Mechanical Engineering Project
305-463 D

Project: Ankle Testing Device

Final Report

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Section 1: The Design Problem

Design Brief

The objective is to complete a new experimental apparatus for REKLab, located in the Biomedical Engineering Department, by April 9, 2002. This device will allow measurement of the dynamic ankle stiffness of both ankles of humans in upright stance. Our project tries to simulate normal quiet standing of a person, at least as close as possible. An analogy of this is the ongoing motion of a person on a ship. That is what the apparatus we are building will simulate. The patient will stand perfectly upright. Once this is accomplished, the patient will hold onto the hand railings as a safety precaution. The personnel conducting the experiment will need to explain to the patient that countless random perturbations and/or disturbances will be applied to the actuator assembly, and thus to the feet of the patient. Once the patient is ready to proceed, the experiment begins and measurements of ankle torque and muscle contractions will be taken from various transducers positioned on the experimental apparatus. These measurements will be collected on the master computer that is inputting the various perturbations.

Pertinent Design Information

Space Restrictions:	Height of false ceiling	9’
	Height of concrete ceiling	10’ 6”
	Length of area provided	8’
	Width of area provided	4’ 8”
Rotary Actuator Requirements:	Supply pressure	3000 psi (20,684 kPa)
	Max Torque	3100 in-lbs (350 Nm)
	Max Velocity	20 rad/s
	Max frequency	>40 Hz (resonance) We chose 50 Hz
	Max Rotation	12˚
	Position amplitude	+/- 6˚
	Max acceleration	4000 rad/s
Material Requirements:	Cam	Steel
	Sliding Mechanism	Steel
	Remaining Components	Aluminum

Average Weight of Subject: 77 kg

Average Height of Subject: 1.8 m

Horizontal distance between feet: 10” - 16”

Maximum Fabrication Budget: $10,000

Evaluation Criteria

Customer Requirements:

Inexpensive: As this project is being constructed for the Biomedical Engineering Department headed by Professor Robert Kearney, it must be completed within the budget allocated for this project. This device will be constructed at a minimum cost and by a fixed deadline.

Durable: The Biomedical Engineering Department requires this device to be a permanent testing unit in the REKLAB. Hence, the device is designed to function adequately over a considerable length of time.

Safe to Use: As this device is intended for conducting experiments on humans, the equipment must operate with a high degree of reliability and safety. The device will be evaluated and approved by the Research and Ethics Board (REB).

Minimum Space: This equipment will be installed in a somewhat congested laboratory space where additional testing equipment will continue to be introduced in the future. Therefore, an efficient and compact design is required.

Easy Maintenance: There is a wide variety of sub-components involved in this testing apparatus, hence maintenance and availability of replacement parts is a necessity.

Reliability: The results obtained from this apparatus are intended for use in the studying of human muscle and reactions, thus the device must be constructed so as to generate results of the highest reliability and accuracy.

Easily Replaceable Parts: Should there be a need to replace a failed part or component, such replacement should be facilitated with components that are readily available from suppliers, manufactures or machined locally.

Aesthetics: As many of the people involved in the experiments will be volunteers, the test apparatus should provide comfort and be user friendly.

Design Criteria:

Easily Disassembled: This will ensure that repair and maintenance are an easy process.

Number of Parts: Parts must be kept to a minimum, as this generally affects the production time, mass, and ease of disassembly.

Parts Quality: High quality of parts ensures durability and reliability. However care must be exercised to avoid exceeding the allocated budget.

Material Mass: A greater mass would tend to reduce flexibility, as it would make it difficult to move the apparatus around easily. However, strength and durability must be balanced against overall mass. Moreover, the mass of the overall device is critical in determining the resonance frequency. Since we are expecting to work in the ranges of 0.1 to 20Hz, the resonant frequency should be minimized.

Production Time: To facilitate production, materials selected should be easily machined and components should have relatively short delivery time in order to meet the project deadline.

Flexibility: The device should be flexible in terms of the space it occupies, and in terms of mobility within that given space.

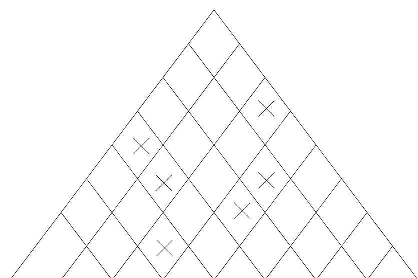
Strong Material: Strong materials will ensure both durability and reliability.

Material Resistance to Fatigue: As the apparatus is built to conduct repeated experiments, it must be constructed with materials that are resistant to fatigue and thus provide for durability.

House of Quality:

To determine the Characteristics needed in the design, the House of Quality is needed. This technique is used by weighing the customer requirements, (which are shown in the left most Column of the figure below), and then multiplying this by its opposite design restriction weight which is shown in the top row. For each design restriction there will be an overall grade, which gives a feel of how important that Characteristic is.

Examining the Chart below, we have that the Parts Quality is the most important factor.



	Weight	Material Resistance to Fatigue	Strong Material	Easily Disassemebled	Number of Parts	Parts Quality	Material Mass	Production Time	Flexibility
Inexpensive	5	10	10	6	8	15	9	15	7
Durable	8	20	20	13	13	18	16	15	16
Safe to Use	15	20	20	10	10	18	9	16	9
Minimum Space	6	6	6	13	16	6	19	12	19
Easy Maintenance	8	6	7	20	20	15	10	11	20
Reliability	8	18	17	13	12	20	14	13	13
Easily Replacable Parts	7	13	12	20	15	15	13	14	19
Aesthetics	1	4	4	4	4	4	4	4	4
Total	58	833	826	770	755	914	709	801	813

Section 2: Description of Solution

Safety Harness

The purpose of our design is to allow the measurement of the dynamic ankle stiffness in upright stance. In order to do this precisely and accurately, it is necessary for the patient to remain calm and feel safe. In the event where the subject might lose his/her balance and fall down, they could seriously injure themselves. This is why we have decided to include a safety device so as to prevent these injuries from occurring.

We weighed each of our options versus the requirements of the experiments to be performed. The experiment is made so that the patient's simulation is as realistic as possible. So, they must not rely on that tension. They must be free to behave normally in this experiment, touch a wall for example, and extend their arm as well with no problems at all. For this reason, it is important that the patient not feel constrained or restricted. They must be able to move freely and remain safe at the same time. After a long discussion amongst ourselves and with Professor Kearney, we to attach the patient to a harness suspended from the beams would be the ideal choice.

Fabrication Components (Steel):

We used steel for the CAM and the sliding mechanism (Tony'. Using the House of Quality for criteria, the main factor is the Quality of the Material. Since Ductility is not a factor in the design criteria the selection will be based on the Ultimate Strength of the Material and its resistance to rust. This is because the steel CAM is the final safety mechanism on the device and needs to have the highest safety factor. Thus we chose Stainless Steel Type W304.

Fabrication Components (Aluminum):

Aluminum was chosen to for the remaining components for the following reasons:

Advantages of Aluminum 6061 over Steel:

- Aluminum has a high strength: weight ratio. Steel is approximately 3 times as dense as aluminum is of course stronger. Thus if the strength of aluminum is adequate for this application, we would be saving on overall weight of the device without compromising on quality. A lower weight is desirable for manufacture and maintenance reasons.

- Aluminum is more weldable than steel, and since the base of the device is going to be welded together, aluminum will be the better choice.

- Aluminum is more resistant to corrosion and thus there would not be a need to paint the metal, as the situation would be with steel. This provides for a durable product.

- Aluminum is more ductile than steel. This characteristic is important when looking at the cowling and its manufacturing process. It would probably be made by extrusion and thus aluminum is easier to work with a more ductile material.

- Even though steel is cheaper, aluminum is also available at a reasonable cost, \$3.50 per lb

Advantages of Aluminum 6061 over other types of aluminum:

- Aluminum 6061 is the most weldable type of aluminum

- Aluminum 6061 is the most machinable type of aluminum

Choosing the Right Solution:

Studying the House of Quality and the Morphological Chart, since the most important quality we have to consider is the Production Time and Parts Quality, according to the analysis above the best the best combination of solutions is: **Strapped Back Support, Type 304 Stainless Steel, 6061 Aluminum.**

Section 3: Analyses of Solution

a) Stress Analyses

Analytical Stress Analysis

Base:

Referring to the 3D model, the device can be broken into half for the stress analysis. Calculating the weight of the sub assembly on the base using *Pro Mechanica,* and adding half the weight of the person standing (150 lb.) which gave 400 lb. = 182 Kg = M_{sub}. To check if the material chosen/ modified from before (aluminum) is strong enough to handle such a load, we check if it is going to fail in normal stress, and/ or shear stress.

Normal Stress

There are two normal stresses acting on the U piece. First, in the Z- direction (vertically perpendicular to the ground), and second, in the X- direction (parallel to the ground) due to the bending moment.

1. Given that the yield stress of Aluminum is: **225 MPa**, while the normal stress applied to the base =

$$\sigma_{\mathbf{applied}} = \frac{\mathbf{F}}{\mathbf{A}}$$

Where F = = M_{sub} X g X F.S
= 182 X 9.81 X 2
= 3.56 KN

Where g is the gravitational acceleration, F.S is the factor of safety, and A ≅ 250 in^2 = 0.161 m^2, which is *half* of the ***TOTAL*** area of the base. The total area includes the 4 U tubes (Base piece 6 – 9), the two edge supports (Base Piece 3 – 4), and the middle support (Base Piece 5) and was calculated from *Pro E 2000I*$^{\wedge 2}$.

⇒ $\sigma_{\mathbf{applied}}$ = 22.1 **KPa**

Which is considerably less than the yield stress of Aluminum.

2. Drawing the bending moment diagram, we find that the maximum bending moment is equal to half of the maximum force times half the length of the U tube all over two (2) since we have two tubes. Including a safety factor of two, we will end up with the maximum bending moment **M = 553 N.m**.. Therefore, the max stress in X- direction is:

$$\sigma_{applied} = \frac{M\,z}{I_{xx}}$$

where z is the distance from the center of mass to the surface of maximum bending moment (0.66 in ≅ 0.0168 m, and I_{xx} is the area moment of inertia in the x direction (9.67 X 10^{-7}m^4). Note that both values are calculated from ProE.

⇒ $\boldsymbol{\sigma_{applied}}$ **= 960 KPa**

Hence less than the yield strength of Aluminum. Therefore the base is in a good condition, and won't fail in normal stress.

Shear Stress

Since the base is a U shape structure, that means the shear can not be simply calculated by dividing the maximum shear force on the total cross sectional area. Although Using the maximum shear force calculated above, ⇒ the maximum shear stress (τ_{max}) is equal to

$$\tau_{max} = \frac{V_{max}\,Q}{I_{xx}\,t}$$

Where Q is the first moment of inertia (0.124 in^3 ≈ 2.03 X 10^{-6} m^3), I is the area moment of inertia, and t (0.2 in ≈ 5.08 X 10^{-3}) is the thickness of the U tube.

Using *Pro E* to calculate the Q, I_{xx},

⇒ $\boldsymbol{\tau_{max}}$ **= 1.47 MPa**

Which is less than the yield stress of Aluminum

Final Remark on base: It can with hold the weight. The only concern is that the weld between the U base pieces and the side base pieces, is going to handle this amount of shear stress. According to literature (Mechanical Design Handbook, Harold A. Rothbart), the material we are using holds well in welding (gas metal arc) and can handle up to 18 Ksi (≈ 125 MPa).

Bolts

In analyzing the bolts, most of them did not need to be analyzed since there is no significant loading on them, which means that It is safe to use 18-8 stainless steel, which is highly available. The only bolts that should be analyzed are the one connecting the actuator and the torque sensor. The amount of torque produced there is equal to 350 N.m. Calculating the shear stress for a group of bolts, first the $\mathbf{F^{1}}$ have to be calculated which is due to the shear force, and then $\mathbf{F^{11}}$ have to be calculated which is due to pure moment. Since the center for all the bolts is right on the center of the Moment, and there is no pure shear force, therefore, we only would have $\mathbf{F^{11}}$. This is equal to

$$\mathbf{F^{11}} = \frac{\mathbf{Mr_n}}{\mathbf{r_a^2 + r_b^2 +}}$$

Hence the Maximum shear stress on each bolt is equal to :

$\tau = \mathbf{F^{11} / A_{c.s}}$

$\Rightarrow \mathbf{F^{11} = M / (4r^2)}$

For the M8 Bolts

F^{11} = (350)/ (4*(.0127)) =6889.7 N

τ = **6889.7 / (π (0.004^2)) = 137.06 MPa** which is less than the yield strength of the purposed Bolt 800MPa (≈116,000 psi). The Purposed Bolts are 8.8 metric, Zinc plated, available at **Imperail (http://www.imperialinc.com/items.asp?item=0120990)**.

for the M4 Bolts,

τ = **6889.7 / (π (0.002^2) = 550 MPa** which is less than the yield strength of the purposed Bolt, 800MPa.

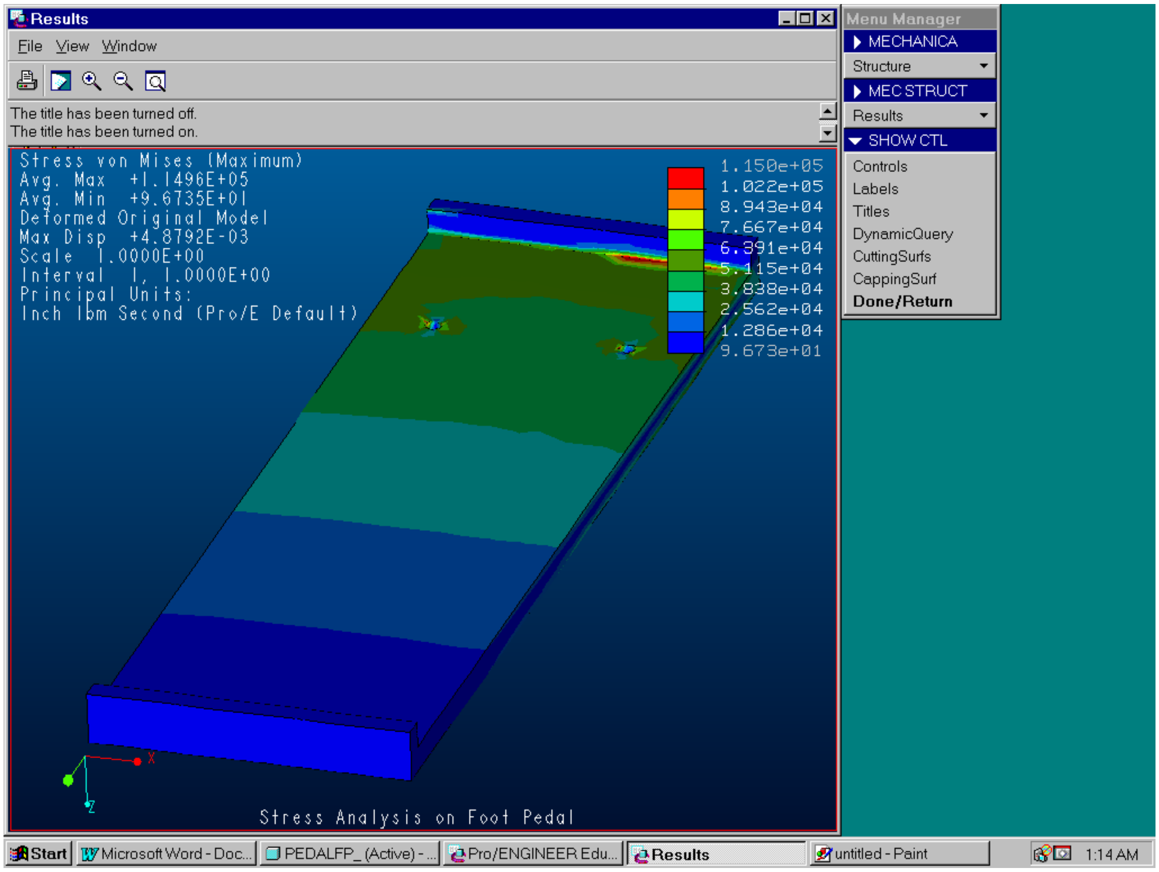
It could be noted that since the bolts the do not satisfy a Factor of Safety of 2 (instead it satisfies 1.45), it is safer to M5 bolts instead of M4. However M4 will do since the Factor of safety of 2 has been accounted for in the torque calculations (the 350 N.m figure).

Final Remark: Bolts designated can with hold the stress, but it is a bit safer to use M5 bolts instead of M4.

b) PRO-Mechanica Stress Analysis

Foot Pedal

For this analysis, we approximated the model as a cantilever with a load of 300lbs at the end of the pedal. This model would thus be an overestimate for because of the weight of 300lbs which is much higher than the average weight, and because in the real model there is a block beneath the foot pedal. Therefore, this overestimate would be considered our safety factor.



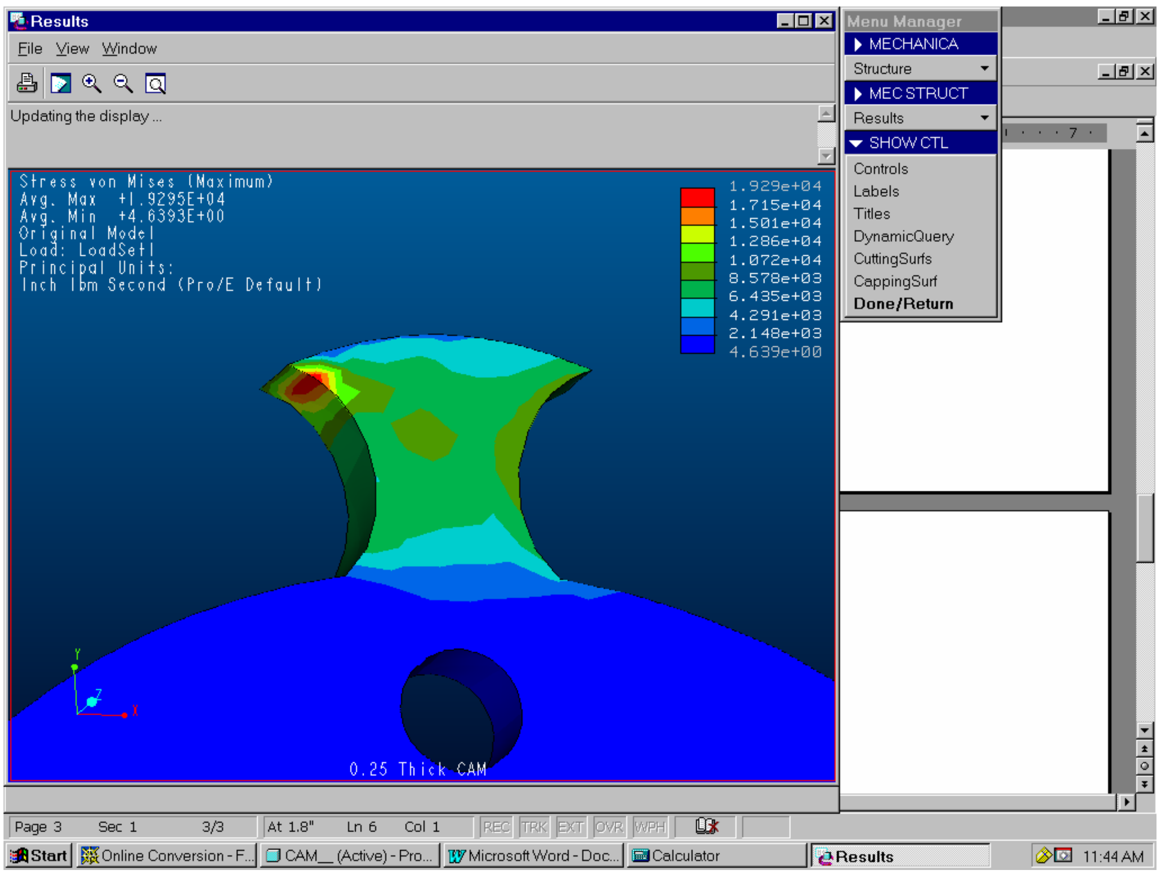
The results show that the maximum stress on the pedal is going to occur at a small region near the fixed end of the assumed cantilever. This maximum stress was found to be $1.02x10^5$ psi. The maximum deformation was found to be $4.88x10^{-3}$ inches.

The yield stress of aluminum is $40x10^3$ psi. One may believe that this device would fail, however, if one considers the addition of block underneath the pedal, the region of interest would be from the end of the foot-pedal to where the foot pedal touches the block. In this region and according to the above figure, the stresses vary from $9.67x10^1$ psi. to $3.84x10^4$ psi. Thus it is always less than the yield strength.

Thus the design achieves its requirement because the pedal does not deform by an appreciable amount and thus there would not be any considerable error in the tests that would use this device. Moreover, the stress analysis in the region of interest show a safe operating range.

CAM

Since the CAM is the final safety mechanism on the device, we had to make sure that it has a high factor of safety. Hence we decided to increase the thickness of the CAM as the original thickness of 0.125 gives a factor of safety of only 2.4.After analyzing different thickness for the cam we realized that a thickness of 0.5 is adequate as it provides a factor of safety of 4.875. Following is the results form PRO Engineer.



Location	Stress	Safety Factor
Maximum Point	19.3 ksi	1.61
Contact Point	6.4 ksi	4.875

Note: Yield Stress of Stainless Steel 304: **31.2 ksi**

The most critical region in the CAM is where the flange meets the disc, this is what we named “Contact Point”, it is very important that this has a high factor of safety because if it breaks, the foot pedal will rotate fully and possible injure the patients ankle. This region has a factor of safety of 4.875, very safe for our application. As for the weakest point on the CAM, it has a factor of safety of 1.61, this is acceptable, as it is a very tiny region as indicated and its failure will not result in the failure of the entire flange, it's only a chip on the edge.

Note the installation of this thicker CAM required milling 0.125 in of each stopper frame to create space. This was done and the device still looks aesthetically pleasing.

Section 4: Vibration Analysis

Vibration Analysis of a Plate

$$\omega_{mn} = \pi^2 \left(\frac{m^2}{a^2} + \frac{n^2}{b^2} \right) \left(\frac{D}{\rho h} \right)^{\frac{1}{2}}$$

m, n = 1, 2, 3, 4 …

where $D = \dfrac{Eh^3}{12(1-\nu^2)}$

ν = 0.34

E = 62.1 E+9

h = plate thickness = 0.1

$$D = \frac{62.1 * 10^9 (0.01)^3}{12(1-0.34^2)} = 5851.42$$

$$\omega_{11} = \pi^2 \left(\frac{1^2}{0.3^2} + \frac{1^2}{0.4^2} \right) \left(\frac{5841.42}{2.69 * 1000 * 0.01} \right)^{\frac{1}{2}}$$

$$= \pi^2 (17.36)(217.47)^{\frac{1}{2}}$$

$$= 2526.68 \text{ rad/s}$$

ω = 402.13 Hz

The lowest natural frequency of a 30 X 40cm aluminum plate of 1cm thickness is 402.13 Hz, well above our operating frequencies of 0 – 20 Hz.

It is imperative to build this device so that the patient will be as comfortable as possible. Experiments will be conducted within the range of 0-20 Hz. This means that we must verify if any of the components will have natural frequencies within this range. If so, the patient will feel the vibrations of the rotary machines. We assumed that the most important components were the eight platforms and the handrail. These were the most susceptible to be within the above range. The natural frequencies of the platforms were in the 100Hz range. The handrail gave us the most problems since we had to remodel the existing design. Aluminum hollow U-tubes were used to make the handrail. By doing this we reach 21Hz as the natural frequency of that system.

However, we have changed the design of the handrails. The handrails will be circular tubes instead. They are much lighter than the four channels discussed above.
The total mass for one circular U-shaped handrail is 1.27 kg. The effective stiffness came out to be $1/4\pi R^4$. This came out to be a total of 4767.2 N/m. So the natural frequency of the handrails came out to be approximately 40Hz. This is well above the operating range! The handrails are made out of aluminum material as well.

Section 5: Design Modifications

Upon machining and talking to different technicians at various stages during production, we encountered different issues which we have overlooked in the design stage. This is was expected and hence our timetable was scheduled to account for such issues. Following is a list of the main changes we did.

CAM & Stopper Frames

As mentioned above, the CAM was made thicker to ensure a higher factor of safety, this required milling of 0.125in of each stopper frame to allow for the new CAM. Moreover, we decided to machine the Stopper Frame by CNC, this required us to drill 4 additional holes on the frames so that we can fix the part to the CNC machine easily.

Stopper Frame Spacer

Also, spacers needed to be placed between the stopping mechanisms as to avoid inner deflection of the two components. This was an addition to the existing design because the users found that when the bolts holding the two frames together are over locked, the two frames bend and block the path of the CAM. The spacers we will use are nothing but washers that can be placed in the holes, drilled for CNC machining, purposes on the stopper frames.

Foot Pedal Spacers

The foot pedal assembly was redesigned the most. Minimizing parts was a necessity, but making sure that the assembly won't collapse under the person's weight was the most vital reason. We fabricated the following 5 blocks.

- 1 two centimeters block
- 2 one centimeter blocks
- 2 half centimeter blocks

This will ensure to cover all possibilities within the zero to five-centimeter range required to lift the pedal to the correct height. Why must we raise the pedal? We are simulating, as close as possible, the 'normal' standing of a person. So we need to align the ankle to the point of rotation located on the side brackets. The point of rotation is the center of the circle created by the 8-drilled holes on those side brackets.

Cowling and Cowling Back

Upon investigation of the Cowling and Cowling Back, it turned out that manufacturing them as designed is going to be tedious task. This is because they would require complex processes such as extrusion. So we decided to contract the machining of these parts out. The cost turned out to be very high, the best deal we got was $295 per piece for four pieces. We got this deal from Mallock Ltd. And they needed one week to deliver. This was a very expensive option, thus we finally decided to modify the parts as follows.

The parts do not carry any loads and their function is only to guide the foot and make sure it does not get caught up. That being said, the most important aspect of the parts is the shape and not the strength. Hence we decided to make each piece from two separate parts, a curved sheet metal and a curved block. The sheet metal was ordered and bent using a roller. The curved block was machined on the CNC from scrap blocks. Then the two parts were glued together using aluminum epoxy. We decided to glue the parts and not weld them because the sheet metal was too small to be welded and might alter shape. We could have screwed them on but since the parts will not be disassembled, gluing them is the easiest and fastest solution.

Sliding Mechanism

In the original design, the whole subassembly consisting of the actuator, foot pedals, bearings and stopper frames should slide on a steel plate. When the parts arrived, we found that the steel plate was too rough to enable sliding and hence the design has to be changed.

We replaced the steel plate in the original design by four smaller aluminum plates to be bolted on the base. These aluminum plates were selected to have a very smooth surface finish to enable sliding. However, we felt that due to the weight of the entire subassembly and the patient standing on the device, the load might still be too heavy to slide. We therefore decided to introduce a turning mechanism that will assist the sliding.

This mechanism is composed of to steel block, one to be attached to the base plate and the other to the base. One of these blocks would be threaded and the other will have a thru hole. Once these blocks are aligned, a threaded, steel shaft would be installed in the holes and by the means of a handle, this will be rotated and will push the subassembly promoting the sliding.

All the material for this mechanism was provided by the Machine Tool Lab. Our client greatly welcomed this new design and was impressed, as this would guarantee sliding of the subassembly that is very critical to the proper functioning of the device.

Feet Supports

The idea of installing feet to the entire device was introduced after the vibration analysis was concluded. These feet would also act as dampers and hence would minimize the possibility of vibrations. The feet where ordered and to be installed, we needed to drill holes in the square channels of the base. However, upon consultation from the machinists, we were advised to make and install feet supports and place them in the square channels. This is because, without these supports, all the load will be concentrated on the small area of the feet's nut and this high concentration of load will result in a high concentration of stress and could resulted in crushing of the square channels. Installing these supports would increase the area and hence reduce the stress. This was not predicted by the earlier stress analysis because there were no feet in the original design.
So four feet supports where installed in the base by welding, and these supports now house the feet we ordered.

Base

For the base, seeing as it holds everything together, it would be best to ensure a solid structure. The material used was Aluminum 6061. This is due to its high strength and resistance to fatigue failure. Aluminum 6061 is also more easily weldable than other aluminum alloys. Standard 3” X 3” hollow beams were purchased at the appropriate lengths. These were all welded together in the desired format. We also purchased 4” X 2” C-section beams for the base plates to be bolted on to the base. The choice of C-section was to facilitate the task of placing and removing the bolts.

Railing & Base Plates

These will also be made of Aluminum, seeing as the material is more than capable of supporting the necessary loads. Once again, standard ¾” plates were purchased, and milled for a nice finish. The actuator, safety mechanism frame, and foot pedal frame will all be attached to this plate, making it a very crucial part of the mechanism. M10 holes were drilled and tapped for these parts. M14 holes were drilled in the appropriate locations and bolts will be used to hold the plates to the C-section beams from the base. Slots were milled into one of the base plates, making its location adjustable. The addition of the moving mechanism to our design required the drilling of two extra holes we had not anticipated.

Handrail

At first a handrail was constructed by using straight tubes welded together. Upon completion, the railing was not aesthetically pleasing so we decided to redesign the handrails so that they can be easily purchased ready made. The railing was replaced by two U-tube railings, which would be simply bent at two sections. This does not affect he functioning of the device and since they are purchased ready made, they are very pleasing aesthetically.

Coupling

The coupling we needed to purchase would have to fit on the actuator shaft and be aligned with the torque sensor. There are eight through holes on the sensor. Hence, we needed to duplicate those holes on the coupling so that attachment is made possible.

The main issue when ordering these couplings was to make sure that backlash was avoided at all cost. Backlash is a type of rotation allowed by the rotational clearances between coupling parts. Some couplings contain a small amount of this clearance between hub teeth and sleeve teeth. Basically, the coupling slightly rotates on the actuator shaft.

We contacted Kinecor and spoke with one of the engineers there, Mr. Jacques Mosienko. He helped us choose an appropriate coupling that will meet our requirements. We notice on the figure containing the coupling, a slit is being machined so as to press fit the coupling on the actuator shaft. Also, also a hole is drilled so as to fix a setscrew to tighten the coupling. We believe this type of coupling will do the job well.

Section 6: Miscellaneous

Painting

We decided to paint to the platforms and the outsides of the base. This was important to cover the different scratches and marks on the metal created during machining and assembling. We had the option of anodizing the entire thing but we learnt that finding a place to anodize a base of this size would be a very difficult task. Thus we investigated the best type of paint to use to achieve the best finish. We finally decided to go for rubberized car paint for many reasons. This type of paint does not require a primer and hence saves us time, moreover this paint takes only 15 minutes to dry up. The only restriction is that we only found two colors, beige and black, originally our client told us to go for blue but finally he approved the black color.

Hydraulic Tubing and Coupling

This part of the project focuses on the installation of appropriate piping to the two Rotac actuators. We would need to arrange this set up so as to avoid having a messy layout of the various tubing we may use.

We have contacted Mr. Benoit Fortin, engineer from MCS-Servo to learn more about the installation process. With him, we will be able to make arrangements on the exact dimensions of the tubing we need to order. Further, knowing the angle of the bend that the tubes make is a crucial parameter in laying out a neat arrangement.

We have investigated the three wires connecting to the potentiometer, servo valve, and the torque transducer. The wires are strapped in a plastic ring and run along the hydraulic tubing on the existing experimental device. We would like to replicate this set up as close as possible.

The appendix provides some specifications on the actuators ordered, as well as, some the accessories such as couplings. The detailed drawings have been obtained from Textron, an American based company. There is also one page from Magnaloy Coupling Company. The steel bushed, splined bore is the coupling we ordered.

Section 7: Cost Analysis

This cost analysis of this project consists two parts:

I. Parts ordered from outside university.

II. Materials ordered from outside but were machined in the university.

III. Other Costs.

I. **Parts ordered from outside university:**

Some parts of the projects were very critical for its success. Hence therefore they had to be machined with great deal of accuracy that the machine tool lab in McGill University could not accommodate for or it would cost more. The following parts were critical for the project:

1) The two rotary actuators.
2) Hydraulic power unit.
3) Torque sensors
4) Swivel base (used as shock absorber and leveling mechanism)
5) Couplings.
6) Bearings
7) Safety Harness

Since the parts above are to be ordered from outside, then the main focus is to search for companies that will provide competitive prices and good services such as installation and maintenance. Since the Biomedical Engineering Lab (the Client) has been dealing with MCS-Servo Inc. for a long time, they were an apparent choice to order parts 1) and 2) from the list above. The final quotation for these parts was $44,100.00 and $19,100.00 for parts 1) and 2) respectively.

For Part 3) of the list, the client recommended that INTERTECHNOLOGY Inc. since the client have already purchased this part from before from the same company and have the necessary maintenance equipment for it. The final price this part was $3,950.00.

As for part 4), there were a number of companies considered. The company that offered the best price and located in Canada was VIBRASYSTEMS Inc. .The location of the company does matter for the project since there is shipping costs to be considered. The project needed 8 pieces of part 4), and the quotation on it was $24.00 each plus $3.00 shipping and handling, plus GST tax, which added up to a grand total of $235.40.

For part 5), there were two companies competing to supply the couplings. The first one EXTRON Inc. which is located in the United States of America and the other one was KINECOR which is located in Saint-Laurent, Quebec, Canada. EXTRON provided more competitive price than KINECOR, but if you add up the shipping and handling cost KINECOR comes with a cheaper price. Another factor that KINEKOR had over EXTRON, was the location. The location was important factor to consider in this case because of maintenance costs. Hence the closer the company to the client, the faster and more convenient for the client to get the service he acquires. The final price for the two couplings was $982.00.

For part 6), the criteria for selecting the company to provide the bearings was it had to be a Montreal based company. The reason for this is to make it more convenient for the client, faster delivery and easier to replace in case another bearing with a different dimension is needed. Four (4) bearings, at $47.78 each, plus tax added up to $219.84.

Finally, for part 7), the criteria for selecting the provider was it had to have a retail outlet in Montreal. This due to the fact that the harness may have to be exchanged in case the device had to be repositioned elsewhere on the lab. The only company that satisfied these requirements was SafeEX. The price of the harness was $100.00.

Hence the total amount for the first part of the cost Analysis is **$71937.24**.

II. Materials ordered

Most of the material used in this project was Aluminum 6061. Hence, It is better to find a retailer here in Montreal. The technicians in the machine tool lab suggested V-METAL.

After getting their quotes, it was understood that they will not deliver, and they will have the material by a very late date. Due to the lack of services, it was better to search a new company to the material. After faxing a number of companies, RAPIDO METAL was the only one to fax back and provided excellent service where they would deliver and had the materials required for the project. The specific order is presented in the following bill: Hence the grand total of the material ordered from RAPIDO METAL was $1,1750.00. Note that the final price given above wasn't the final, since the company have given us a discount.

RAPIDO METAL had all the great qualities of service, but it did not provide "specialty cuts". "Specialty cuts" means that it will cut a piece specifically for our dimension needs. The only company that we heard can do such a thing was SUPERMARCHE DU METAL. Hence the remaining material (Aluminum and Steel parts) was ordered from them. The total amount paid for them was $385.34. Please refer to bills that are present at the end of the Appendix.

Hence the total amount of material cost was **$12135.34.**

III. Other Costs.

This includes day to day accessories that are needed to complete the project such as paint brushes, rollers, extra Allan keys, extra Bolts, etc.

The estimated extra costs where $150.00

The Total Costs: $84122.58 CDN.

Section 8: Conclusion

This project has given us the challenge to put in practice the theory we have learned in our 4years at McGill. The expectations were high because the project had to be almost finished for the month of April.

The challenge was to take an original design proposed by an honors student and have a working device come out of it. However, many problems surfaced from this original design. No stress and vibrational analysis was made at the outset of the project. Secondly, as the months went by, many changes in the actual design and aesthetics of the project were made to facilitate the experimental task. Changes were made to the foot pedal assembly, the handrails, and the sliding mechanism. Also, we purchased vibrational absorbers to take away as much of the operating vibrations encountered by the actuator.

This project has provided us with the opportunity to familiarize ourselves with PRO-ENGINEER software. Also, we had direct hands on experience with many power tools and machines for milling and drilling. We would like to thank all the machinists those have helped us the last four months with the machining process. In particular, we'd like to thank Tony, Roy, and Danin. Also, we'd like to thank John for welding various channels and our base together.

Finally, we would like to thank our supervisor, Professor Rosaire Mongrain who has always given us words of encouragement throughout the past few months. Also, we would like to thank our client Prof. Robert Kearney for having given us valuable suggestions on how to approach the workmanship of the experimental device.

The following pages include the detailed design of our parts, and the required technical components we purchased as well.