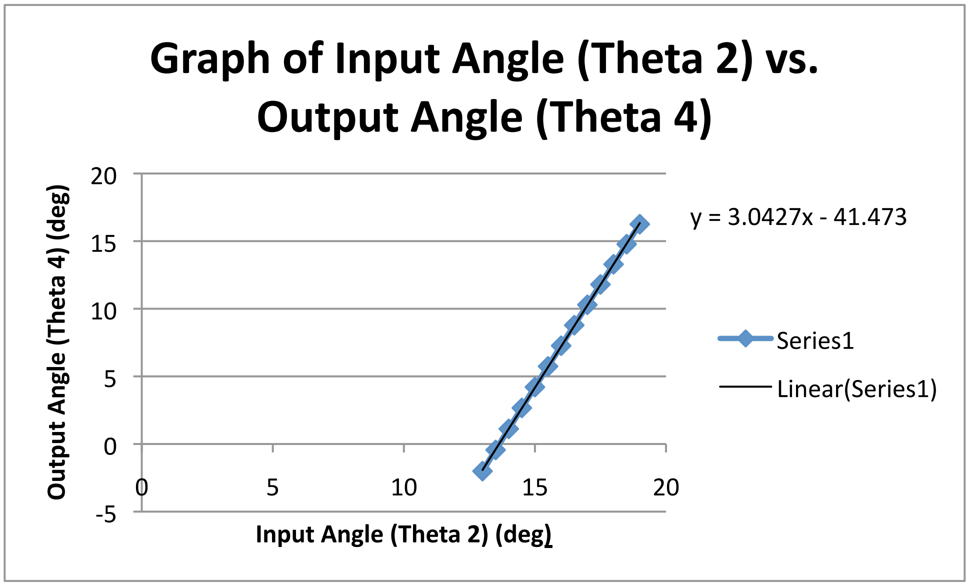


Figure 11: Graph of Input Angle vs. Output Angle

Solid Works Model Dynamic Analysis

In addition to the kinematic analysis, dynamic analysis was completed using a SolidWorks model of the bass pedal and the mechanism motion simulation feature shown in Figure 12,. The model is a representation of the Pearl P-2000B bass drum pedal. The reason for this simulation was to better understand how the more traditional type of bass pedal worked before the team delved into developing any heel-operated bass pedals. The model was assembled into four separate sub-assemblies: one for the base of the pedal in the color green, the beater/shaft sub-assembly colored red, the spring assembly in orange and pink, and the foot rest in blue. Note that the model does not include the strap which connects the footboard to the cam on the beater assembly. The purpose of the sub-assemblies was to ensure that the only components that moved were the beater, the footrest and the spring assembly. The software is not capable of modeling the flexible strap between the footrest and the cam so the footrest has no effect on the beater. This was remedied by adding an equation for the relationship between the rotation of the footboard and the rotation of the beater shaft, as derived in the vector loop analysis above. The simulation of the model allowed retrieving data on the angular velocity, angular acceleration, and displacement that occurs in the model.

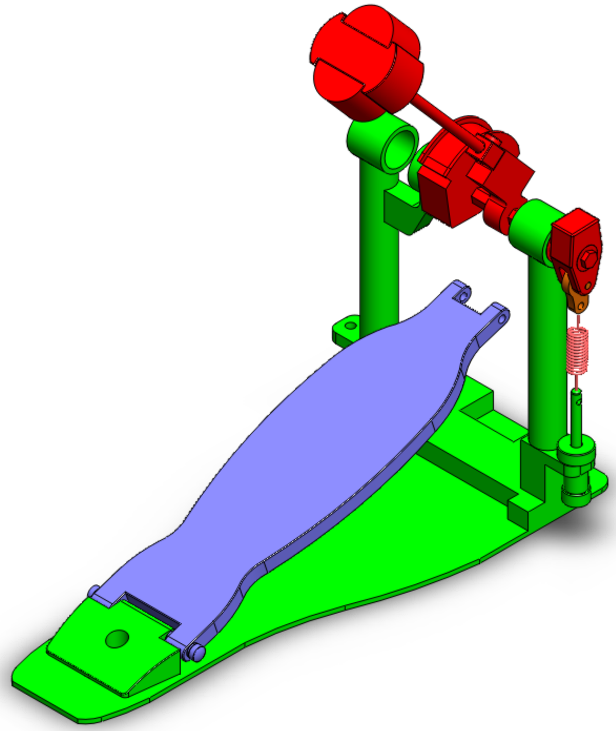


Figure 12: SolidWorks Model of Pearl P-2000C

For most of our model analysis the same pin joint is used. This concentric is shown in Figure 15 and is highlighted in orange. The remainder of the analysis on the model is given in Appendix B and covers velocity and displacement of the beater. Note that in both velocity and acceleration only the x-component was taken because there is no movement about the y and z-axes.

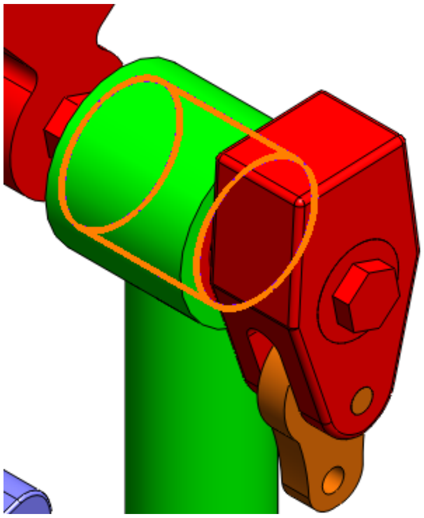


Figure 13: Concentric 2 Used for Analysis

To obtain data for the angular displacement, velocity, and acceleration of the beater, an oscillating motor was attached to the axis of the beater shaft at concentric 2. The settings used for the motor were a frequency of 1 Hz and an oscillation of 45 degrees (π/4 rads) with the beater head initially set at an angle of 135 degrees (3π/4 rads) from the horizontal. With these settings the motor made a full cycle every second.

To properly model the spring in the simulation, the team needed to obtain the spring constant of the physical spring on the existing pedal. The spring constant used in the simulation was 3.31 N/mm and has a free length of 51.2 mm, which was obtained by measuring the force required to stretch the spring to several length values within its operating range. The length and force values are plotted in Figure 13. A trend line was drawn through the points and the slope of that line was converted from lbs/in to N/m to give the spring constant of 3.31 N/m

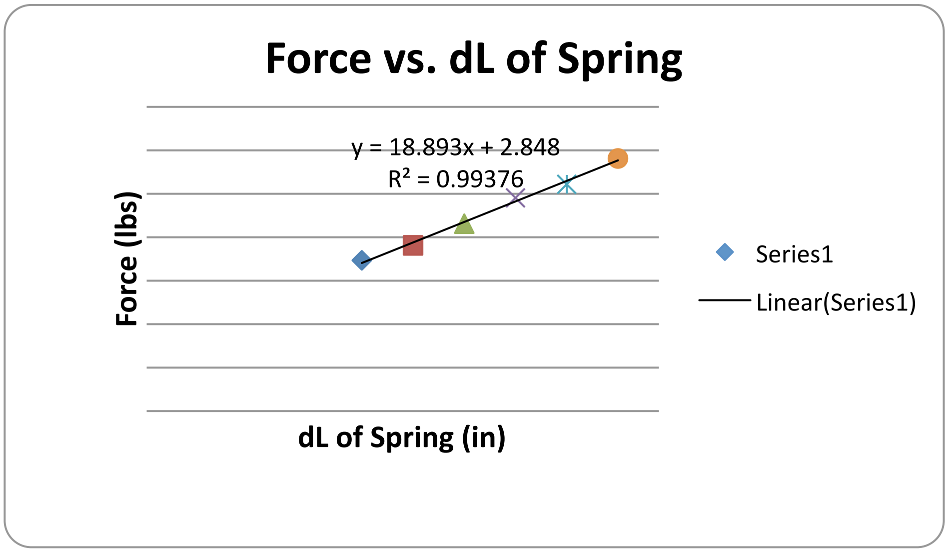


Figure 14: Spring Constant Analysis Graph

The materials used for the simulation were the same as those used to fabricate the Pearl P-2000B bass drum pedal: plastic, aluminum, steel, and felt. Figure 14 shows the angular displacement of the beater and was taken from the pin joint where the shaft/beater sub-assembly is linked to the base sub-assembly, which is our ground. Note that since it is an angular displacement only the magnitude can be shown.

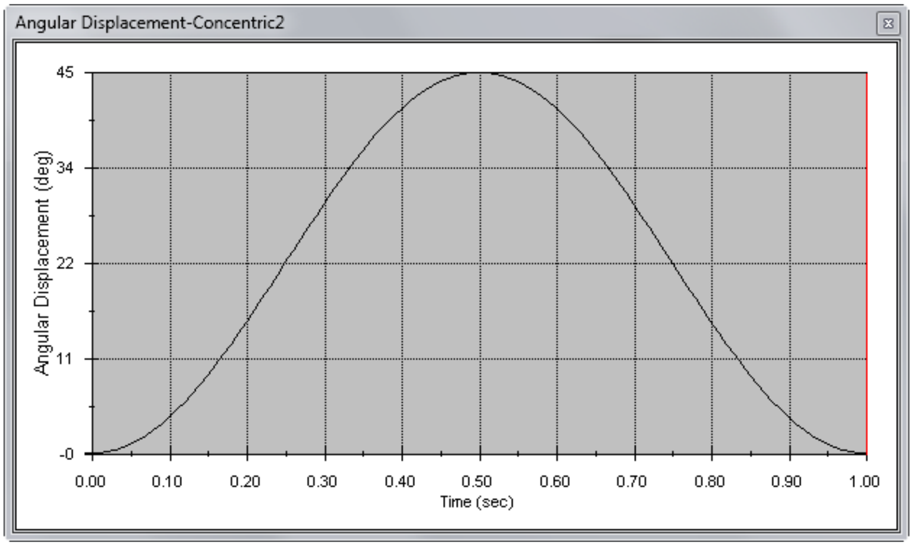


Figure 15: Graph of the Angular Displacement of the Beater (Magnitude)

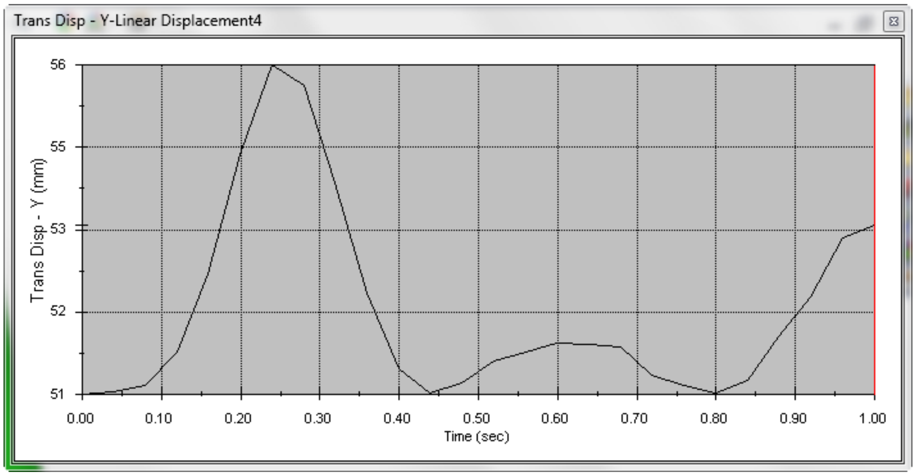


Figure 16: Spring in Y-Axis When a Force is Applied

Figure 16 above shows the displacement of the spring in the y-axis when a torque force of 105N*mm is applied to the beater shaft for an instant and then released. This was achieved using a linear interpolated function in the force function feature of the Force/Torque application found in Motion Study tab. Note that it takes less than 0.5 seconds for the spring to return to its free length and then oscillate in accordance with the force applied.

Tempo Testing

The maximum playable rate of the standard pedal was determined by counting how many times a drummer could make the beater strike the drumhead over a ten second period. Then, by multiplying that number by six, the rate was converted into beats per minutes. The outcome of this test revealed that the maximum playable rate of the standard pedal was 330 beats per minute.

Slow Motion Video Analysis

Using a combination of recorded videos taken at 30 fps and Adobe Premier, Adobe Photoshop and a grid placed in the background of the videos as a reference scale the team was able to take measurements of displacement of the beater head and toes in a real time environment. In order to accomplish this export frames were set up of the different extremities of each point during operation that I sought to measure into Photoshop. In Photoshop a parameter was then changed for the measure tool in measurement scales to transform pixels into inches 30.5 pixels to 2 inches. This was done by using the reference grid that was placed in the background. After this the measure tool was used to determine the displacement by determining the position and angle of ten points in the trajectory of every moving component that needed to be measure. Once done with these changes in displacement, the time lapse in the video is referenced with the displacement and derive the velocity. The data used for the calculations is in Table 2 below. The time interval for one full completion of stroke of the beater was ~ 0.0833 seconds.

The angular velocity was derived from the equation $\omega = d\theta/dt$.

Figure 17: Beater Slow Motion Image

Table 2: Beater Head Displacement Over Time

Beater Head					
Position	X	Y	R≈	ϴ	Time
1	4.88	3.5	6	35.6	0.00833
2	4.77	3.63	6	37.3	0.01666
3	4.57	3.89	6	40.4	0.02499
4	4.22	4.26	6	45.3	0.03332
5	3.76	4.68	6	51.2	0.04165
6	3	5.27	6	60.4	0.04998
7	1.97	5.66	6	70.8	0.05831
8	0.68	5.95	6	83.5	0.06664
9	0.88	5.93	6	98.4	0.07497
10	1.51	5.8	6	104.6	0.0833

Figure 18: Foot Plate Slow Motion Image

Table 3: Toe Displacement Over Time

Toe Displacement					
Position	X	Y	R≈	ϴ	Time
1	8.71	3.82	9.5	21.7	0.00833
2	8.73	3.74	9.5	21.2	0.01666
3	8.79	3.58	9.5	20	0.02499
4	8.84	3.45	9.5	19.3	0.03332
5	8.89	3.32	9.5	18.5	0.04165
6	8.94	3.17	9.5	17.5	0.04998
7	9.02	3.01	9.5	16.5	0.05831
8	9.05	2.85	9.5	15.5	0.06664
9	9.15	2.59	9.5	13.8	0.07497
10	9.2	2.25	9.5	11.8	0.0833

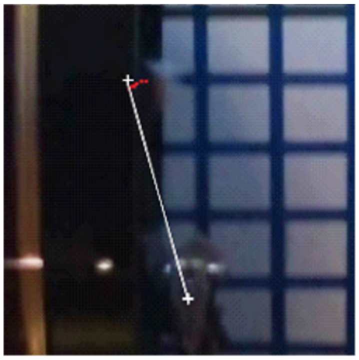


Figure 19: Drum Head Slow Motion Image

Table 4: Drumhead Deflection Over Time

Drumhead Deflection					
Position	X	Y	R≈	Θ	Time
1	1.62	5.75	6	105.8	0.004167
2	1.53	5.78	6	104.8	0.008333
3	1.44	5.79	6	103.9	0.0125
4	1.4	5.83	6	103.5	0.016667
5	1.26	5.85	6	102.1	0.020834
6	1.13	5.86	6	100.9	0.025

Force of Beater on Drumhead

To obtain the force of the beater on the drumhead the team decided to strike pressure sensitive paper with a single stroke of the pedal, and then convert the pressure measurement into a force. Because different pressure sensitive papers measure different pressure ranges, the team had to first estimate the force with which the beater strikes the drumhead. The method used to estimate the pressure involved measuring the deflection of the drumhead. The deflection of the drumhead was measured by placing weights on a small surface, a spool of gimp, so that the force would be acting on a smaller region of the membrane. Deflection measurements were taken with every 2.5 lbs of force on the head over a range of 0-20 lbs. The results can be seen Figure 20.

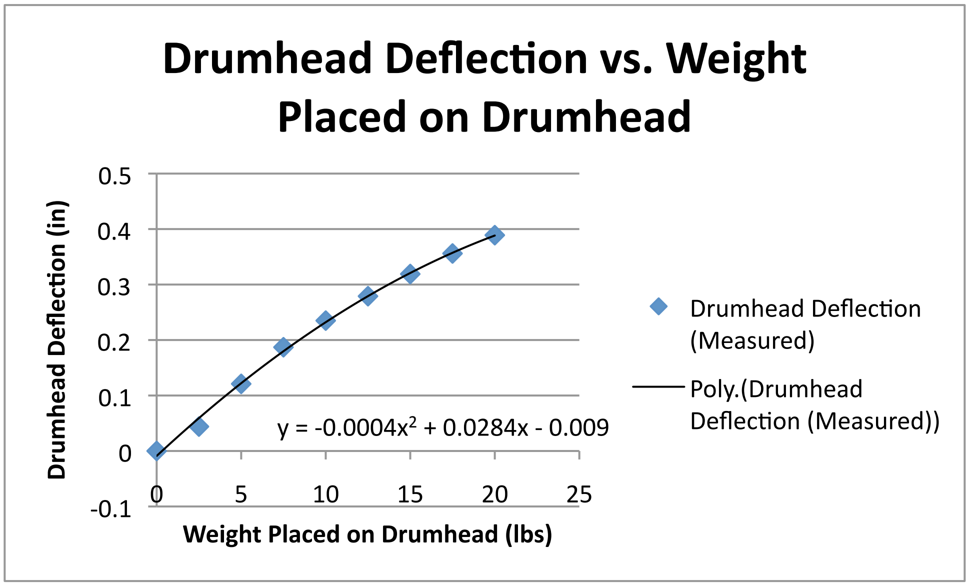


Figure 20: Graph of Drumhead Deflection vs. Weight Placed on Drumhead

As can be seen on the graph the deflection equation found from the results is:

$$y = -0.0004x^2 + 0.0284x - 0.009$$

where "y" is the distance the drumhead deflects and "x" is the amount of force on the head.

From the slow motion video analysis, mentioned earlier, it was determined that the total deflection of the drumhead from a single stroke of the beater was 0.49 inches. If we use that number as the "y" variable in the equation found from the measurements we can calculate the force with which the beater hits the head.

$$0.49 = -0.0004x^2 + 0.0284x - 0.009$$

so

$$x = 31.9 \text{ lbs}$$

Now that the amount of force the beater hits the head with is known we can calculate the pressure on the drum membrane by dividing the force by the approximate contact surface area of the felt.

$$\text{Pressure} = \text{Force/Area}$$

$$\text{Pressure} = 31.9 \text{ lbs}/0.72 \text{ in}^2$$

$$\text{Pressure} = 44.3 \text{ psi}$$

Knowing the pressure put on the membrane by the beater allowed us to purchase the proper pressure sensitive paper to get a more accurate measurement, and create a performance specification for our design.

Pressure Indicating Film Test

To measure the beater head force applied on the drum membrane, we used ultra low pressure indicating film capable of measuring pressures between .2 and .6 MPa, which falls within our estimated pressure range. When pressure is applied, tiny micro-bubbles burst within the film to show the various levels of pressure, through color density, that corresponds to the pressure and pressure distribution. Higher pressures result in darker red colors on the pressure indicating film. For calculating the pressure applied to the drum membrane, we cut the 8x11 inch paper into 2x2 inch sections, taped the pressure film in the center of the drum head, and struck the drum membrane with the beater head as hard as possible. For accurate results, we carried out the test four times to ensure consistent results and could factor out any potential outliers. We then matched the color density obtained from the tests and compared them to the color density chart to obtain the MPa that corresponds closest to the observed color. From the color density we estimated that the beater head struck the drum membrane at .45MPa (converting to lbf/in^2).

$$0.45\ MPa * \frac{20885\ \frac{lbf}{ft^2}}{1\ MPa} * \frac{1\ ft^2}{144\ in^2} = 65\frac{lbf}{in^2}$$

To obtain the applied force we measured the contact area between the beater head and the drum membrane. We found the surface area of the beater head to be 0.125 in^2. Then to obtain the force we multiplied the calculated pressure value of lbf/in^2 by the surface area in which the beater head makes contact with the drum membrane.

Figure 21: Toe-Operated Pressure Paper Results

$$65\,\frac{lbf}{in^2} * 0.125\ in^2 = 8.16\ lbf$$

Heel-Operated Bass Pedals

The final element of background research to be completed was an analysis of an existing heel-operated bass pedal. For this the team examined a heel-operated patent and broke down the mechanism into several parts, determining how each part contributed to the motion of the pedal.
Heel Driven Actuator for a Percussion Instrument
United States Patent Number: 5,866,830

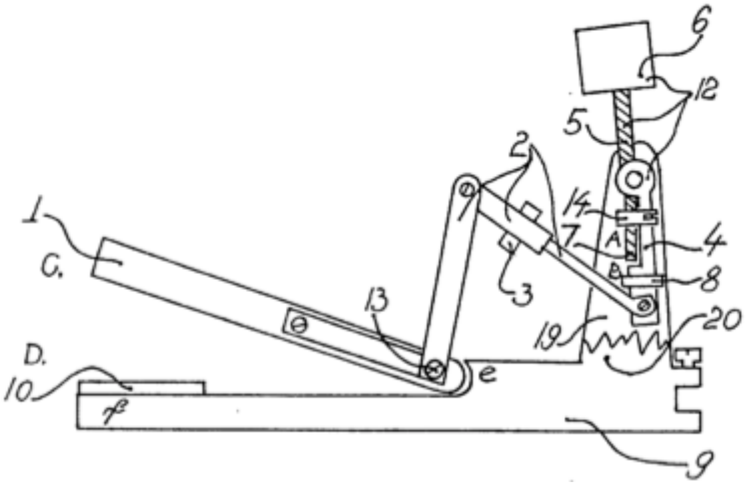


Figure 22: Heel-Operated Pedal Patent 5,866,830 Position A

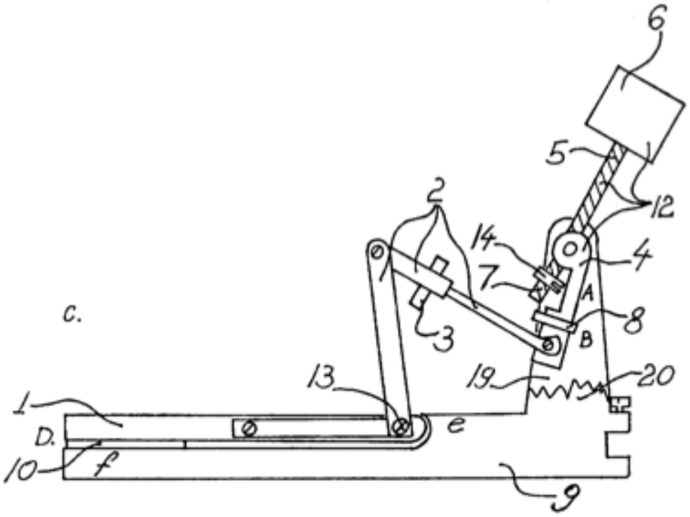


Figure 23: Heel-Operated Pedal Patent 5,866,830 Position B

The heel-operated pedal shown in Figures 21 and 22 is activated by pressing downwards on the footboard (1). This causes the linkage to work as a bell crank and pivot at joint (13). The pivoting motion pulls on the transmitting levers (2), which in turn pivots the pushing arm (4) and the responding arms (5 and 7). As the responding arms (5 and 7) are pivoted the beater (6) moves toward the drumhead.

This design also includes another feature that can alter the performance of the pedal. Fixing nut (8), which is threaded on to the pushing arm (4), can be placed in either A or B position, as show in Figures 7 and 8. When in position A the fixing nut (8) acts to rigidly fix together pushing arm (4) and responding arms (5 and 7). This means that the three arms will pivot in unison about a central axis (not shown). When in position B the fixing nut (8) does not rigidly fix the arms together and thus responding arms (5 and 7) are free to pivot independently of pushing arm (4).

Position A allows the player more control over the system with the rigid link creating a direct relationship between input and output forces. Position B, however, allows the player to use a smaller input stroke because the inertia of the responding arms (5 and 7) will carry the beater towards the drumhead even after the stroke is halted. Depending on the return speed of the beater position B could allow for faster playing speeds, as a result of the shorter strokes. On the other hand it could also create a muffled drum sound because the player will not be putting as much force on the pedal as they would in position A.

Concept Design and Selection

Following the completion of the background research, the team proceeded to develop several concepts for the heel-operated design. Starting with a list of design specifications a total of seven concepts were created and individually analyzed to determine which design would be the best to explore further.

Heel-Operated Bass Drum Pedal Design Specifications

Based on the background research, the team developed the following list of design specifications as a guideline for creating the heel-operated pedal concepts.

1.0 Performance

- 1.1 The pedal should be heel-operated.
- 1.2 Be able to play at a rate of at least 330 beats per minute. (Based on testing done by an amateur player using a Pearl P-2000B standard bass pedal. This was the fastest rhythm that could be played.)
- 1.3 Input to output angle ratio no less than 3:1.
- 1.4 Hit drumhead with at least 7 lbs of force. (Determined from the average force of a Pearl P-2000B bass pedal hitting a drumhead.)
- 1.5 Mechanism should not have any toggle positions.
- 1.6 If using a rigid linkage, transmission angle should be no less than 60º. (90º would be optimal but according to knowledge obtained from ME593K Kinematics, anything of above 60º is acceptable.)
- 1.7 The mechanism should attach to the drum for stability.
- 1.8 The player's foot should not easily slide off of the footboard.
- 1.9 Initial angle of the beater shaft should be between -45º and -25º from vertical. (Angle of the beater shaft for the Pearl P-2000B pedal is adjustable within this range.)

2.0 Environment

- 2.1 The pedal may experience humid conditions.
- 2.2 The pedal may experience freezing temperatures.

3.0 Life in Service

- 3.1 Should withstand an operating period of 1 hour per day for 3 years.

4.0 Target Costs

- 4.1 The product should have an end user cost of $350.

4.2 The cost of manufacturing the prototype should be less than $250.

5.0 Maintenance

5.1 Standard drum key should be the only tool required for maintenance.
5.2 To be maintenance free except for light lubrication, if using a chain, once a month.
5.3 All adjustable parts should be easily accessible without injuring the adjuster.

6.0 Size and Weight

6.1 Weight should not exceed 10 lbs.
6.2 Length should not exceed 15”
6.3 Width should not exceed 8”.
6.4 Height should not exceed 16”

7.0 Aesthetics

7.1 If cost allows, the pedal should look attractive to improve its perception within the market.

8.0 Ergonomics

8.1 Length of the footboard must be at least 11”. (This is the length of the footboard for the Pearl P-2000 pedal.)
8.2 Initial angle of footboard must be less than 18º from horizontal. (Taken from ergonomic research that states this is the plantar flexion angle range of the 5th percentile of humans.)

9.0 Safety

9.1 Pedal should have no exposed sharp parts.
9.2 The player’s foot should not be able to slide into the linkage.
9.3 Mechanism should not fail due to normal playing stresses.

Preliminary Designs

Preliminary designs were based on existing designs as well as ideas of our own creation.

Concept 1

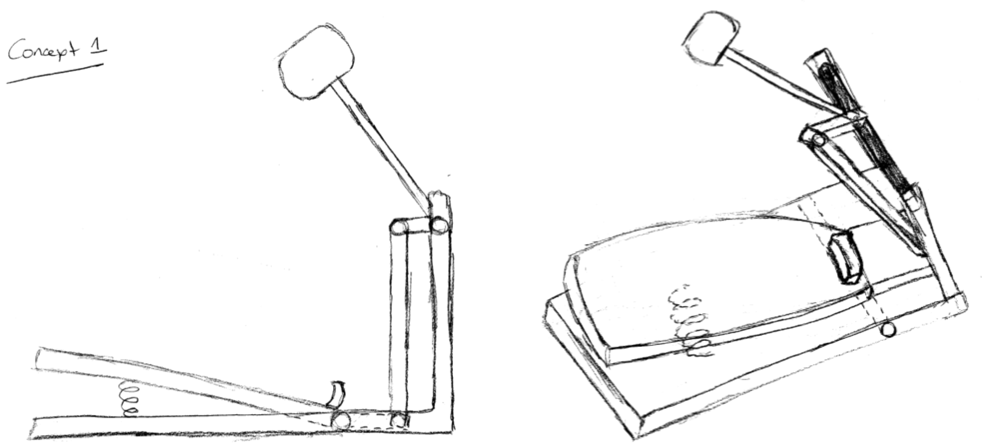


Figure 24: Concept 1: 4-Bar Rigid Linkage Design

Concept 1, seen in Figure 23, is a rigid linkage system that uses the front part of the footboard as a bell crank to push the vertical link upwards and rotate the shaft with the beater attached to it. When the pedal is released the spring underneath the footboard pushes the system back towards the starting position. This concept is simple, can meet the design specifications, and variations have been used in existing designs so it is a good possibility for the team's design.

Concept 2

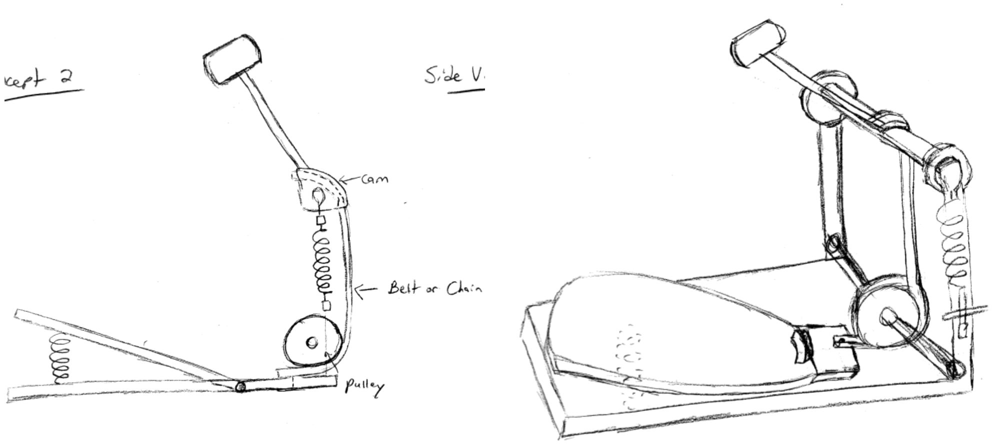


Figure 25: Concept 2: Chain and Pulley Design

Concept 2, seen in Figure 24, is a combination of Concept 1 and a standard bass drum pedal. It incorporates the bell crank from Concept 1 and a pulley so that it can use the belt, or chain, and cam system from a standard pedal. The bell crank pulls on the attached belt or chain and uses the pulley as a guide so that the chain or belt rotates the cam and beater. The spring attached to the beater shaft keeps the system in tension and helps return it to the starting position, along with the spring under the footboard, when the footboard is released. This concept can also meet the design specifications and is a good option for the team's design.

Concept 3

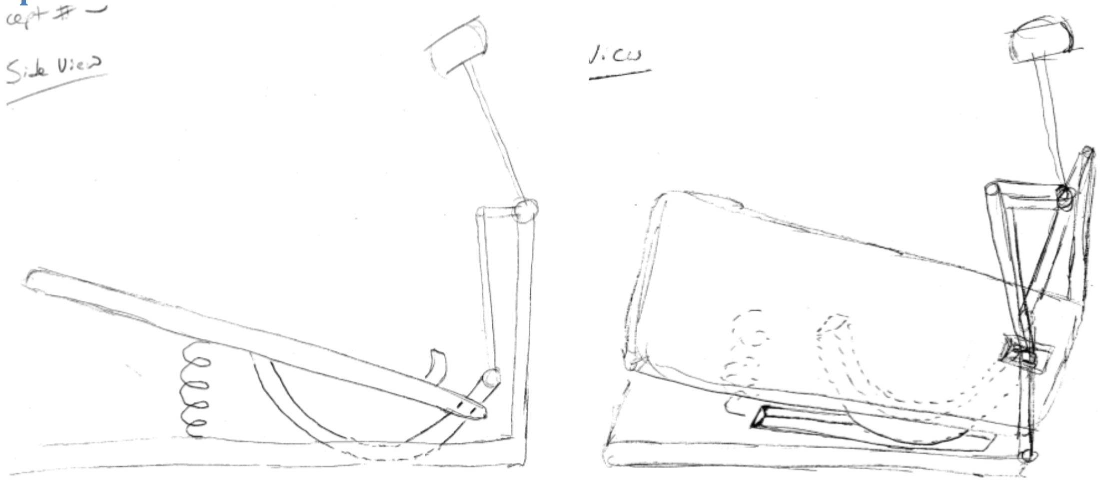


Figure 26: Concept 3: 4-Bar Linkage With Curved Input Link

Concept 3, seen in Figure 25, is similar to Concept 1 because it is a rigid linkage, but it uses a different kind of input link. This concept uses a curved link that sits in a slot cut into the base plate to guide its motion. The curved crank pushes up on the vertical link thus rotating the beater towards the drumhead. The spring underneath the footboard is again used to return the system to the starting position. This design, although very similar to Concept 1, overcomplicates the idea and will most likely not have any significant advantages so it will not be used.

Concept 4

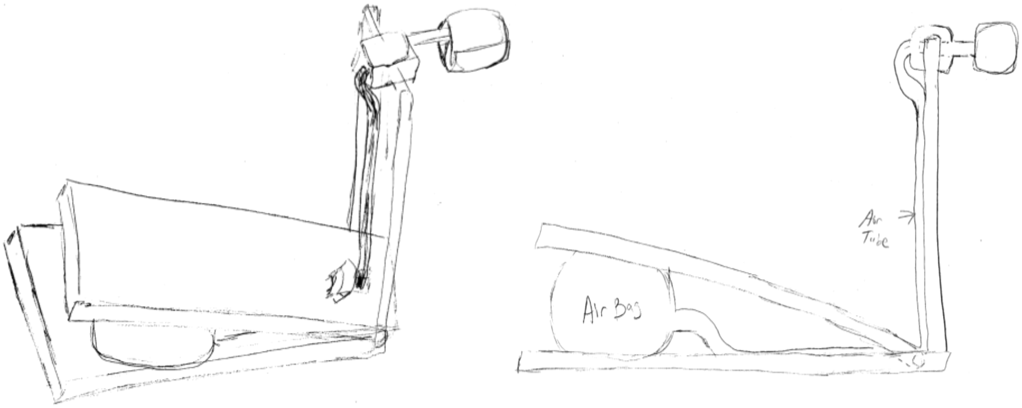


Figure 27: Concept 4: Pnematic Design

Concept 4, seen in Figure 26, works as a pneumatic system. When the footboard is pressed downwards it forces air through a tube and the air pushes the beater straight out to strike the drumhead. There would be an intake hole on the back of the air bag to return the system to its starting position when the footboard is released. This design could be interesting but it has some flaws. The rate at which the air bag is able to inflate is a major issue because the pedal needs to be able to play at a certain speed. If the bag inflates too slowly the beater will not be retracted fast enough to play at high speeds. Another issue with this design will be setting up the pneumatics to push the beater in and out of its box. For these reasons this concept will not be selected.

Concept 5

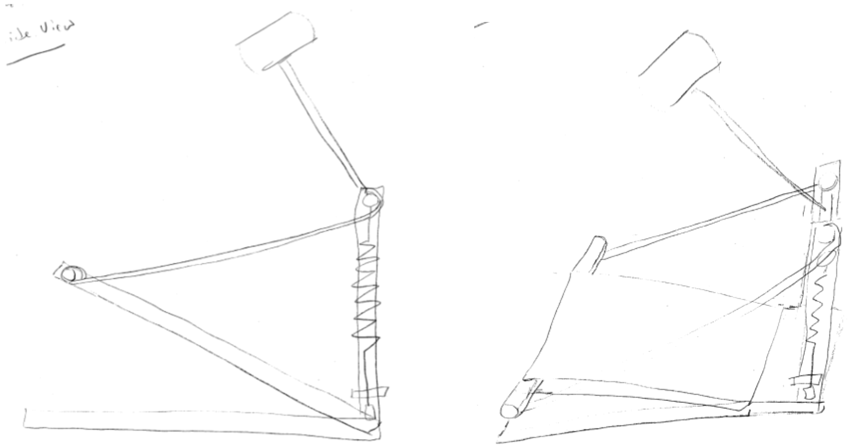


Figure 28: Concept 5: Side Band Design

Concept 5, seen in Figure 27, involves two bands that are attached to both the footboard and the beater shaft. The bands are wrapped around the beater shaft so that when the footboard is pushed downward they uncoil and cause the beater to move towards the drumhead. The spring on the side acts to keep the system in tension and return it to its original position when the footboard is released. This concept could work but the team believes that the straps could interfere with the player's foot, and would probably become slack when the pedal was released due to the speed at which it needs to return to the starting position. For these reasons this design was not chosen for further exploration.

Concept 6

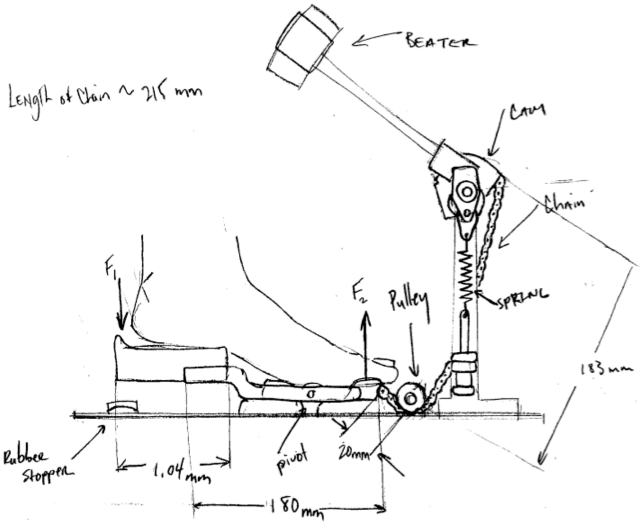


Figure 29: Concept 6: Alternate Chain and Pulley Design

Concept 6, seen in Figure 28 above, involves a chain that is attached to a lever arm, which makes it so that the device has some mechanical advantage. Thus the force F1 applied by the operator when transferred to F2 is much greater. The device uses a pulley in order to allow it to pull on the cam. A disadvantage to this system is that it has a lot of moving components and currently has no good way of protecting the user's toes from injuries. The spring on the side of the device keeps it in tension. The same design could be done with a strap rather than a chain, which would reduce the device's weight. There needs to be a solution to resolve the same tension problem as in the previous concept, so this will most likely not be a final design.

Concept 7

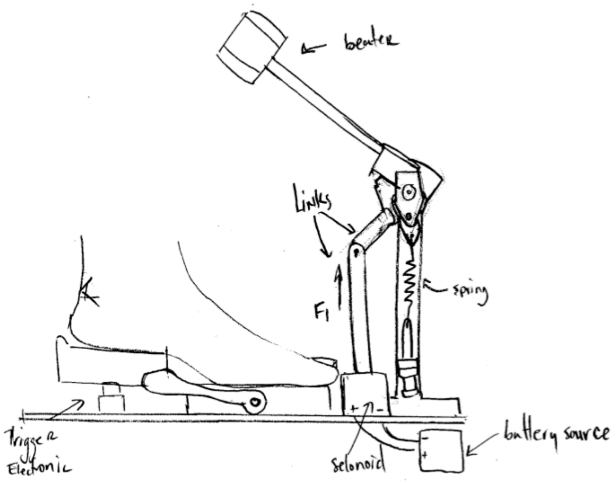


Figure 30: 4-Bar Linkage With Solenoid Design

This concept as seen in Figure 29 uses a solenoid. The solenoid would be used to drive the linkage system as shown by the force F1; this design requires an electrical system. The problem with this concept was that the response time would be delayed and the speed and strength with which the drumhead would be hit could not be changed. Thus this made Concept 7 impractical.

Choosing a Design

After considering each concept the team decided to choose a combination of two designs for further analysis. The two designs were Concept 1 and Concept 6. The rigid linkage idea from Concept 1 will be combined with the heel plate and side bar input link of Concept 6. These designs were chosen because of their simplicity, practicality, and most importantly because they can meet all of the design specifications. The combined design was subjected to a kinematic analysis in order to determine the optimum link lengths and ground pivot positions so that the mechanisms would meet the team's performance specifications.

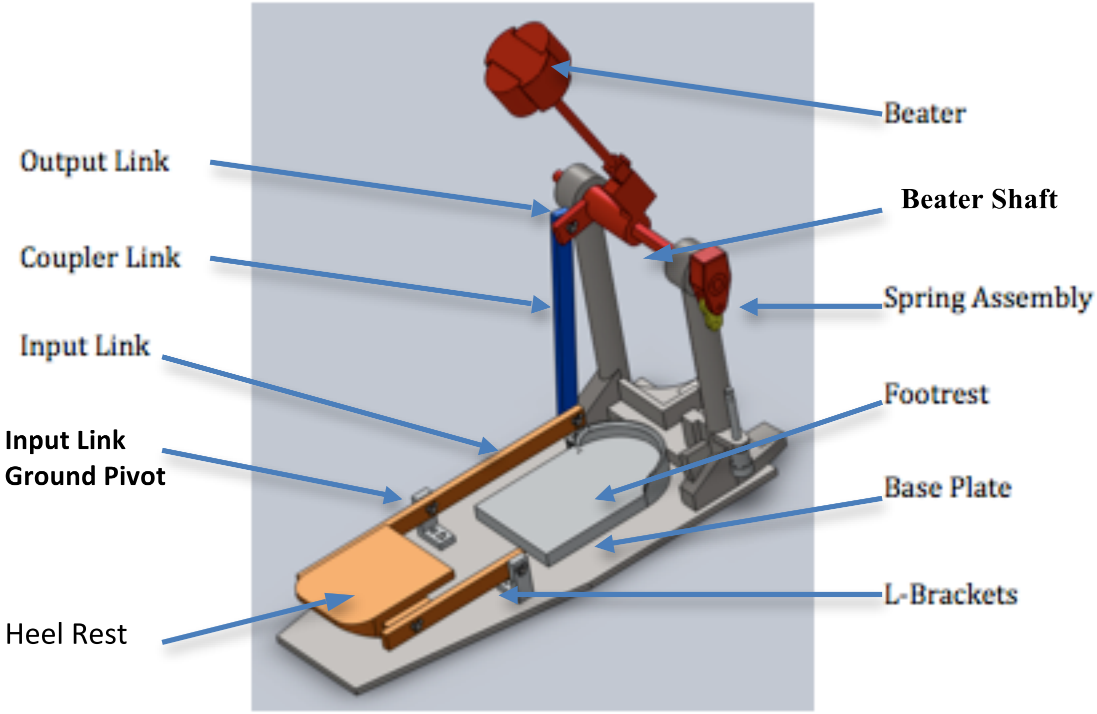
Developing the Concept

After choosing what the team considered to be the best design out of the seven initial concepts, a more in depth analysis was done to create the final model. The first step in this analysis was to determine the optimum starting angle of the player's foot. To find the optimum starting angle the team had each member place his foot on the ground then raise his heel to a position in which it felt comfortable. The angle from the point where the foot made contact with the ground along the bottom of the foot was then measured to obtain a possible starting angle. Analyzing the combined data from multiple tests of each person revealed that a starting angle between 6° and 10 ° would be ideal.

This range of starting angles was also equivalent to the possible input stroke, or range from the initial to final angle of the foot, of the pedal. A shorter stroke is ideal so that the drummer can play faster rhythms and use less energy. The actual determination of the stroke, however, must come after the link lengths are determined, and is significantly influenced by the input-output, or I-O, ratio of the pedal. This I-O ratio is described as the ratio between the angle of the footboard and the angle of the beater shaft. As mentioned in the design specifications, an input-output ratio of 1:3 is desired. In addition to the I-O ratio it is also necessary that the linkage have no toggle points. Toggle points are where the mechanism locks up and can get stuck, and they can be avoided by having good transmission angles throughout the stroke. Also mentioned in the design specifications, a good transmission angle is 60° or above.

The final consideration before constructing the preliminary 2-dimensional linkage was the vertical location of the ground pivot on the rotation shaft. Because the team planned on salvaging the uprights of an older bass drum pedal, the beater shaft ground pivot had to be 6" higher than the top of the base plate. With these requirements established the team then used Program Four-bar to iterate through several designs until an optimized linkage was developed. The optimized link lengths were as follows: input link– 5.75", coupler link – 5.5", output link– 1.0", and ground link– 9.0". The input-output ratio of this linkage was 1:5.9, and the transmission angles ranged from an initial angle of 62° to a maximum angle of 90°. Having met both the I-O ratio and transmission angle specifications, this linkage was then made into a SolidWorks model.

The final SolidWorks model, shown in Figure 30, has each part labeled and will be referenced when explaining the design.



The beater shaft and the parts that attach to it are similar to the standard bass pedal that was examined during the benchmarking process, and thus will not be discussed again in detail. The only major difference is that instead of being chain driven, the beater shaft is rotated by the optimized rigid linkage, which attaches directly via the output link. The beater was placed directly in the center of the beater shaft, between the upright supports, and the output link and spring on opposite sides, to balance the forces and therefore the bending moment on the shaft. This keeps the beater shaft from twisting in an unwanted direction while the pedal is in use.

The output link is connected to the coupler link, which is then connected to the input link. These individual links are connected to each other using pin joints and bushings so that they may rotate along the z-axis and translate in the x and y-directions. The input link is extended past its pivot and connects to the heel plate so that a downward force from a player's heel on the plate directly causes the linkage to move and the beater to rotate towards the drumhead. A shorter bar on the opposite side of the input link is also connected to the heel plate and both are attached to individual L-brackets. This was done to increase the stability of the mechanism. The optimal starting angle determined from the initial analysis is incorporated into the model as the angle between the top of the heel plate and the top surface of the footrest.

The heel plate and footrest were designed as special features that can adjust to the person using the pedal. The heel plate adjusts automatically because it is fastened using a single pin joint that allows it to rotate and adjust to the angle of the player's heel as it goes through the stroke. This feature is significant because it increases the efficiency of the force from the player's heel by ensuring that it is always in surface contact with the heel plate. The player does not need to position his/her foot/toe joint in line with the ground pivot of the input link. The footrest is also adjustable but the player must do so manually. The front-to-back position of the footrest can be altered by screwing it into different holes located on the bottom of the base plate. These holes allow the footrest to be moved in half-inch increments to accommodate players with different foot sizes. This feature makes the pedal more universal and increases the range of people it can be marketed to.

After completing the design in SolidWorks a force analysis needed to be done in order to make sure the pins and coupler link could withstand the forces being applied to them. To determine these forces a three-segment free body diagram analysis was done on the entire design, representing each of the moving links in a separate free body. The free body diagrams and calculations for this analysis can be found in Appendix C.

The first of the three segments to be analyzed was the beater shaft. The beater shaft had two known applied forces, the drumhead force, as measured in the experiments described in the background chapter, and the spring forces. Using these in the equilibrium equations allowed the team to find the forces in the coupler link and both bearings. The coupler link was assumed to be a two force member and therefore had the same force, but in different directions, at the top and bottom of the link. The force from the coupler link on the third segment, the input assembly, could then be used to find the remaining three unknown forces on the input link. These were the forces in each of the pins as well as the input from the player's foot.

All of the forces found to be acting on the different parts of the pedal can be seen in Table 5.

Table 5: Forces Acting on the Heel-Operated Pedal

Part	**Force X-Direction (lbf)**	**Force Y-Direction (lbf)**
Beater	8.16	0
Coupler Link	7.32	69.62
Spring	3.60	23.4
Bearing 1	5.68	24.33
Bearing 2	0.64	68.67
Pin 1	0.00023	58.72
Pin 2	7.32	148.69
Foot	0	137.8

With all of the forces now found a shear stress and tear out force analysis was done on the pins and a buckling force analysis was done on the coupler link. In order to ensure that the pedal would not break, a safety factor of three was used when determining whether or not the design was acceptable. All of these analysis calculations can be found in Appendix C.

For the pedal it was decided that the pins be made out of 1018 steel so that they would be strong and could easily rotate within the 6061 aluminum links. The links were chosen to be aluminum so that they were lightweight and could move quickly, but were also strong and sturdy. The material properties of the pins and coupler link were used when determining the stresses in each.

The shear stress on the pins was the first analysis done and the yield stress of 1018 steel, divided by three, was used as the maximum allowable stress on the pins. All three of the pins passed this analysis and were then analyzed for tear out force. The force in each of the pins was compared to the maximum allowable tear out force and pin 1 along with the coupler link pin passed the safety factor, but pin 2 did not. However, since the force in pin 2 was so close to passing, within four lbfs, it was decided that it was close enough to be acceptable; this reduces the safety factor to 2.9. The data from the pin analyses can be seen in Table 6.

Table 6: Pin Force Analysis Data

	Max Allowable Shear Stress (psi)	Calculated Shear Stress (psi)	Safety Factor	Max Allowable Tear Out Force (lbf)	Calculated Tear Out Force (lbf)	Safety Factor
Pin 1	55,986	531.7	105.3	437.4	58.7	7.5
Pin 2	55,986	4043.8	13.8	437.4	148.9	2.9
Coupler Pin	55,986	1901.4	29.4	437.4	70.0	6.3

The buckling force on the coupler link was the other significant factor to consider when determining whether or not the pedal could handle the forces acting on it. The critical force, or maximum force the 6061-aluminum link can handle without buckling, was calculated to be 160,156 lbf. Including the safety factor this force was lowered to 53,385 lbf. From the free body diagrams the team had found the total force on the coupler to be just 70 lbf and thus it will not buckle when the pedal is played.

SolidWorks Model Dynamic Analysis

As a result of the benchmarking that included ergonomics, the team conducted an analysis of the standard pedal, done prior to building the prototype. With this information we were able to create a mechanism model of the heel-operated bass drum pedal in SolidWorks that is a very accurate representation of the physical prototype. In the model all of the proper forces, contact surfaces, gravity, and springs that were needed were applied. All forces were based on the forces obtained from the impact force result recorded using piezoelectric paper . The model was organized into four sub-assemblies: the beater shaft, represented in red; the base of the pedal represented in gray; the coupler link assembly represented blue; and heel assembly represented in orange. See Figure 32.

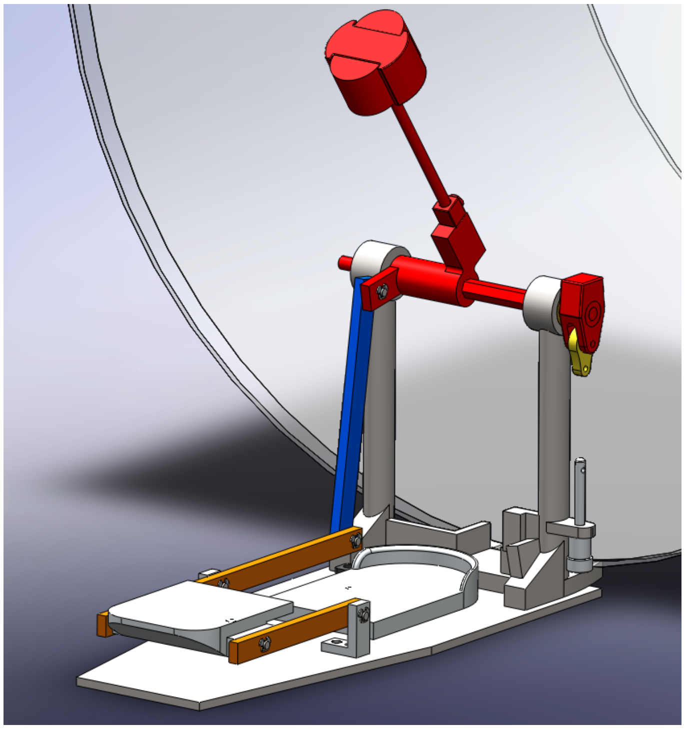


Figure 32: SolidWorks Model

In the model two step forces were applied on the heel rest. These forces were equivalent to the force that was calculated using our free body diagram equations of the heel-operated pedal. The forces are in the positive y- direction and in the negative y- direction as represented by the blue arrows in the figure below.

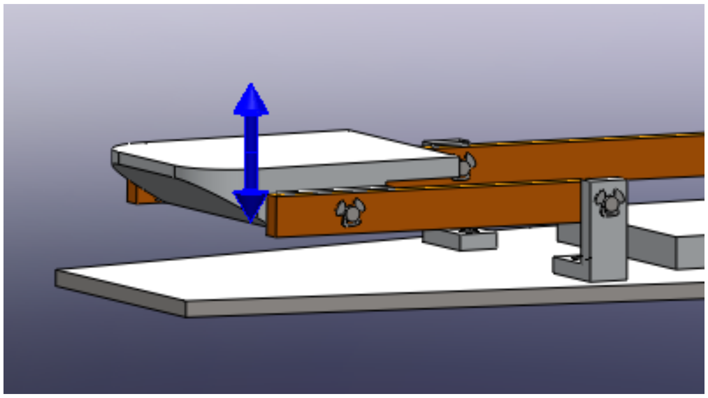


Figure 33: Heel Forces

The applied forces are shown below; the figure to the right shows the inputs for the forces in the negative y direction and the left figure for the forces in the positive y direction.

Face<1>@HeelRest-1

Force relative to:
● Assembly origin
○ Selected component:

Step
F1 0 lbf
t_1 0.00s
F2 8.5 lbf
t_2 0.00025s

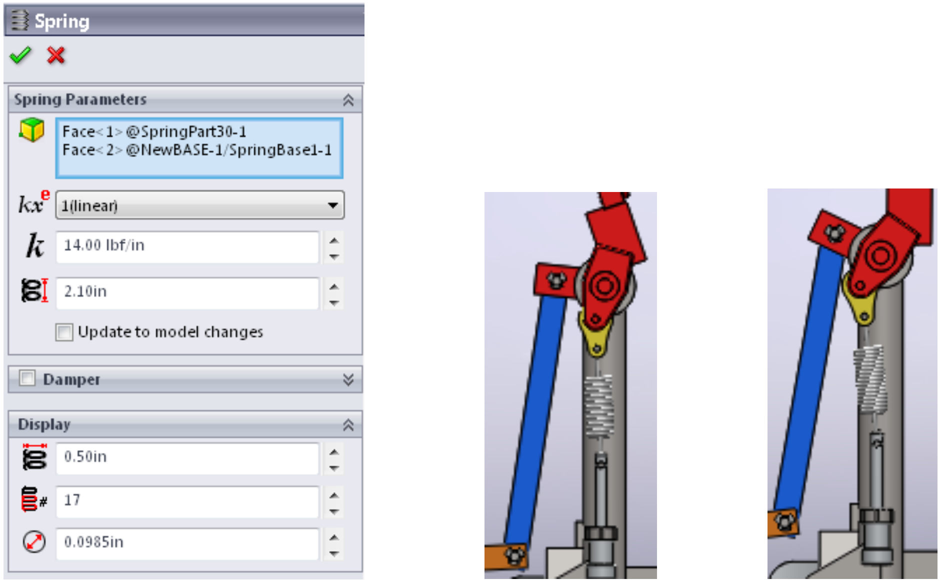
Face<1>@HeelRest-1

Force relative to:
● Assembly origin
○ Selected component:

Step
F1 13.95 lbf
t_1 0.00065s
F2 0 lbf
t_2 0.007s

Figure 34: Force Inputs/Graphs

For the motion analysis a spring with a constant of 14 lbf/in and a free length of 2.1 inches, shorter than the 2.4 inches the spring would need to stretch to make sure that it was pretensioned. The configuration for this can be seen in Figure 35 below.



Spring

Spring Parameters

Face<1>@SpringPart30-1
Face<2>@NewBASE-1/SpringBase1-1

kx^e 1(linear)

k 14.00 lbf/in

2.10in

☐ Update to model changes

☐ Damper

Display

0.50in

17

0.0985in

Figure 35: Spring Configurations

The model also included contact points between the drumhead and the beater head as represented by the blue color in Figure 36.

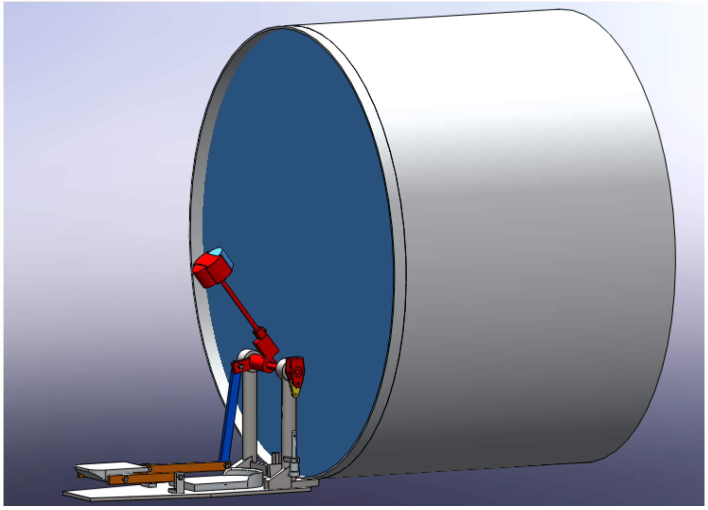
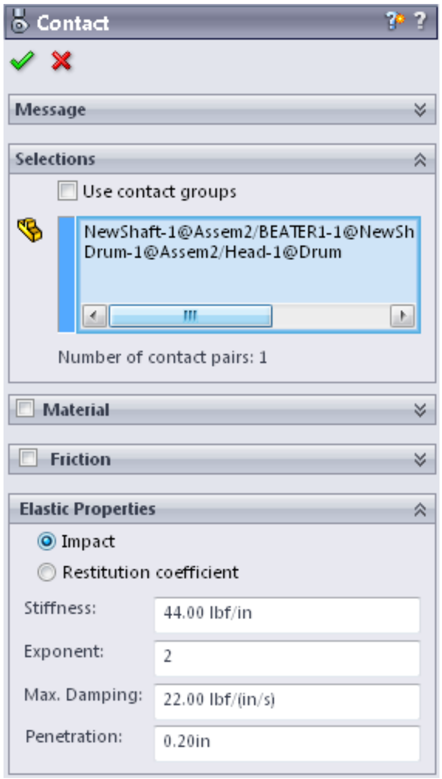


Figure 36: Contact Points on Beater and Drum Head

The material properties of the beater (synthetic felt) and the drumhead (PPT clear Plastic) were applied to the contact pair.



☐ Use contact groups

NewShaft-1@Assem2/BEATER1-1@NewSh
Drum-1@Assem2/Head-1@Drum

Number of contact pairs: 1

☐ Material

☐ Friction

● Impact
○ Restitution coefficient

Stiffness: 44.00 lbf/in
Exponent: 2
Max. Damping: 22.00 lbf/(in/s)
Penetration: 0.20in

Figure 37: Contact Configuration

Once these parameters were put into place, the motion analysis simulation of the model was run. The input forces were adjusted to simulate a playing speed of 330 bpm, which we determined as the peak tempo that a professional player can play. This equates to 0.18 seconds for a complete cycle of the beater head. The angular displacement, velocity, acceleration of the beater shaft and the impact force between the beater head and drumhead were calculated by the simulation. This can be seen in Figure 38 below.

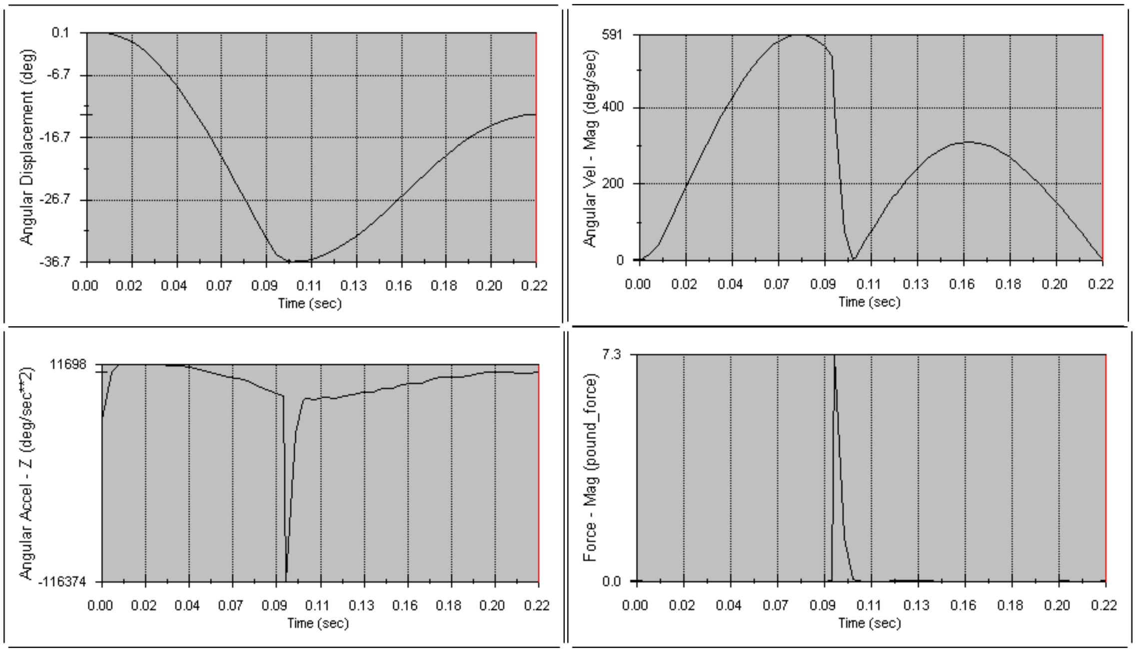


Figure 38: Simulation Graph Results

The total angular displacement from the rest position to impact the drumhead was 36.7 degrees. The peak angular velocity on the beater head just before impact was 591 deg/s. The maximum angular acceleration just before impact is 3130 deg/sec^2. The impact force was approximately 7.3 lbf as shown the in the graph in the lower right corner of Figure 38. All calculations were derived from using the cylindrical mate between the bronze bearings and the beater shaft shown in orange/purple in Figure 39.

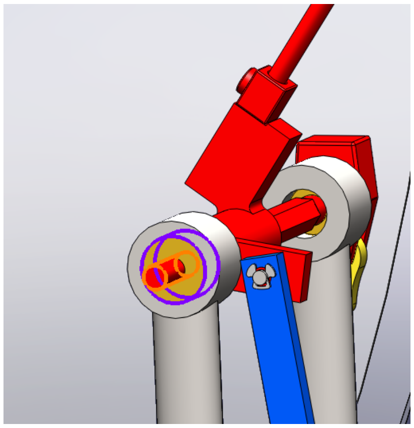


Figure 39: Concentric1 Mate

Manufacturing

The main support of the hex bar was salvaged from a pre-existing bass drum pedal because the only way to properly manufacture that part would be to create a cast iron mold. The hex bar was salvaged and modified from the same bass drum pedal along with the beater head and shaft. Our original design included a manufactured hex bar but we were incapable of manufacturing the parts with the equipment provided in the machine shop. The modified hex bar was originally designed for a double bass drum design. To accommodate for that problem, additional washers were added in between the set screw to fit tightly in place within the support. With there being a set screw, the hex bar needed to be oriented so when the beater head struck the drum membrane the set screw would tighten. Using this orientation of the hex bar prevented it from loosening upon striking the drum membrane. There were no modifications made to any of the other salvaged parts.

There were two materials used in the assembly of the pedal. Taking overall weight of the pedal into consideration, the base plate was manufactured out of steel to add weight and keep the pedal from sliding while playing. The toe plate, heel plate, u-bar, and all the links were manufactured out of aluminum 6061. Aluminum 6061 was chosen because after working out the calculations its characteristics gave us a safety factor of three. Aluminum 6061 also added the necessary weight to the pedal and still be light enough for the spring to effectively function.

The toe-plate was manufactured from a piece of stock that was 4x4x1.25 inches. This size stock was used so that it could be securely placed in the mini-mill. The machining of this part required two steps. In the first step of the machining process a drilling function was used to create the two holes used to adjust the location of the toe-plate on the base plate. Refer to Figure 40. Once the drilling operation was completed the machine changed to a 3/8 inch end mill and used a chain operation to cut away all the excess material around the raised toe stopper as well as the excess stock material. Once these operations were completed the part was taken to the band saw to remove excess material not cut off in the first step. The stock was chosen to have a height higher than the piece itself so it could be securely position in the mini-mill. Once the excess material was removed the toe-plate was placed back into the mini-mill upside down. From here another chain operation was used specifically to remove excess stock and cut the toe-plate down to its desired size.

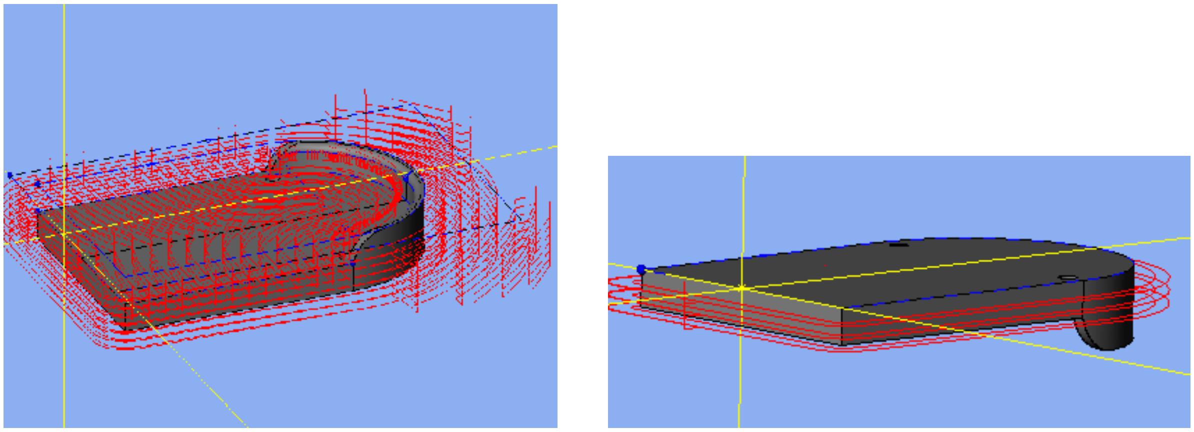


Figure 40: Toe Plate Esprit

The heel-plate was manufactured using a Haas mini-mill as well. The esprit file was based off a stock piece 5x3.5x1. Refer to Figure 41. After securely placing the part in the mini-mill the first operation used was to face the top of the block to create a smooth finished look on the top of the heel plate. After the facing operation a chain operation using a 3/8 end mill to remove excess stock around the heel-plate were necessary. The stock used was the exact width of piece to cut down on manufacturing time. Instead of using a drilling feature the screw holes were manually taped and drilled. The part was then placed back in the Haas mini-mill upside down. A wire operation was used to machine the part with the right angle so that the pedal would stay flush and level with the heel-plate.

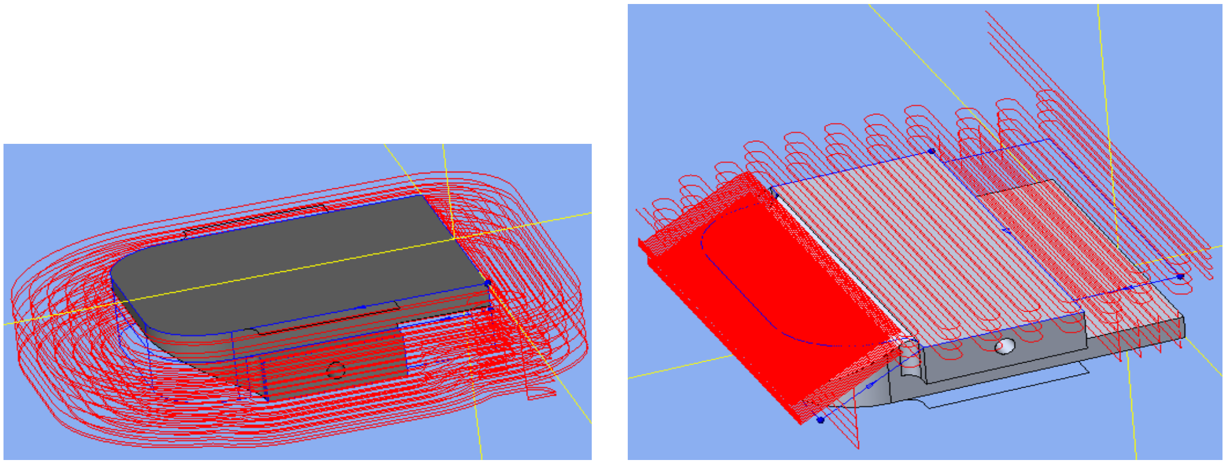


Figure 41: Heel Rest Esprit

The U-bar was made from a stock piece 4x4x1.25. To save on machining time the stock piece was cut down using a ban saw to the proper length and width of the piece. Refer to Figure 42. First a facing feature was used to face the stock piece to give it a finished look. Then a chain operation was used to remove the excess material in between the two U-supports. The part was then placed upside down to remove the excess material on the underside of the U-bar. All of the holes were manually tapped and drilled out.

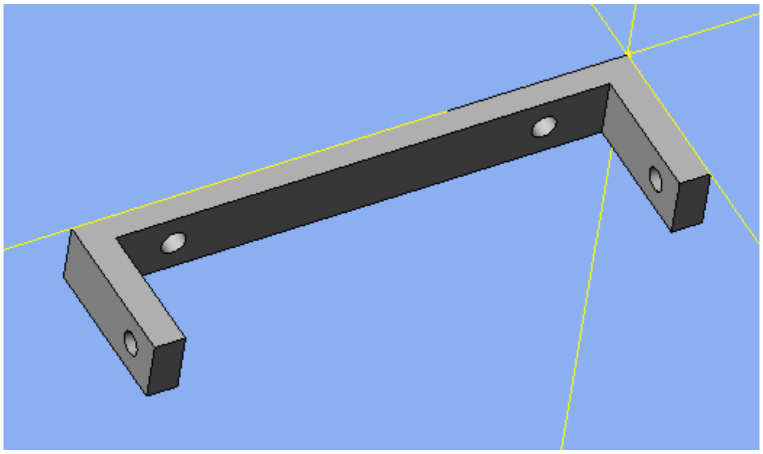


Figure 42: U-Bracket Esprit

Unforeseen Problems

One of the unforeseen obstacles that the team came across was in the manufacturing of turned parts. These particular parts were the bushing, pins, and hex bar. In the case of the bushings and the pins, the lathes that were available could not make such precision parts at such a small scale. As for the hex bar it was machined in the lathe but suffered tapering at the ends and thus did not allow for the bearings to fit as seen in Figure 43 below, caused by a lack of precision from the machines available.

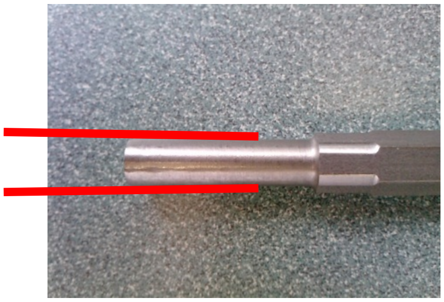


Figure 43: Hex Bar Taper

Shoulder bolts were used to replace the pins and bushings. The hex bar was scavenged from the uprights of an existing pedal and was modified for our prototype with a set screw, lock-rite and washer as seen in Figure 44 below.

Figure 44: Modified Salvaged Hex Bar

Since the tolerances on these parts were not precise, the mechanism exhibited more slop than desired. To remedy this we added a cross plate to the heel bars made from a strip of steel plate for support as shown in Figure 45 below.

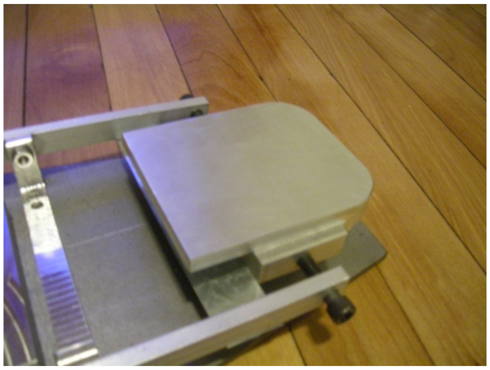


Figure 45: Support Modification

We also faced an issue with the width allow for the operator's foot, which we remedied by putting the heel bars to the outside of the u-bracket as seen in Figure 46.

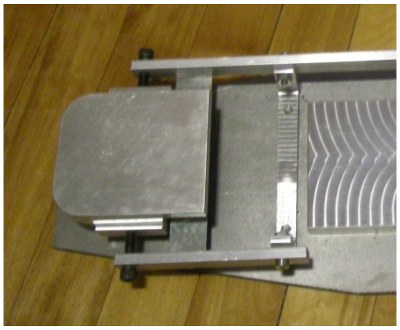


Figure 46: Modified Heel Bar Position

Recommendations

One major recommendation for future work on manufacturing would be to outsource as many of the components that require manufacturing that cannot be manufacture with the machines available, and to get more off the shelf parts and adapt them for the prototype. On the note of machining parts, the fabricator should determine whether equipment is available to manufacture the parts before deciding on manufacturing or buying off the shelf parts.

Testing

Testing the heel-operated prototype consisted of three elements. The first two, a pressure indicating film test and a tempo test, were repeats of those done on the toe-operated pedal. These tests were done as a quantitative comparison of the two pedals. The third test, a feedback test from drummers who have tried the heel-operated prototype, is a qualitative analysis of the pedal.

Pressure Indicating Film Test

The same test procedure as used for the toe-operated pedal was repeated with the heel-operated design. From the three tests we matched the color of the pressure paper to be approximately .4MPa, which corresponds to a net force of 7.25 lbf. This test showed the maximum force that the beater head could generate. Our design is capable of producing the same sound quality and loudness as a standard toe-operated bass drum pedal. While it wasn't able to generate quite as much force on the drum membrane, there is rarely any need for that much force to be generated while playing in a song.

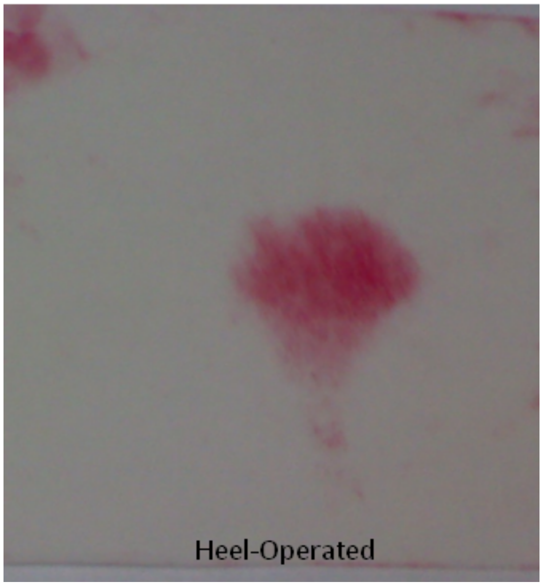


Figure 47: Heel Tested Pressure Paper

Tempo Testing

We tested to see how many beats per minute could be generated using a standard toe-operated bass drum pedal and our heel-operated design. To determine the maximum beats per minute, we had a drummer play as fast as he could for a 15 second time period while someone counted the number of beats. We then multiplied that number by four to get beats per minute. A drummer using the toe-operated pedal was capable of producing 330 beats per minute. The same drummer using the heel-operated design was capable of hitting the drum membrane at 270 beats per minute. An average song generally has a tempo of a 150 to 200 beats per minute. A fast temps song will have beats in the 240 beats per minute range. The heel-operated drum design is capable of performing at a fast enough tempo for use in most songs. Our design is capable of performing at the same level as a toe-operated pedal.

Drummer's Feedback

Three other drummers tested the pedal to provide feedback on how the pedal felt compared to the toe-operated pedal they use. One thing that all of the drummers liked about the pedal was that it was definitely more comfortable to play than a toe-operated pedal. All of the drummers involved use the heel up method of playing. They all felt that the heel-operated pedal didn't respond as quickly as a toe-operated pedal. To solve this problem they recommended adding a second spring underneath the heel plate to help return the beater head to its original position in a shorter time.

After assembling the pedal we realized that we should have modified the design to accommodate for the average width and length of the average foot. The test drummers felt it was a little snug with our design trying to play and occasionally their foot would be obstructed by the U-bar support. It was the general consensus that with the heel-operated pedal their leg never really felt tired or had any shin pain. For a song that involved a slower repetitive bass drum segment they thought the heel-design would be very beneficial to saving the drummer's stamina for faster up-beat songs. They really liked the fact that our design had an open heel plate area so if they wanted they could shift their foot back and use it exactly like a toe-operated pedal. They felt this would be very beneficial to beginner drummers because it takes a while to build up strong shin muscles, so with a heel-operated pedal it would allow them to get the feel of the rhythm. They all recommended it would be extremely useful if it were possible to incorporate the heel and toe-operated pedal all in one drum pedal.

Conclusions

The team was able to successfully design, manufacture, and test a functional heel-operated bass drum pedal. The pedal had performance specifications comparable to those of a standard toe-operated pedal and was more ergonomically friendly.

Recommendations

The size of the average human foot was overlooked in the original design of the bass drum pedal. The pedal should be able to accommodate someone with a size twelve or thirteen shoe. The width of the area where the foot rests should be widened to eliminate the foot potentially coming into contact with the U-bar and interfering with the stroke of the foot. To accommodate for these certain aspects a heel-operated pedal it will have to be slightly larger than a toe-operated pedal because the entire foot is resting on the pedal. With a toe-operated pedal the heel can just rest off the end of the pedal. Bearing and pins should be used instead of screws and bolts to decrease friction loss and increase the force capable of being produced. Not only would this increase but also create a more rigid linkage. The issue with using screws and bolts is they have the potential of loosening over time creating a weak unstable pedal. The toe-plate should be easily adjustable and not require an Allen Wrench to adjust the toe-plate back and forth along the base plate. To increase the beaters return back to the original position and readied for the next strike of the foot, placing a spring underneath the heel rest would be beneficial for increasing the beats per minute the pedal is capable of producing.

To record a more accurate pressure reading from the pressure paper, the use of multi-colored pressure indicating film could be used. There are companies that offer an analysis of the pressure paper which is significantly more accurate than eyeballing the color on the pressure paper to a pre-determined range of pressure on a graph. While this would be a little more expensive it would better the results and calculations. Taking all these considerations and putting them into practice would help create a better design for a heel-operated bass drum pedal.

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Appendix A: Foot and Toe Muscles

Table 7: Foot Muscles(Martini, 2010)

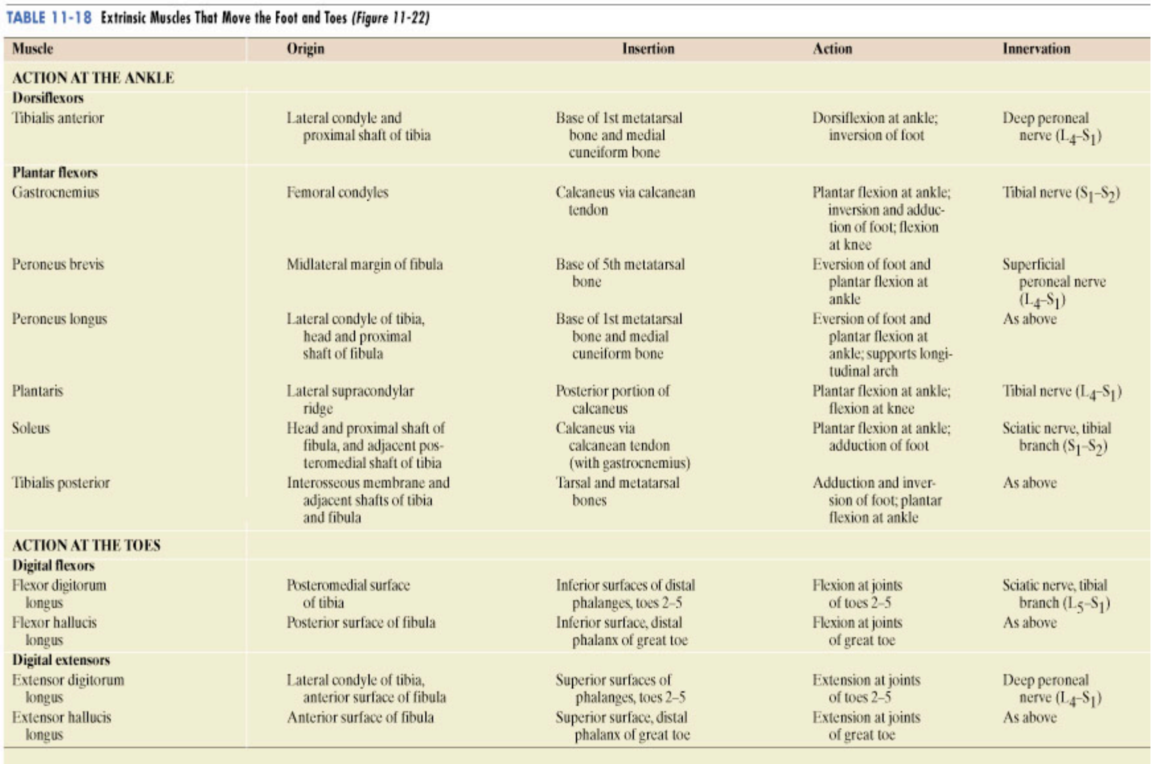


TABLE 11-18 **Extrinsic Muscles That Move the Foot and Toes** ***(Figure 11-22)***

Muscle	Origin	Insertion	Action	Innervation
ACTION AT THE ANKLE				
Dorsiflexors				
Tibialis anterior	Lateral condyle and proximal shaft of tibia	Base of 1st metatarsal bone and medial cuneiform bone	Dorsiflexion at ankle; inversion of foot	Deep peroneal nerve (L_4–S_1)
Plantar flexors				
Gastrocnemius	Femoral condyles	Calcaneus via calcanean tendon	Plantar flexion at ankle; inversion and adduction of foot; flexion at knee	Tibial nerve (S_1–S_2)
Peroneus brevis	Midlateral margin of fibula	Base of 5th metatarsal bone	Eversion of foot and plantar flexion at ankle	Superficial peroneal nerve (L_4–S_1)
Peroneus longus	Lateral condyle of tibia, head and proximal shaft of fibula	Base of 1st metatarsal bone and medial cuneiform bone	Eversion of foot and plantar flexion at ankle; supports longitudinal arch	As above
Plantaris	Lateral supracondylar ridge	Posterior portion of calcaneus	Plantar flexion at ankle; flexion at knee	Tibial nerve (L_4–S_1)
Soleus	Head and proximal shaft of fibula, and adjacent posteromedial shaft of tibia	Calcaneus via calcanean tendon (with gastrocnemius)	Plantar flexion at ankle; adduction of foot	Sciatic nerve, tibial branch (S_1–S_2)
Tibialis posterior	Interosseous membrane and adjacent shafts of tibia and fibula	Tarsal and metatarsal bones	Adduction and inversion of foot; plantar flexion at ankle	As above
ACTION AT THE TOES				
Digital flexors				
Flexor digitorum longus	Posteromedial surface of tibia	Inferior surfaces of distal phalanges, toes 2–5	Flexion at joints of toes 2–5	Sciatic nerve, tibial branch (L_5–S_1)
Flexor hallucis longus	Posterior surface of fibula	Inferior surface, distal phalanx of great toe	Flexion at joints of great toe	As above
Digital extensors				
Extensor digitorum longus	Lateral condyle of tibia, anterior surface of fibula	Superior surfaces of phalanges, toes 2–5	Extension at joints of toes 2–5	Deep peroneal nerve (L_4–S_1)
Extensor hallucis longus	Anterior surface of fibula	Superior surface, distal phalanx of great toe	Extension at joints of great toe	As above

Table 8: Thigh Muscles(Martini, 2010)

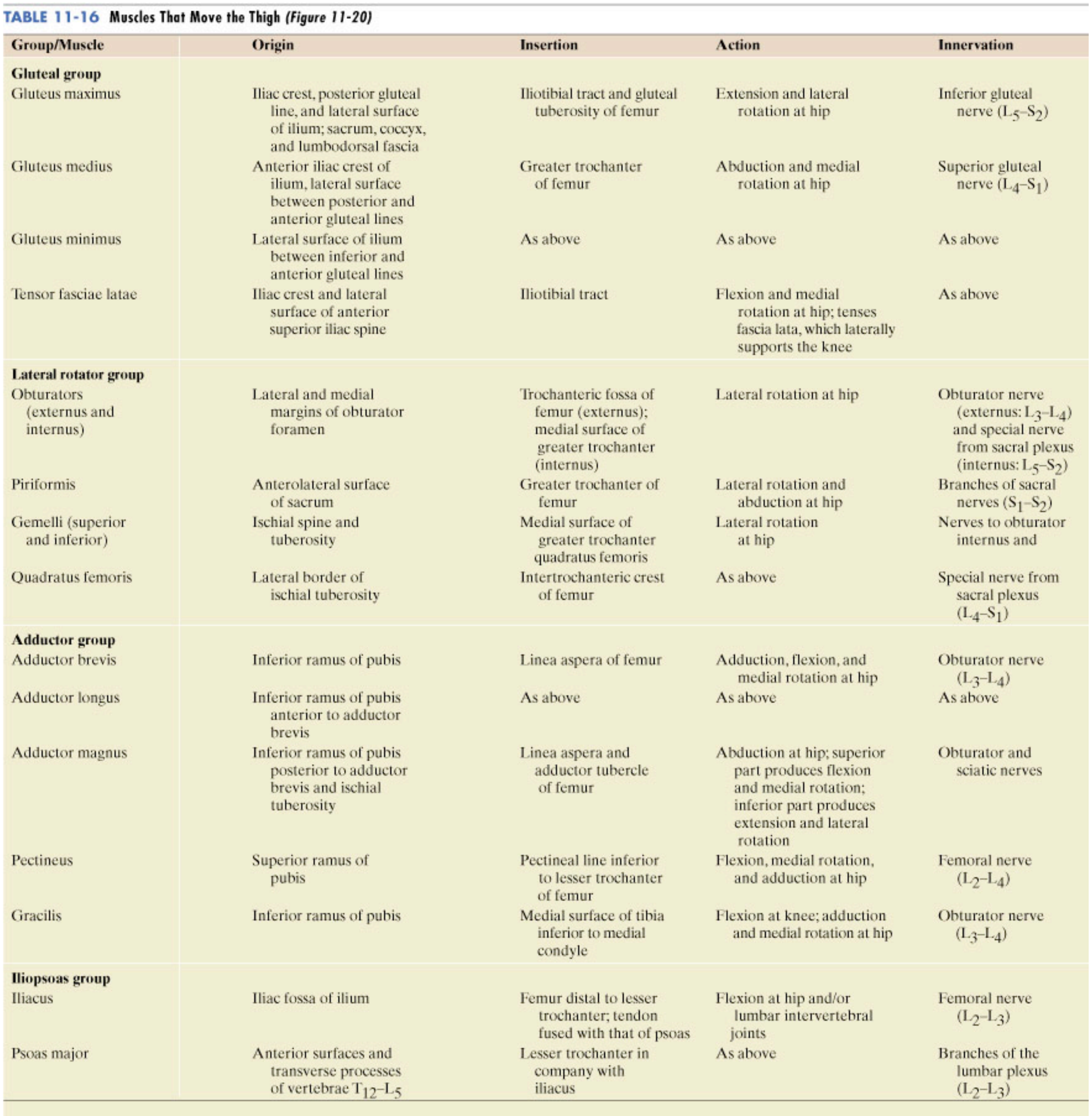


TABLE 11-16 **Muscles That Move the Thigh** ***(Figure 11-20)***

Group/Muscle	Origin	Insertion	Action	Innervation
Gluteal group				
Gluteus maximus	Iliac crest, posterior gluteal line, and lateral surface of ilium; sacrum, coccyx, and lumbodorsal fascia	Iliotibial tract and gluteal tuberosity of femur	Extension and lateral rotation at hip	Inferior gluteal nerve (L_5–S_2)
Gluteus medius	Anterior iliac crest of ilium, lateral surface between posterior and anterior gluteal lines	Greater trochanter of femur	Abduction and medial rotation at hip	Superior gluteal nerve (L_4–S_1)
Gluteus minimus	Lateral surface of ilium between inferior and anterior gluteal lines	As above	As above	As above
Tensor fasciae latae	Iliac crest and lateral surface of anterior superior iliac spine	Iliotibial tract	Flexion and medial rotation at hip; tenses fascia lata, which laterally supports the knee	As above
Lateral rotator group				
Obturators (externus and internus)	Lateral and medial margins of obturator foramen	Trochanteric fossa of femur (externus); medial surface of greater trochanter (internus)	Lateral rotation at hip	Obturator nerve (externus: L_3–L_4) and special nerve from sacral plexus (internus: L_5–S_2)
Piriformis	Anterolateral surface of sacrum	Greater trochanter of femur	Lateral rotation and abduction at hip	Branches of sacral nerves (S_1–S_2)
Gemelli (superior and inferior)	Ischial spine and tuberosity	Medial surface of greater trochanter quadratus femoris	Lateral rotation at hip	Nerves to obturator internus and
Quadratus femoris	Lateral border of ischial tuberosity	Intertrochanteric crest of femur	As above	Special nerve from sacral plexus (L_4–S_1)
Adductor group				
Adductor brevis	Inferior ramus of pubis	Linea aspera of femur	Adduction, flexion, and medial rotation at hip	Obturator nerve (L_3–L_4)
Adductor longus	Inferior ramus of pubis anterior to adductor brevis	As above	As above	As above
Adductor magnus	Inferior ramus of pubis posterior to adductor brevis and ischial tuberosity	Linea aspera and adductor tubercle of femur	Abduction at hip; superior part produces flexion and medial rotation; inferior part produces extension and lateral rotation	Obturator and sciatic nerves
Pectineus	Superior ramus of pubis	Pectineal line inferior to lesser trochanter of femur	Flexion, medial rotation, and adduction at hip	Femoral nerve (L_2–L_4)
Gracilis	Inferior ramus of pubis	Medial surface of tibia inferior to medial condyle	Flexion at knee; adduction and medial rotation at hip	Obturator nerve (L_3–L_4)
Iliopsoas group				
Iliacus	Iliac fossa of ilium	Femur distal to lesser trochanter; tendon fused with that of psoas	Flexion at hip and/or lumbar intervertebral joints	Femoral nerve (L_2–L_3)
Psoas major	Anterior surfaces and transverse processes of vertebrae T_{12}–L_5	Lesser trochanter in company with iliacus	As above	Branches of the lumbar plexus (L_2–L_3)

Appendix B: Free Body Diagrams and Detailed Calculations

Position Analysis Equations

① Position Analysis

$$a e^{i\theta_2} + d e^{i\theta_3} - c e^{i\theta_4} - b e^{i\theta_1} = 0$$

✓ $a = 10.75''$ ✓ $c = 1.5''$

✓ $b = 11.00''$ ✓ $d_0 = 4.09''$

✓ $\theta_1 = 34.9^\circ$? $\theta_3 = 90^\circ$

? $\theta_4 = 0$ $\theta_2 =$ input

$d = ?$

$$(a\cos\theta_2 + i a\sin\theta_2) + (d\cos\theta_3 + i d\sin\theta_3)$$

$$-(c\cos\theta_4 + i c\sin\theta_4) - (b\cos\theta_1 + i b\sin\theta_1) = 0$$

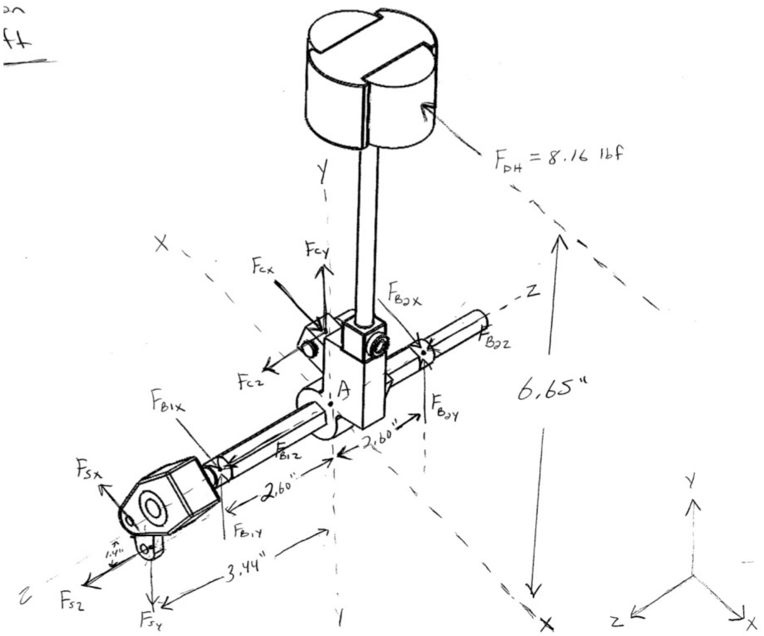
(1) $a\cos\theta_2 + d\cos\theta_3 - c\cos\theta_4 - b\cos\theta_1 = 0$ (x component)

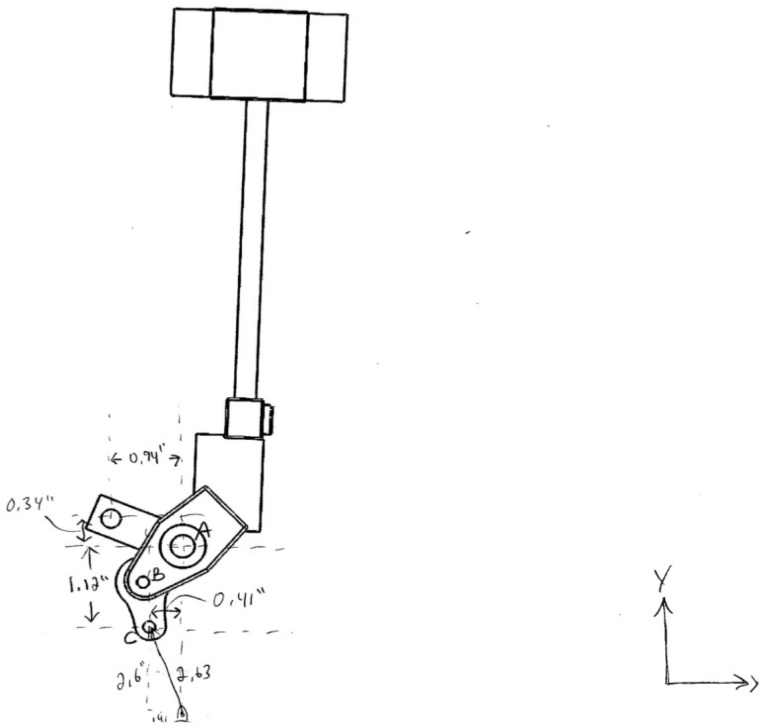
(2) $\cancel{i} a\sin\theta_2 + \cancel{i} d_0\sin\theta_3 - \cancel{i} c\sin\theta_4 - \cancel{i} b\sin\theta_1 = 0$ (y component)

(3) $d = d_0 - c\theta_4 - c(90 - \theta_3)$

$$\frac{dd}{dt} = -c\omega_4 + c\omega_3 = d'$$

Stress and Force Analysis Free Body Diagrams and Calculations





Rotation Shaft

$F_{sx} = 3.60\ lb$

$F_{sy} = 23.4\ lb$

$F_{DH} = 8,16\ lb$

$$\Sigma F_x = F_{sx} + F_{B_1x} + F_{cx} + F_{B_2x} - F_{DH} = 0$$

$$= 3,6 + F_{B_1x} + F_c \cos(84) + F_{B_2x} - 8,16 = 0$$

$F_{cx} = F_c \cos(84)$

$F_{cy} = F_c \sin(84)$

$$\Sigma F_y = F_{sy} + F_{B_1y} + F_{cy} + F_{B_2y} = 0$$

$$= 23.4 + F_{B_1y} + F_c \sin(84) + F_{B_2y} = 0$$

$F_c = 70\ lb$

$F_{B_1x} = 5,681\ lb$

$F_{B_1y} = -24,326\ lb$

$F_{B_2x} = +0,638\ lb$

$F_{B_2y} = -68.674\ lk$

$$\Sigma F_z = F_{sz} + F_{B_1z} + F_{cz} + F_{B_2z} = 0$$

$$\Sigma M_{Ax} = F_{B_2y}(2,6) \cancel{+} F_{cy}(0.94) - F_{B_1y}(2,6) - F_{sy}(0,41) = 0$$

$$= F_{B_2y}(2,6) + 70\sin(84)(0,94) - F_{B_1y}(2,6) - 23,4(.41) = 0$$

$$\Sigma M_{Ay} = F_{B_2x}(2,6) + F_{DH}(6,65) + F_{cx}(0,34) - F_{B_1x}(2,6) - F_{sx}(1,12) = 0$$

$$= F_{B_2x}(2.6) + 8,16(6,65) + 70\cos(84)(.34) - F_{B_1x}(2.6) - 3,6(1.2) = 0$$

$$\Sigma M_z = -F_{DH}(6,65) + F_{cx}(0,34) + F_{cy}(0,94) - F_{sx}(1,12) - F_{sy}(0.41$$

$$- 8,16(6,65) + F_c\cos(84)(0,34) + F_c(\sin(84))(.94) - 3,6(1,12) - 23,4(.41) = 0 \quad F_c =$$

$$\Sigma F_x = -3.6 + F_{B1x} + 70(\cos 84) + F_{B2x} - 8.16 = 0$$

$$\Sigma M_{Ay} = F_{B2x}(2.6) + 70(\cos 84)(.34) - F_{B1x}(2.6) + 3.6(3.44) = 0$$

$$\Sigma F_x = F_{B1x} + F_{B2x} = 4.443$$

$$\Sigma M_{Ay} = F_{B2x}(2.6) - F_{B1x}(2.6) = -14,872$$

$$F_{B2x} - F_{B1x} = -5,720$$

$$2F_{B2x} = -1,277$$

$$\boxed{\begin{array}{l} F_{B2x} = -0.638 \text{ lb} \\ F_{B1x} = 5,081 \text{ lb} \end{array}}$$

$$\Sigma F_y = 23.4 + F_{B1y} + 70(\sin 84) + F_{B2y} = 0$$

$$\Sigma M_{Ax} = F_{B1y}(2.6) - 70(\sin 84)(.5) - F_{B1y}(2.6) - 23.4(3.44) = 0$$

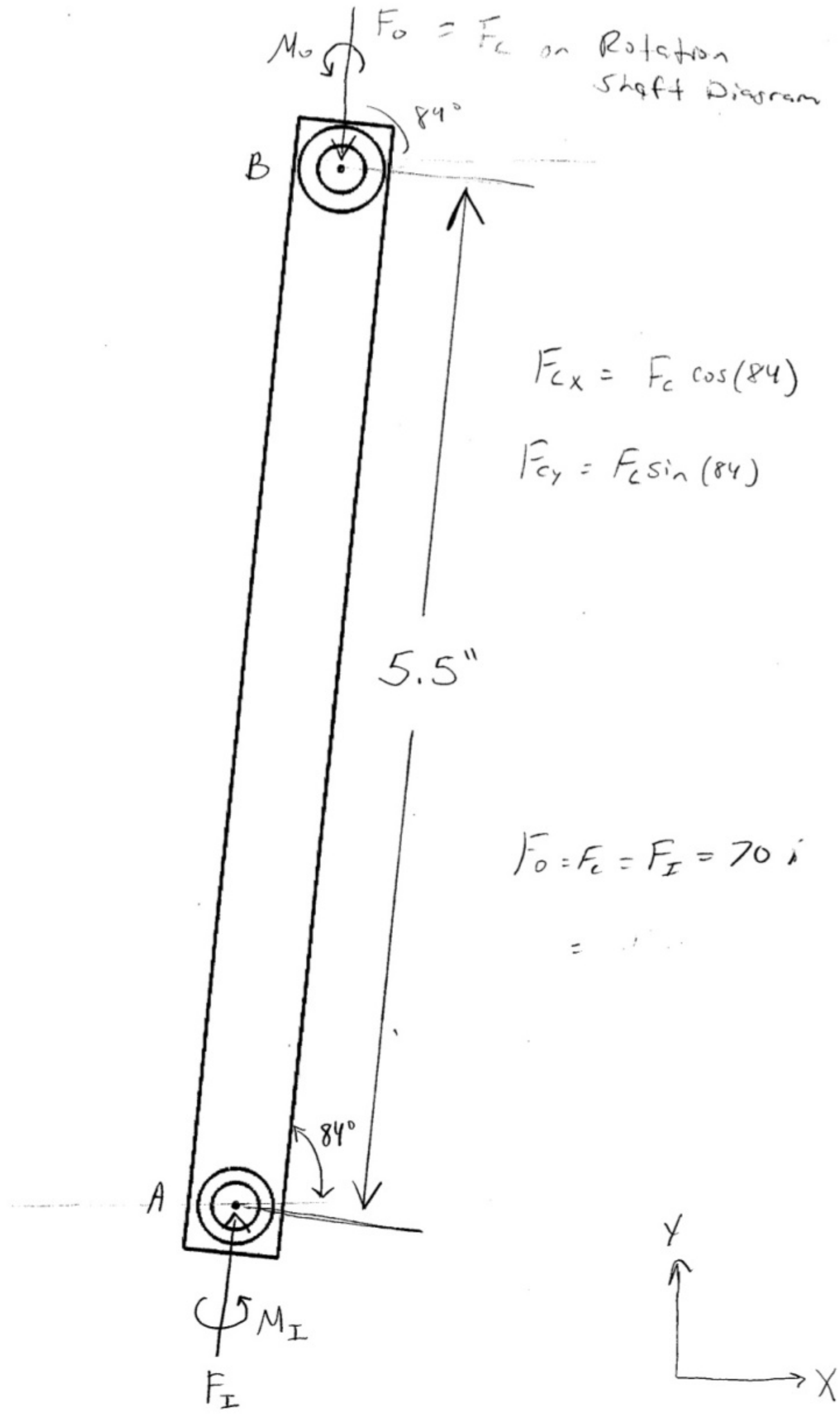
$$\Sigma F_y = F_{B1y} + F_{B2y} = -93 \text{ lb}$$

$$\Sigma M_{Ax} = F_{B2y} - F_{B1y} = 44.348 \text{ lb}$$

$$2F_{B2y} = -48.65$$

$$\boxed{\begin{array}{l} F_{B2y} = -24.326 \text{ lb} \\ F_{B1y} = -68.674 \text{ lb} \end{array}}$$

Coupler Link



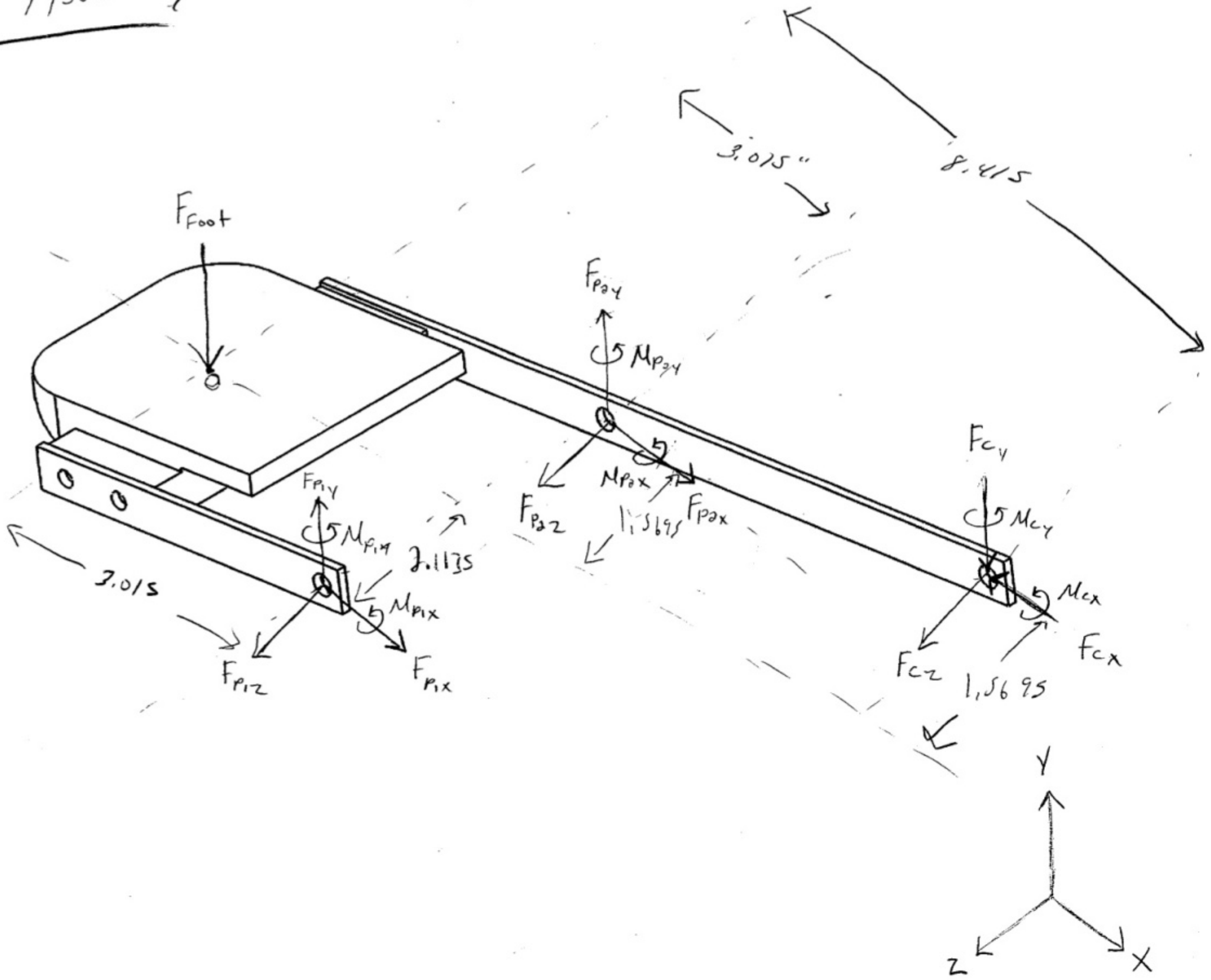
$F_{Cx} = F_C \cos(84)$

$F_{Cy} = F_C \sin(84)$

$F_O = F_C = F_I = 70\, i$

=

t Assembly



8.415
3.075"
F_Foot
F_P2y
M_P2y
M_P2x
F_P2z
F_P2x
1.5695
F_Cy
M_Cy
M_Cx
F_Cz
F_Cx
1.5695
F_P1y
M_P1y
2.1135
M_P1x
F_P1z
F_P1x
3.015
Y
Z
X

Input Link

$F_{c_x} = F_c \cos(84)$

$F_{cy} = F_c \sin(84)$

$F_c = 70 \text{ lb}$

$$\Sigma F_x = F_{P1x} + F_{P2x} - F_{cx} = 0$$

$$= F_{P1x} + F_{P2}x - 70 \cos(84) = 0$$

$$\Sigma F_y = F_{P1y} + F_{P2y} - F_{cy} - F_{Foot} = 0$$

$$F_{P1y} + F_{P2y} - 70 \sin(84) - F_{Foot} = 0$$

$$\boxed{F_{P2y} = 148.694 \text{ lb}}$$

$$\Sigma F_z = F_{P1z} + F_{P2z} + F_{cz} = 0$$

$$\boxed{\begin{array}{l} F_{P1x} = 2{,}270 \times 10^{-4} \text{ lb} \\ F_{P2x} = 7{,}317 \text{ lb} \end{array}}$$

$$\Sigma M_{\text{t}_{bar}x} = -F_{Foot}(1.5695) + F_{P1y}(3.683) = 0$$

$$\boxed{F_{P1y} = 58.723 \text{ lb}}$$

$$\Sigma M_{oy} = -F_{P1x}(2.1135) + F_{P2x}(1.5695) - F_{cx}(1.5695) = 0$$

$$-F_{P1x}(2.1135) + F_{P2x}(1.5695) - 70 \cos(84)(1.5695) = 0$$

$$\Sigma M_{P_1P_2 z} = -F_{Foot}(3.015) + F_{cy}(5.4) + F_{cx}(5.4) = 0$$

$$-F_{Foot}(3.015) + 70 (\sin 84)(5.4) + 70 (\cos 84)(5.4) = 0$$

$$\boxed{F_{Foot} = 137.8 \text{ lbs}}$$

$$\Sigma F_x = (F_{P1x} + F_{P2x} - 70 \cos(84) = 0)\, 2.1135$$

$$\Sigma M_{oy} = -F_{P1x}(2.1135) + F_{P2x}(1.5695) - 70 \cos(84)(1.5695) = 0$$

$$-F_{P1x}(2.1135) + F_{P2x}(1.5695) = 11.484$$

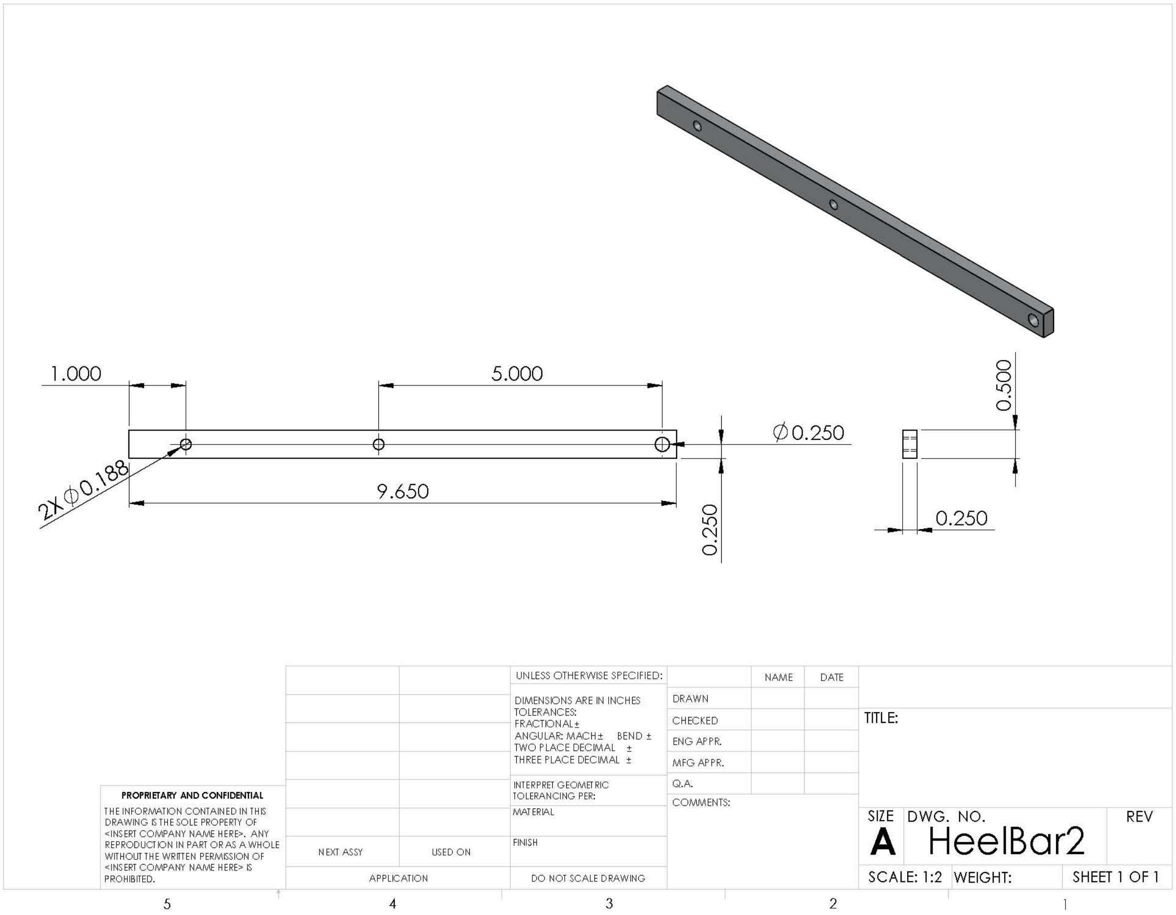
$$2.1135 F_{P1x} + 2.1135 F_{P2x} = 15.464$$

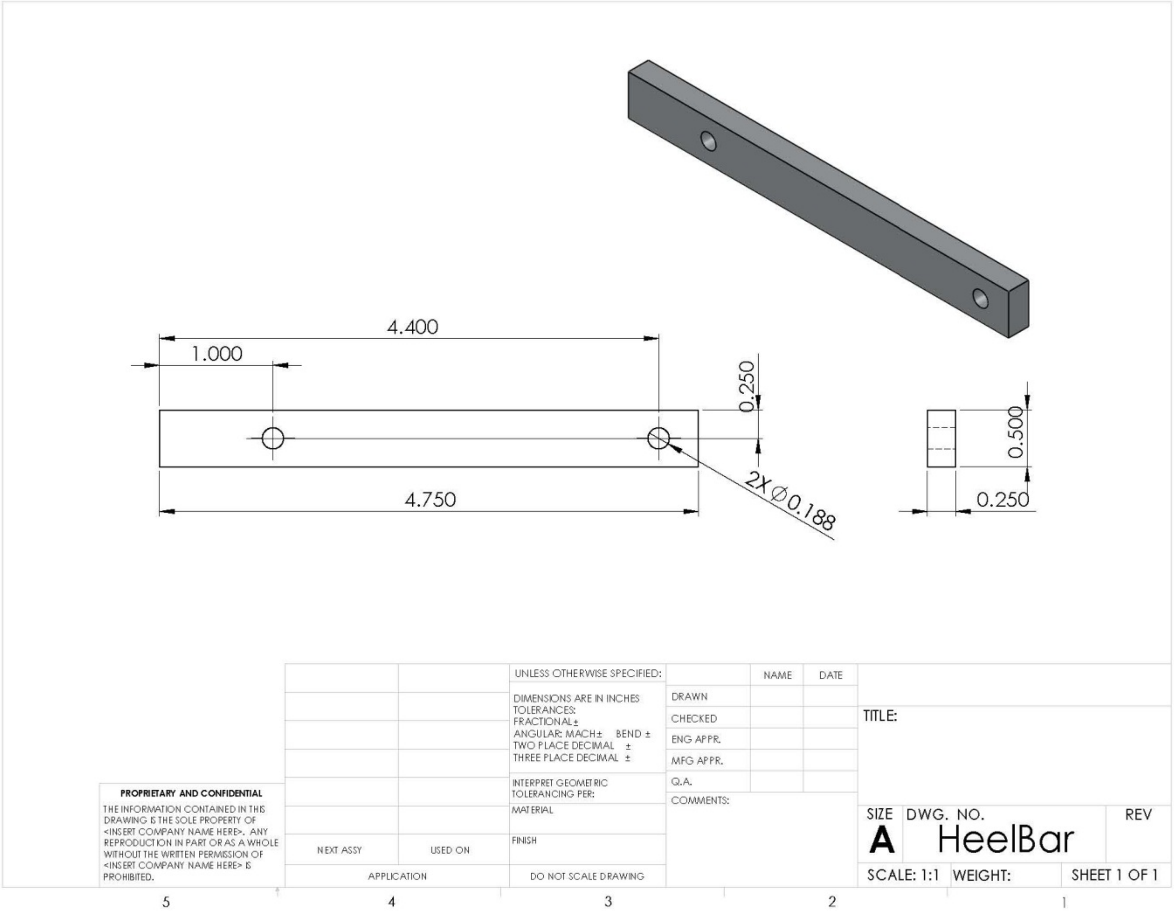
$$3.683 F_{P2x} = 26.9485$$

$$F_{P2x} = 7{,}317 \text{ lb}$$

$$F_{P1x} = 2{,}270 \times 10^{-4} \text{ lb}$$

Appendix C: Part Drawings




4.400
1.000
0.250
4.750
2X ⌀0.188
0.500
0.250
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
NEXT ASSY
USED ON
APPLICATION
PROPRIETARY AND CONFIDENTIAL
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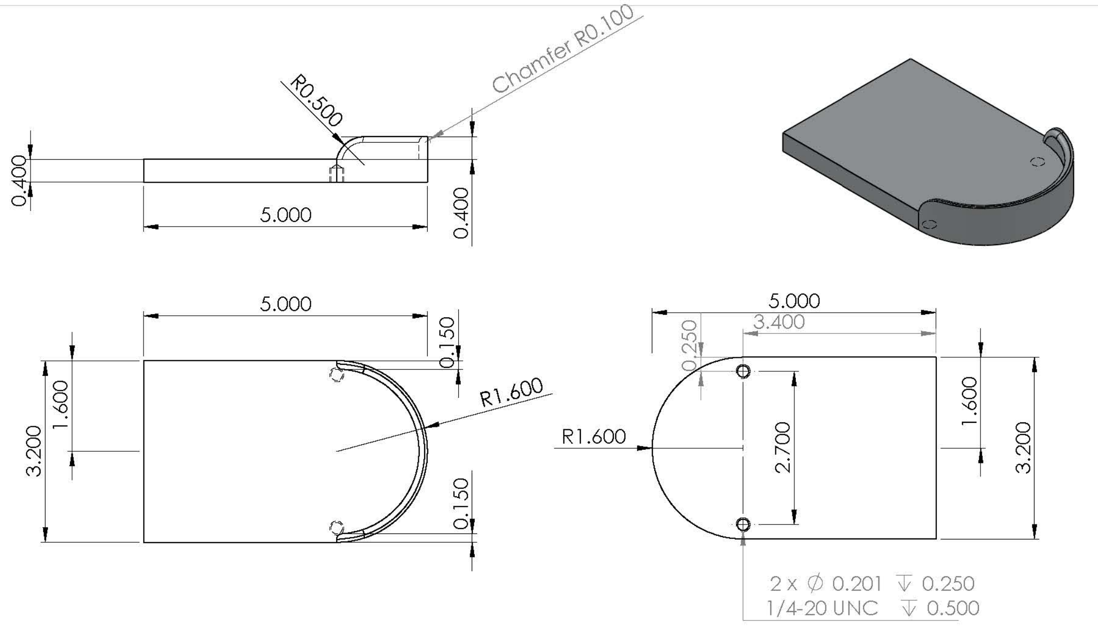
SIZE
A
DWG. NO.
HeelBar
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
5
4
3
2
1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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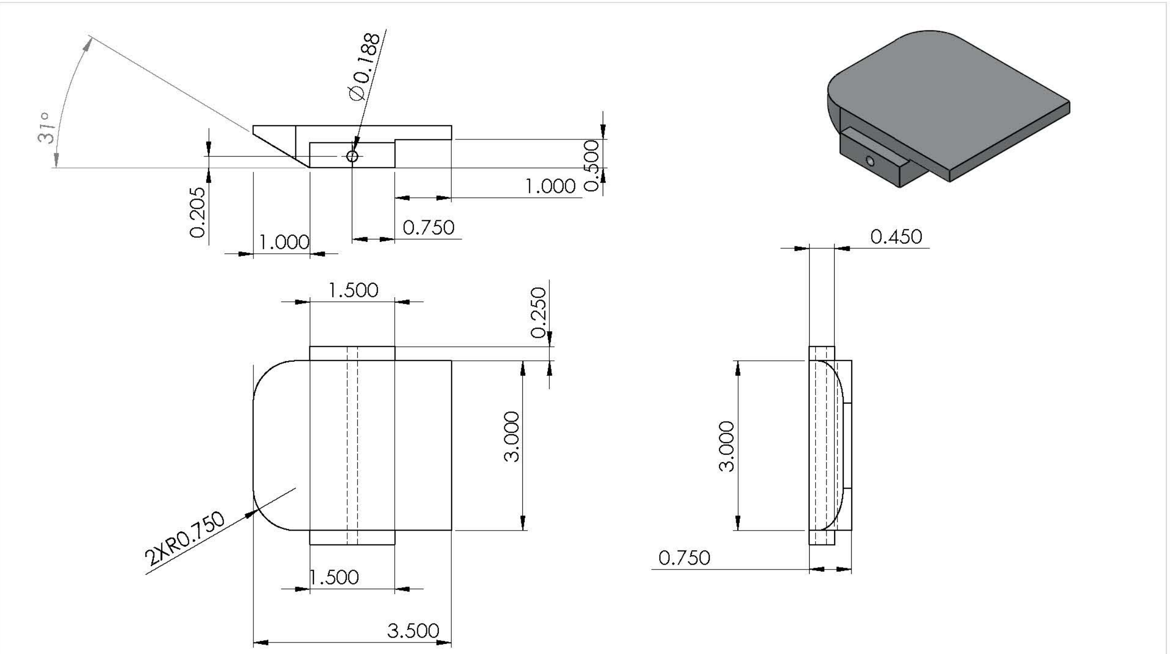
TITLE:

SIZE **A** DWG. NO. FootRest REV

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

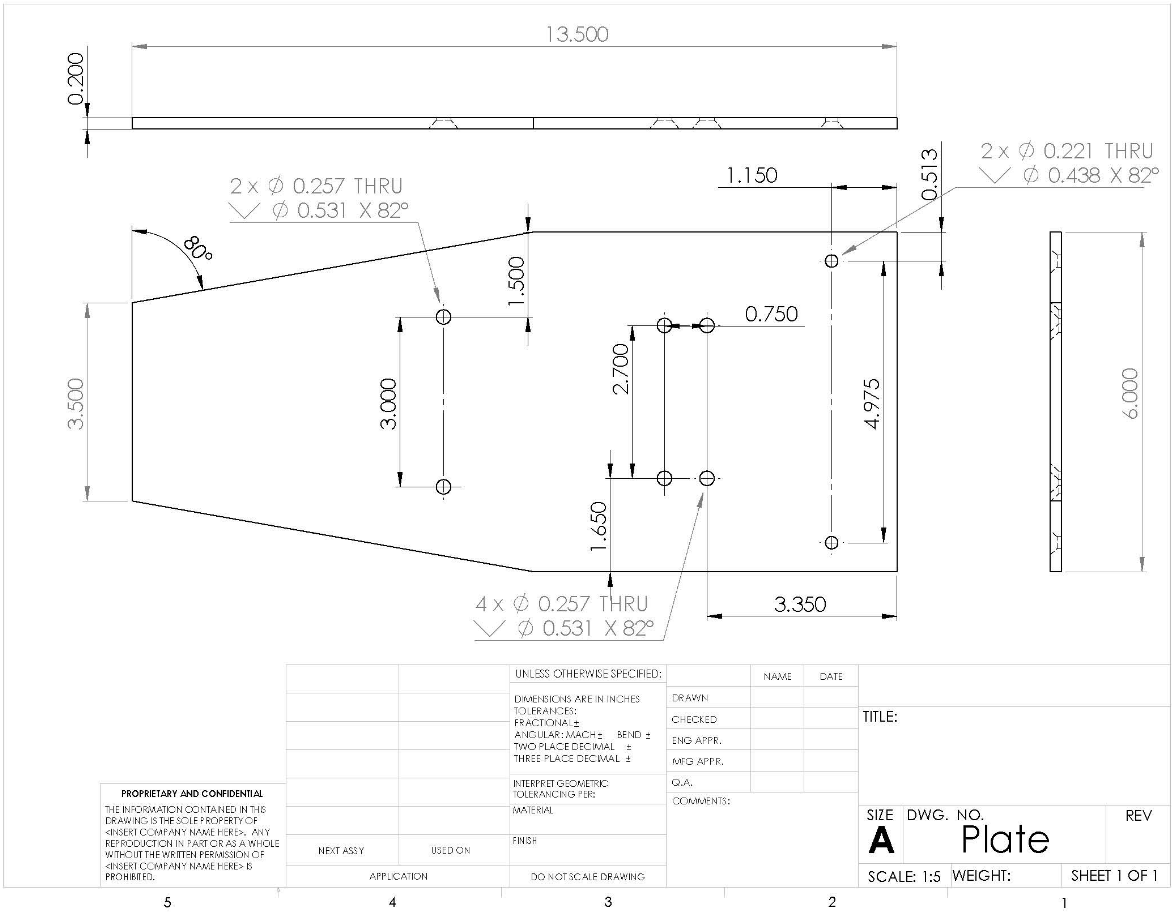
COMMENTS:

TITLE:

SIZE	DWG. NO.	REV
A	HeelRest	

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1


13.500
0.200
2 x ∅ 0.257 THRU
⌵ ∅ 0.531 X 82°
80°
1.150
0.513
2 x ∅ 0.221 THRU
⌵ ∅ 0.438 X 82°
1.500
0.750
3.500
3.000
2.700
4.975
6.000
1.650
4 x ∅ 0.257 THRU
⌵ ∅ 0.531 X 82°
3.350
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
A
DWG. NO.
Plate
REV
SCALE: 1:5
WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
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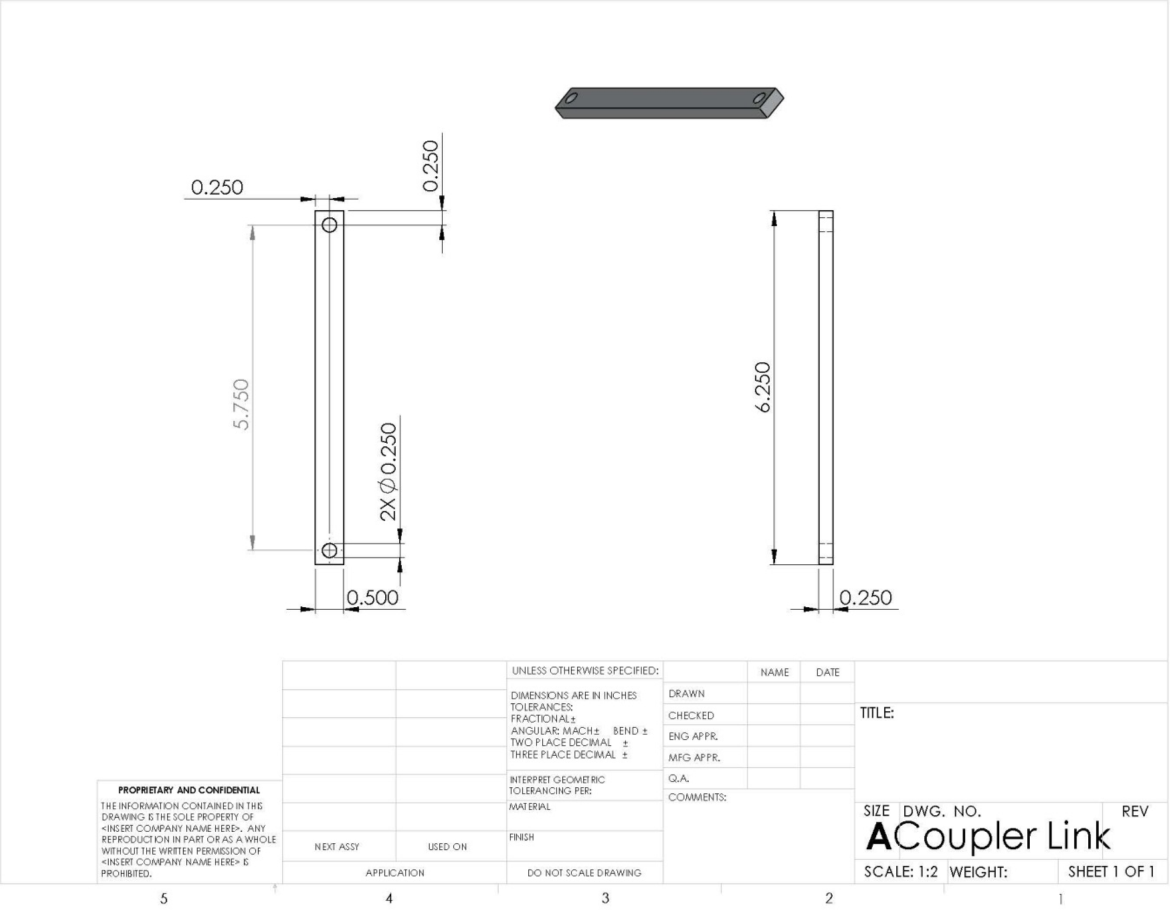
5
4
3
2
1


0.250
0.250
5.750
2X ⌀ 0.250
0.500
6.250
0.250
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
DWG. NO.
REV
A
Coupler Link
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
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5
4
3
2
1