Design and Analysis of a Pedal Box System for FSAE 2010

by

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CERTIFICATION OF APPROVAL

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Approved by,

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CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

WAN NOR MAAWA BIN WAN GHAZALI

ABSTRACT

The main purpose of this project is to design and analyze an effective and practical pedal box for the 2010 UTP Formula SAE car by improving the parameters such as pedal box size, pedal travel, pedal ratio, input force and ergonomic concept. The pedal box that used for braking and accelerating is a vital part of the vehicle because the driver directly interacts with it and this makes ergonomics considerations essential to the success of the design. The scope of study includes the material selection, fabrication and stress-strain analysis without violating Formula SAE design specification outlines rules. The calculations of the pedal ratio and pedal travel angle were done using Microsoft Excel. Once the geometry was finalized, the design was drawn using CATIA. FEA on certain key components was conducted using the functions of the same software. The dynamics and kinematics of the design was analyzed using ADAMS. In ADAMS, the design was simulated to obtain the force or load distribution of the pedals. The results verified the improvement in the final design compared to the previous design such as pedal box size reduction, optimum pedal ratio and pedal travel values, and the ideal range of input force.

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ABBREVIATION

SAE	Society of Automotive Engineers
FSAE	Formula SAE
CAD	Computer Aided Design
FEA	Finite Element Analysis
CATIA	Computer Aided Three Dimensional Interactive Application
ADAMS	Interactive Motion Simulation Software Modules
EDM	Electrical Discharge Machine

CHAPTER 1

INTRODUCTION

1.1 Background of Study

Formula SAE is a collegiate design competition that encompasses more than 300 teams around the world that compete in eight different competitions located across the globe. The purpose of the competition is to design and manufacture a small formula-style race car. The competition contains two events namely static and dynamics events. The static event consists of cost analysis, engineering design and presentation while the dynamic event consists of acceleration, skip-pad, auto cross, fuel economy and endurance [1].

Of particular importance to the design of pedal box are the cost, design and overall dynamic events. There is a total of 1000 points in the competition, of which 100 points related to the cost, 150 points to the design, and 675 points are directly related to the performance of the vehicle in the dynamic events [1]. Due to the tremendous point differentials, the focus of the design of the pedal box relies heavily on performance, then design, with cost being a minor consideration.

The performance of the pedal box fundamentally can be measured by the effectiveness of the brake pedal, throttle pedal and possibly the clutch pedal to send a signal to the system for immediate activation. While performance is the top priority, the ergonomics of the pedal box together with the driver's feel must also be taken into consideration in designing the system as the driver is the only person that controls the car in a race.

1.2 Problem Statement

The effectiveness of the pedal box needs to be taken into considerations as it plays a major role in FSAE competition. But in order to design and manufacture a good-quality pedal box for the competition, the small team of students must also deeply cogitate about the constraints that always rise the problems to the car.

While designing the pedal box, the effectiveness and efficiency of the system majorly depends on the parameters such as pedal travel, pedal ratio and force input. It is important to know that the value of the parameters that used for race application are different compared to the normal application. Base on the observation from current design, pedal ratio and force input always become the problem to the car because the driver cannot control the car efficiently if the pedal ratio is out of the ideal range and the input force is too high.

From the previous UTP FSAE car, there were some problems regarding the mass and cost budgets of the entire car. According to Schiller [2] the overall weight of the vehicle must be less than 500 lbs ($\approx$226.8 kg) to be considered remotely competitive in the competition and after the calculations, the allocation weight for the pedal box is only 5 lbs ($\approx$2.27 kg). According to 2009 FSAE Rules [1] the cost report contributes 100 points from overall points while our previous car spends much budget on the pedal box. Therefore, it is a must to design a light weight and cost-effective pedal box for 2010 FSAE car.

In section 7.1.4 from Formula SAE Rules [1], it is stated that "Brake-by-wire" systems are prohibited. Therefore, the only choice that the designers have is by using hydraulic system. We already know that hydraulic system is more effective than using cable but the problem that arises is the required space needed for housing two master cylinders. The pedal box may not extend beyond the bulkhead plane of the car. It must fit between the lower frame rails that extend from the lower suspension point to the bulkhead. With the limited space, the cylinders must be positioned properly to avoid interfering driver's foot. The limited gap between the pedals gives difficulties to the drivers with large foot to move from pedal to pedal.

The size and the position of the driver and the body frame also must be taken into account while designing the pedal box. If the study of ergonomics is neglected, it will lead several major problems to the driver and the car during racing. But in order to design a certain system ergonomically, it is important to study first about people that will interact with the system. The previous study shows that neglecting the ergonomic concept affect the safety of the drivers during the race.

1.3 Objectives

The objectives of this project are;

a) To improve the previous design of Formula SAE car pedal box in term of efficiency and performance involving the parameters such as pedal box size, pedal ratio, pedal travel, force input and ergonomics concept.
b) To design a new pedal box system for Formula SAE that future generations can use with minimal design changes.

1.4 Scope of Study

The goal of this thesis is to develop a pedal box that future generations of vehicles can use with very minimal design changes. The thesis will outline reasons for design decisions, part selection, material selection, as well as provide all the analysis to back up the decisions. Technically, the Formula SAE car must be designed to comply with the FSAE Rules. The designs that not follow the rules will be disqualified during the inspection. Therefore all the specifications and details will be based on the regulations. As for the pedal box, the design will be referred to the previous design but with some adjustments to improve its performance and efficiency.

The project involves the ergonomics study, material selection, stress-strain analysis, kinematic and dynamic analysis and fabrication process. The study will be based on research and experiments with the aid of several software, namely Microsoft Excel, CATIA, and ADAMS. Microsoft Excel was used to program the formula for the calculation involving pedal box fundamental geometries. CATIA was used for designing and modeling of the CAD and ADAMS is used to simulate the system. For keeping the track of time, the Gantt chart was developed for the both semesters; the first semester was focused on the literature review while the second semester was for project activities and results.

The elements that beyond the scopes are controller, clutch, spring and retainer mechanism and hydraulic cable linkage system.

CHAPTER 2

LITERATURE REVIEW / THEORY

2.1 Pedal Box System Fundamental

The pedal box for Formula SAE car consists of the throttle pedal and brake pedal. The system receives the driver's command by foot to control the vehicle motion. The pedal box may also contain clutch pedal. If the pedal box does not contain clutch, then the clutch must be placed on the shifter.

2.1.1 Frame

To accommodate drivers of all sizes, the pedal box must be adjustable and for the driver's comfort, the foot rest will be installed. The pedal box must fit the two lower of frame rails the run from the lower front suspension to the bulkhead. This means the pedals initial position cannot be beyond the bulkhead but when the pedals are pressed down at full travel, they can be beyond the bulkhead if this is desired. Figure 2.1 below shows the location of the pedal box in the front part of the car [3]. The rules also state that a crash structure to absorb any impact must be in front of the bulkhead. If the pedals are to extend beyond the bulkhead when in full travel, then the crash structure must be made to accommodate a pedal travelling in the middle of it. The frame needs to accommodate the full length of the pedal box plus room for adjustability. The pedal box moves along the frame rails at different locations.

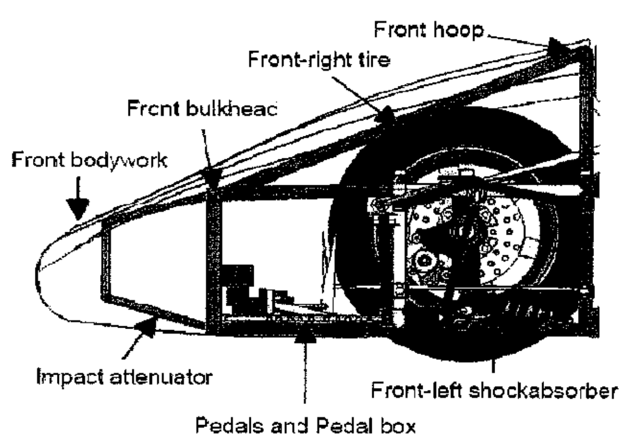


Figure 2.1: Pedal Box location at front part of the car [3]

2.1.2 Brake System

According to the rules, "drive-by-wire" is prohibited and the brake system must have two independent hydraulic circuits [1]. Therefore the pedal box must house two master cylinders, one for the front brake system and one for the rear brake system. The master cylinder has to be chosen properly because the size of the master cylinder's piston has a direct result on brake fluid pressure [4]. The pedal box also must have balance bar (bias bar) to transfer the output force from the pedal to the master cylinders. The main function of the balance bar is to divide the leg input force from the driver to the both master cylinders at the desired ratio. There are two types of hydraulic system configuration, namely front/rear hydraulic split and diagonal split. For FSAE, the first system (refer Figure 2.2 below) has been applied the most. Technically, the total force of the brake system implemented on the car cannot be simply split equally because of the different in braking methods on the front and rear wheels because both require different amount of force to totally work [5].

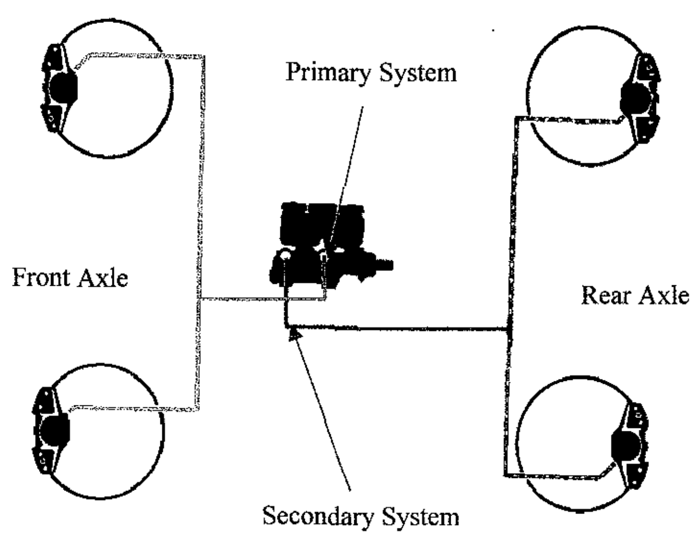


Figure 2.2: Hydraulic system split configuration for front/rear hydraulic split

The brake pedal places the largest forces on the pedal box frame. The master cylinders feed brake fluid to the brake calipers. The pressure in the system needs to reach 5620 kPa for the front and 3420 kPa for the rear in order to lock all four wheels [2]. The forces created by displacing the fluid in the master cylinder go into the pedal box frame. The pedal box frame is designed around the brake pedal because of the large force. The force applied by the human to lock the wheels should be about 445 N from driver experience [2].

2.1.3 Intake System

The accelerator interacts with the throttle body on the intake system. A throttle is the mechanism by which the flow of a fluid is managed by constriction or obstruction. An engine's power can be increased or decreased by the restriction of inlet gases but usually decreased. The term throttle has come to refer, informally and incorrectly, to any mechanism by which the power or speed of an engine is regulated. What is often termed a throttle is more correctly called a thrust lever [6].

In a petrol internal combustion engine, the throttle is a valve that directly regulates the amount of air entering the engine, indirectly controlling the fuel burned on each cycle due to the fuel-injector or carburetor maintaining a relatively constant fuel/air ratio. In a motor vehicle the control used by the driver to regulate power is sometimes called the throttle pedal or accelerator [6].

The throttle which is a basic pull cable attaches the throttle body to the accelerator. The cable must connect to the accelerator without interfering with the driver's foot. The pedal must return to its original position when a force is not applied and the cable must go back to its initial position as well. When the driver presses on the accelerator pedal, the throttle plate rotates within the throttle body, opening the throttle passage to allow more air into the intake manifold. The interactions with the clutch on the engine's transmission acts the same way with that the throttle cable works [6].

2.2 Driver Interface and Ergonomics Concept

Maximizing driver's performance under racing circumstances is one of the top aims of the vehicle design. From the outset, as required by Formula SAE rules, driver ergonomics were considered by including a 95th percentile male driver in the space frame model [1]. The measures taken to optimize driver interaction ensured that cockpit layout was appropriate and that the controls were strategically positioned for comfort and accessibility. Figure 2.3 and Table 2.1 below shows the position of the driver in the cockpit and the joint angle in a comfortable driving position.

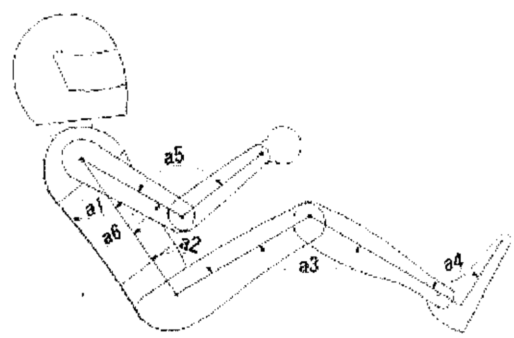


Figure 2.3: Driving position for comfort driving position [7]

Table 2.1: Joint angle for comfort driving position [7]

Joint	Angle [degree]
a 1	10 - 30
a 2	85 - 100
a 3	100 - 120
a 4	85 - 95
a 5	80 - 90
a 6	6 - 50

2.3 Design Concept

2.3.1 Pedals Shape and Design

The effectiveness of the pedal box also depends on the shape and contact surface of the pedal itself. As mentioned before, Zarizambri [8] stated that the pedal must be in curved-shape because in order to pull out the throttle cable during accelerating, the throttle pedal needs to be depress to a certain angle. The UTP FSAE second car pedal was design only using rods. Referring to Zarizambri (2008), this kind of design provides lower contact surface area for foot force (load) distribution. Therefore, this will allow the driver's foot slip below the pedal [8]. The pedal contact surface area, to which the force is applied by the foot, is considered optimal for occasional use when it has a length and width of 80 x 90mm (Eastman Kodak, 1983) [9]. Ergonomic Data (**Appendix 1**) [10] states that maximum Japanese man's foot length which is similar to Asian man is 272 mm, will be set as the maximum height of the pedal pad. This data was used to correctly position the foot pedal contact surface height. Moody [5] in his thesis designed the throttle pedal by utilise a piece of 16mm steel tube, bent to form the accelerator pedal. Refer Figure 2.4 below. The pedal pivot is a piece of 16mm steel bar with a 8 mm hole drilled through, which is seen sufficient as the joint will be rotating at a low velocity. The pedal has a slight bend in the upright section to stop the driver's foot sliding off the pedal when cornering.

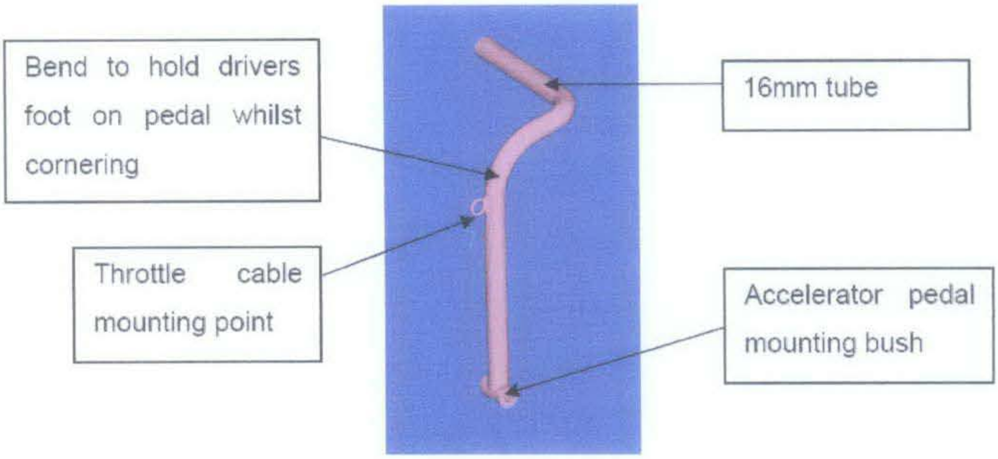


Figure 2.4: Throttle pedal by Bradley John Moody [5]

2.3.2 Clutch Pedal Status

The problem arise whether the clutch to be included in the pedal box or to be put on shifter. Most of other motor sport vehicles decided to put the clutch operation on the shifter. First we will look at the Intercontinental C Kart. In this vehicle, clutch lever is mounted on the steering shaft, which follows the same profile as the steering wheel. Clutch lever operation is performed by pulling the lever towards the steering wheel using the left hand. Champ-Car cars are at the high end of commercially available formula cars; however the cockpit layout used is the same as lower budget cars. For a Champ car, clutch operation is performed by left ankle motion on a foot pedal. Formula 1 is the ultimate of formula car competition, as the competing cars cannot be purchased, and are built by each team and respective technical partners to maximise the potential of their large annual budgets. For this reason no two team's car are exactly the same but driver cockpit layouts have been optimised over decades of racing to produce a similar solution throughout the competition. Clutch operation over recent years has moved from a foot control, to a hand control with the development of electronic control 'by-wire' systems. Currently the clutch is mounted on the back of the steering wheel and is also an electronic control.

Now we back to FSAE car. Schiller (2007) [2] in his thesis stated that the clutch must be on the shifter to reduce the complexity of the pedal box plus it will give more space for pedal and throttle pedals. Regarding to Enomoto et al [7] and their design (Refer Figure 2.5 below), by putting the clutch on the shifter will only interfere with the steering wheel handling during the race. But, in order to put it on the shifter, the system must be designed properly so that the driver will not have difficulty in handling the wheel steering.

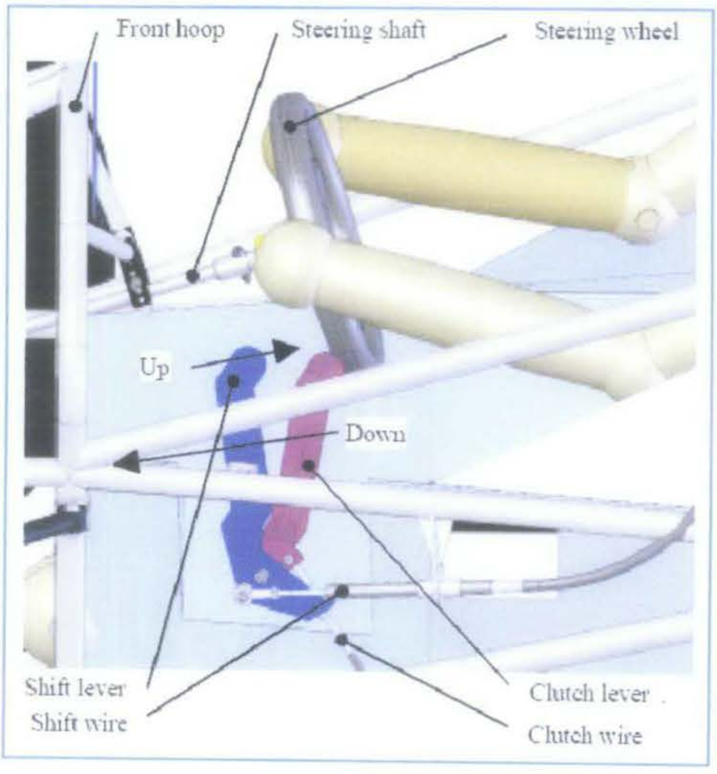


Figure 2.5: The position of the clutch at the shifter [7]

For the clutch at the shifter, a light-weight, small size and high-quality transmission should be mounted on the FSAE car and many teams use the super-sports bike built-in transmissions. For this type of transmission, the drivers should control these transmissions manually shown in Table 2.2(a). For such amateur drivers or the weekend racers defined in FSAE rules, the quick and accurate operations are not easy because large acceleration occurs in accelerating, decelerating and cornering. To make the operating system simpler for the driver, Enomoto et al (2007) [8] has developed a semi-automatic transmission operating system. Mechanical systems with links and wires were used to operate the manual transmissions. Switches mounted around the system controlled the shift/clutch with DC motors and the drivers did not release the hands for the operations. Table 2.2(b) shows the sequence of the system.

Table 2.2: Outline of the shift sequence by Enamoto et al [8]

(a) Manual sequence		(b) Full-automatic sequence	
Up	Down	Up	Down
Reading tachometer ↓ Release the accelerator ↓ Up-shift by hand ↓ Press the accelerator	Reading tachometer ↓ Disengage the clutch by hand ↓ Press the accelerator ↓ Down-shift by hand ↓ Release the accelerator ↓ Engage the clutch by hand	The accelerator kept pressed ↓ The micro computer read tachometer ↓ The engine speed is over the threshold. ↓ Decreasing the engine speed ↓ Up-shift by DC motor	The accelerator kept released ↓ The micro computer read tachometer ↓ The engine speed is below the threshold. ↓ Disengage the clutch by an actuator ↓ Increasing the engine speed ↓ Down-shift by DC motor ↓ Engage the clutch by an actuator

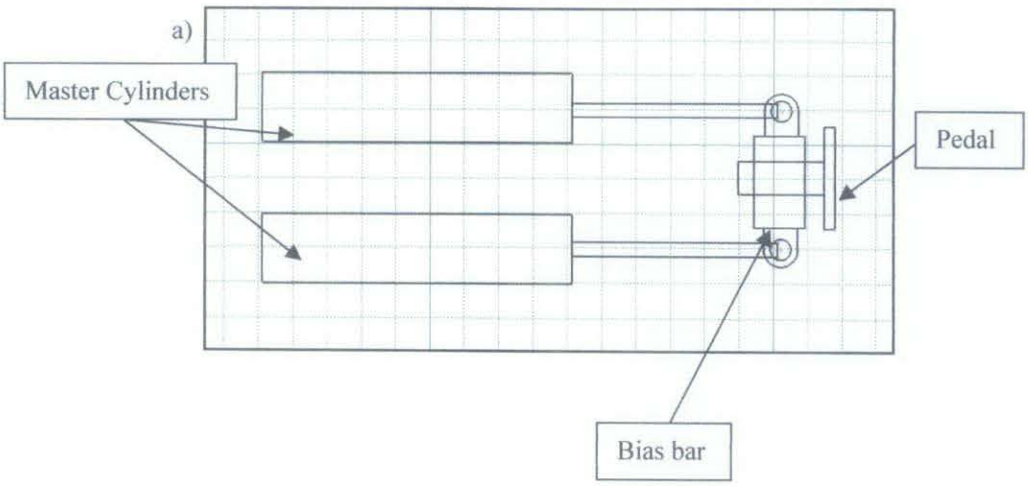
2.3.3 Adjustability Aspect

From adjustability aspect, many teams believe that it is preferable for the pedal box to be adjustable to accommodate different size of drivers. Monash Formula SAE team claimed that it was not enough to have only a two adjustment positions where the pedal box was physically unscrewed and repositioned [11]. Therefore they come out with a new solution by using slide rail. With the use of the LM76 Slide Rail as shown in Figure 2.6 below, the driver had a method of which he/she could alter the positioning of the pedal box to one of six positions with 0.7 inch increments.

Figure 2.6: Monash Formula SAE pedal box with LM76 Slide Rail [11]

2.3.4 Master Cylinder Placement

The master cylinders placement is one of the main aspect need to be taken into account. From Schiller (2007) [2], there are basically three different positions that master cylinder can take. These consist of mounting the master cylinders forward of the pedals, rearward of the pedals, or at an angle (commonly referred to as vertically mounted cylinders even though the cylinders are not mounted perfectly vertically. Refer Figure 2.7 below illustrates the location of the master cylinders in three different positions. His final design is that the angled master cylinder mounting with a hand clutch. In this design, the master cylinder mounts upside down with the spherical bearing on the back of the cylinder attached to the brake pedal and other side attached to a balance bar attached to the pedal box frame.



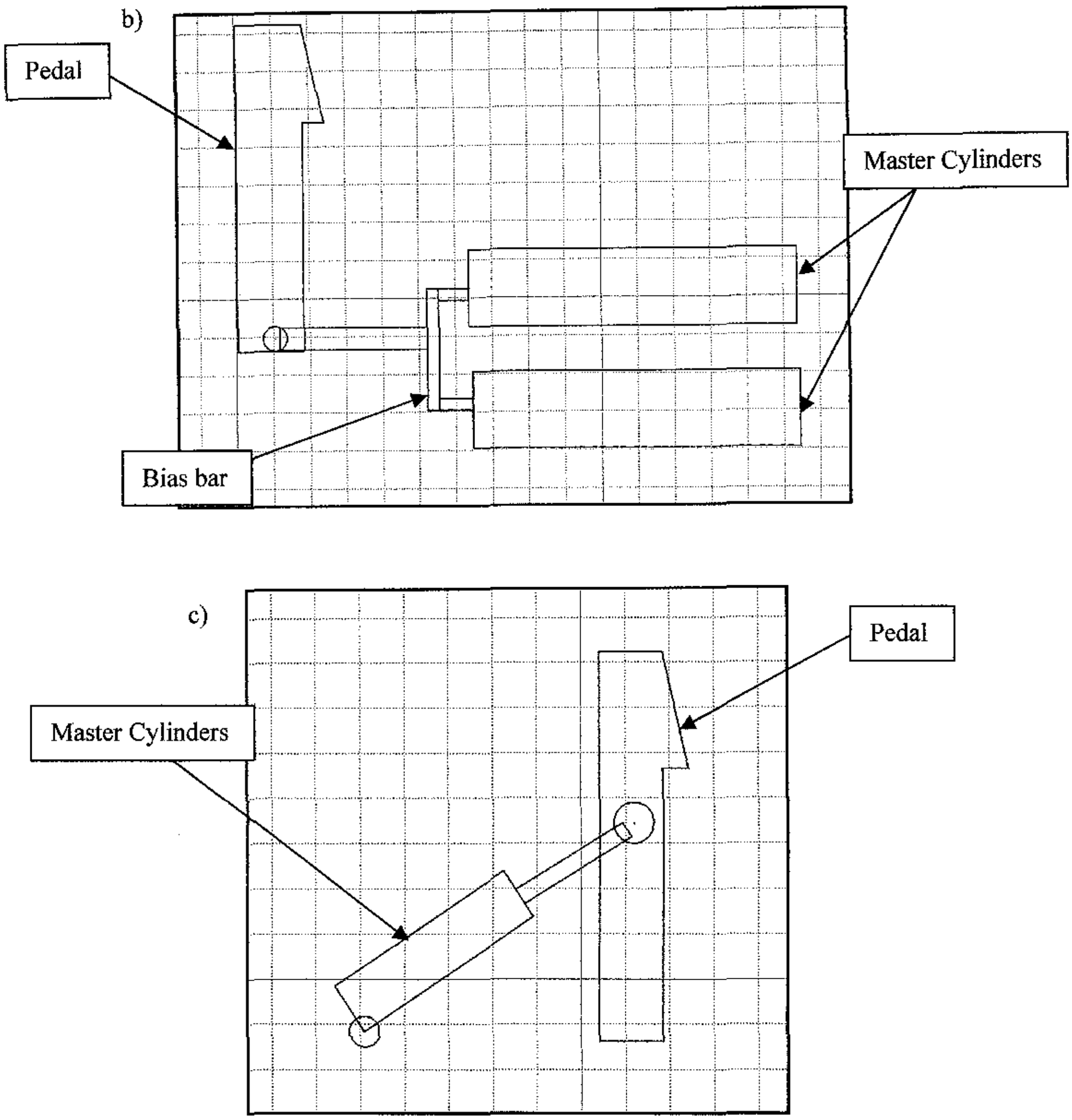


Figure 2.7: Different position of master cylinders in pedal box
a) Forward of the pedal b) Rearward of the pedal
c) Inclined to the pedal

The design has the bias bar running through the pedal with two male right hand threaded ends. The force is transferred from the pedal to the bias bar via a nylon bush and spherical plain bearing which is fixed to the bias bar, and slides within the pedal. The bias bar is connected to each master cylinder by a female right hand threaded attachment. The advantages of this design are it gives good pedal feel as deflection is low and provides less space assembly from other designs.

2.4 Parameters Involved in Pedal Box System Optimization

2.4.1 Pedal Travel

Pedal travel is the distance travel by the pedal from the initial position to the final position which can be expressed in term of angle or the length. The pedal travel is designed by taking into considerations of driver's feel. By default, the range of pedal travel is 5.0-10.0 cm following the usage type of the system. From previous design, Schiller (2007) designed it to be 5.08 cm [2].

2.4.2 Pedal Ratio

Pedal ratio can be defined as the rate of pedal travel per change in master cylinder. In operation, the brake pedal acts as a lever to increase the force the driver applies to the master cylinder. In turn, the master cylinder forces fluid to the disc brake caliper pistons or drum brake wheel cylinders. By varying the length of the pedal, and/or the distance between the pushrod mount and the pivot, the force required to energize the master cylinder can be change. Like pedal travel, the pedal ratio value also depends on the type of usage of the application. For the normal application like that used in the normal passage car, the recommended pedal ratio is 6.2:1 while for the performance and race car application the range varies from 4.0:1 to 5.5:1 [12][13].

2.4.2 Force Input

The input force required by driver's foot depends on the total force entering the pedal box frame from the master cylinders and the pedal ratio. The typical adult male can exert roughly 300 lb (≈1335 N) of force (maximum) with one leg [14]. For the race application, it is important to keep the required input force below 120 lb (≈534 N) and 80 lb (≈356 N) is the ideal most race application [12]. The average manual (non-power boosted) master cylinder requires between 600-1,000 PSI (≈4137-6895 kPa) to be totally effective [14]. Thus, 80-120 lb (≈356-534 N) of leg force has to be translated into 600-1,000 PSI (≈4137-6895 kPa). The way it is accomplished is by adjusting the pedal ratio.

CHAPTER 3

METHODOLOGY

3.1 Procedure Identification

Generally, the project can be divided into two parts. The scope of work for the first part involves doing research about background study, understanding the theory and work planning. The scope of work for the second part includes designing, simulation process, stress-strain analysis and material selection. Below are the steps taken towards the successful of the project.

a) Full understandings of the regulations and requirement of Formula SAE race car specific on driver interface system mainly on pedal box system.
b) Study on theory understanding and parameters acquisitons of the component.
c) CAD design by using CATIA
d) Analysis with focusing on material selection.
e) Study on kinematics and dynamics analysis of the system itself. The analysis consists of equations derivation and iterations and comparison between the analytical method and modeling simulation. Microsoft Excel is used first by coding the formula and putting the values. Assumptions have been made by considering all the possiblities
f) Simulate the design using ADAMS. The force distribution, pedal angle and pedal travel of the system has been ploted to obtain the results.
g) For the improvement, further research and development has been discussed and recommendations have been made in order to create the optimal result for Formula SAE competition.

The procedure identification and has been summarized in Process Flow Chart (see **Appendix 2).**

3.2 Tools and Equipments Required

The tools and equipments required for this project are;

a) CATIA

Some of the parameters involving the geometry are acquired using CATIA. Once the specifications are all set, the drawing and modeling will start take place. CATIA aided to give the 3D projection of the design to give better understanding of the system. Then, FEA analysis was conducted by using the functions in the same software.

b) MSC ADAMS

MSC.ADAMS (Advanced Dynamic Analysis of Mechanical Systems) from MSC Software is a software tool for simulation of motions in mechanical systems. In this project, ADAMS View was used to do dynamics and kinematics analysis. From this analysis, force or load distribution of the pedals can be obtained by running the simulation.

c) Microsoft Excel

Microsoft Excel in this project is used to do the coding for the formulas, calculation involving pedal box fundamental and mathematical equation.

d) Current UTP Formula SAE Pedal Box

The current UTP Formula SAE Pedal Box will be used as datum in this project. All the analysis will be referring to this model initially and improvement will be made to maximize the system performance.

The timeline and key milestones of the project can be summarized in the Gantt chart. See **Appendix 3**.

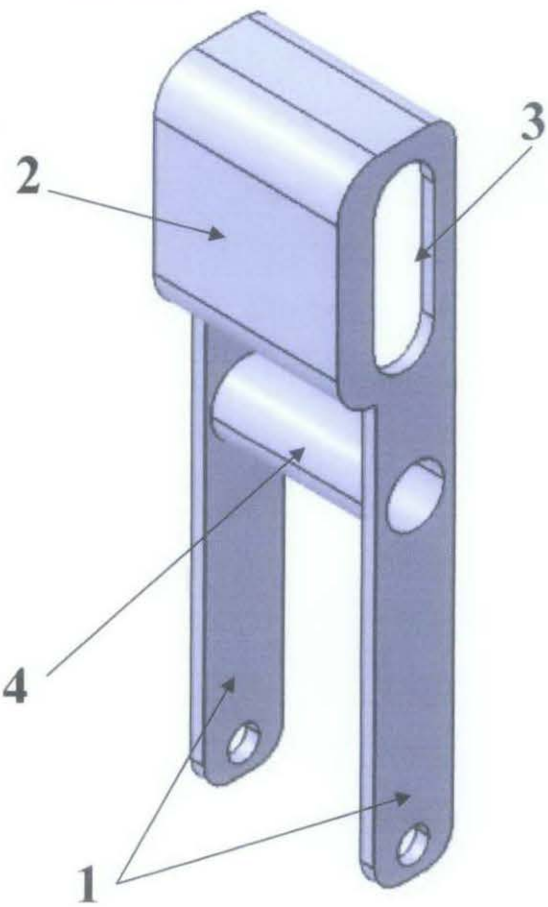
3.3 Project Activities

The project consists of two major activities i.e. designing using CATIA and simulation process by ADAMS software. The designing process took place after all parameters and specifications are obtained.

3.3.1 Designing Process

a) Brake and Throttle Pedals New Design

In the pedal box system, the brake pedal is used to activate the brake to slower and stop the car while the throttle pedal is used to open the throttle to accelerate the car. Based on this major function, the pedals must be designed properly to ensure that the driver interact with the system accordingly. Table 3.1 below shows the design of brake pedal and throttle pedal. Refer **Appendix 4** and **Appendix 5** for detail design.

Features 1. Two pedal support for lower stress concentration 2. High pedal contact surface area 3. Weight reduction at low stress concentration (refer Section 4.3.2) 4. Further support at bias bar location	 Finalized brake pedal design

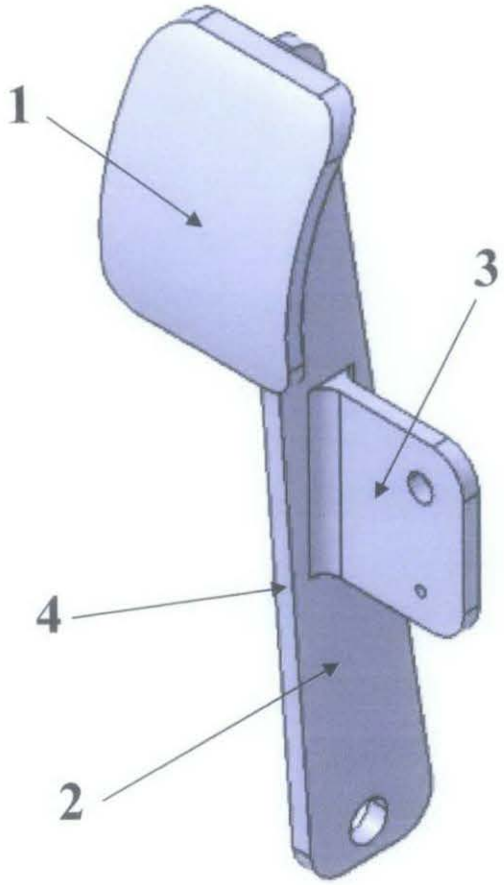
Features	
Features 1. Curve contact design for depress purpose 2. No critical stress in throttle pedal 3. Cable stopper and spring retainer hole 4. Pedal lever is designed to be inclined for better depression and ergonomic	 Finalized throttle pedal design

Table 3.1: Finalized design and features of brake pedal and throttle pedal

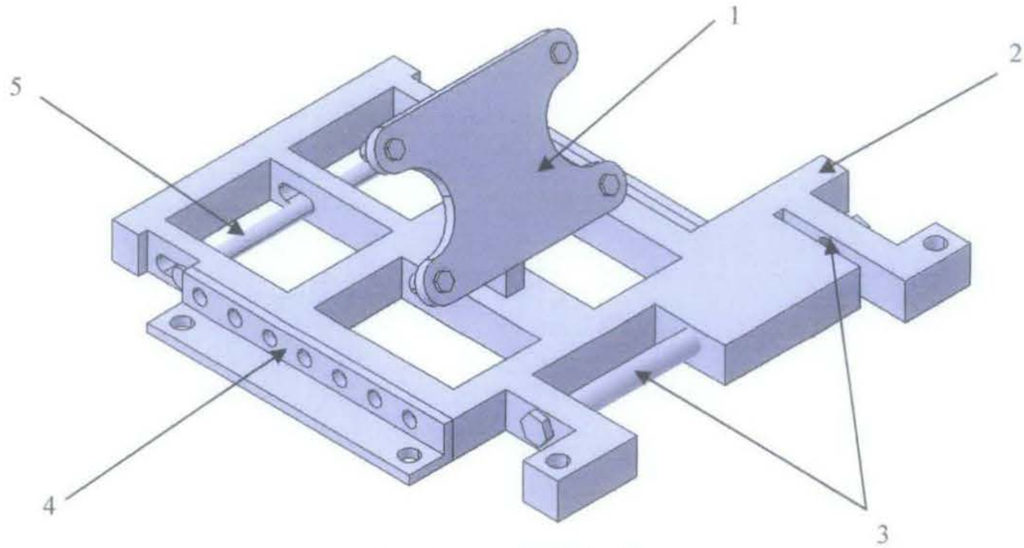
b) Clutch on Shifter

It is already discussed in Section 2.3.4 about the master cylinders location. Each of the three proceeding design can incorporate a clutch with a fairly equal degree of difficulty, so the decision of whether or not the clutch should be included in the design is independent of the proceeding master cylinder mounting decision. Removing the clutch from the pedal box and placing it on the shifter will decrease the complexity of the pedal box. The brake pedal and throttle pedal will be able to be more spaced out. Drivers with larger feet will not have to move their feet from pedal to pedal making driving easier.

The main problem with removing the clutch from the pedal box is that most drivers are used to a foot clutch. A secondary issue is that it increases the complexity of the shifter design. The solution from these problems is that the drivers must get familiarized with the condition in the cockpit first before entering the competition. If the drivers get enough time in the car, they may be able to get used to the hand clutch.

c) Pedal Box Mounting New Design

The main function of the pedal box mounting is to locate the pedals and to house two master cylinders for the brake. The design of the pedal box mounting will focus on elimination excessive material and weight reduction while maintaining its robustness. The weak pedal box mounting will only make the system failure and affect the safety of the driver. With space is crucial, the design will ensure that it fit in the specified location which is discussed before in Section 2.1.1. Besides, the design must be simple enough for ease of fabrication. This will cut off the time for manufacturing process for the whole car. To accommodate different size of the driver, the mounting will have two sliding rails for the movement. The pedal box mounting then will be screwed to the floor of the vehicle. Refer Figure 3.1 below. Refer **Appendix 6** for detail design.



1. Master cylinder holder
2. Pedal box mounting main body
3. Pivot rod
4. Slider rail
5. Master cylinder slider

Figure 3.1: Pedal box mounting finalized design

4.3.2 Simulation Process

After all the parameters are set, the both pedals will undergo simulation process using ADAMS View. In this software, each pedal will be drawn following the exact dimensions of the designed pedals especially the length from the pivot in order to obtain accurate result. Before the simulation process take place, all joints, loads and contact surfaces need to be carefully verified. Figure 3.2 below shows the simulation process in ADAMS for brake pedal.

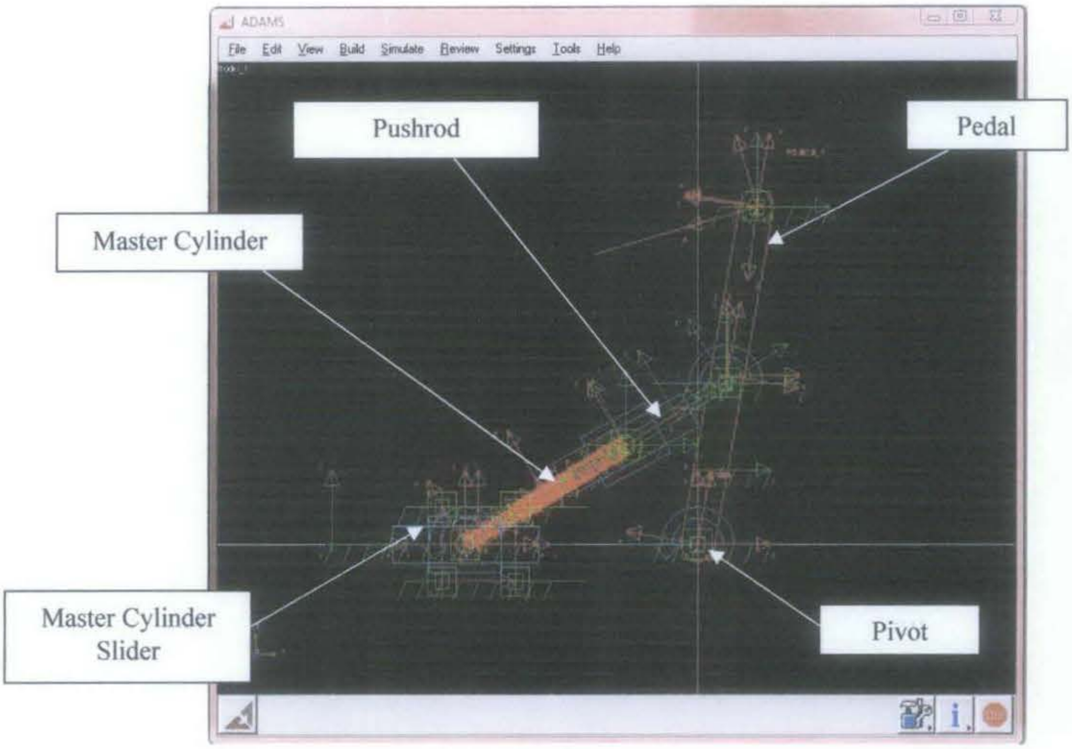


Figure 3.2: Brake pedal simulation in ADAMS

The brake pedal is inclined about 10 degrees to the right from vertical direction. This is based on the analysis to obtain the optimum result and ergonomics consideration that will be discussed in the next chapter. The function of the master cylinder slider is giving the space for the master cylinder to move when the pedal started to rotate. One end of the pushrod is fixed at the pedal at specified location (in the actual design, the pushrod will be fixed to the bias bar and the bias bar is fixed to the pedal) while the other end is letting translate into the master cylinder body. The pushrod will translate when the pedal is pressed.

Figure 3.7 below shows the simulation process in ADAMS for throttle pedal. Similar to the brake pedal, the throttle pedal is inclined about 10 degrees to the right from vertical direction. The cable stopper is located 110 mm from the pedal pivot. Then both the cable and spring is attached to the cable stopper and the pedal. The cable then will connect to the throttle while the function of the spring is to return back the throttle pedal to initial position.

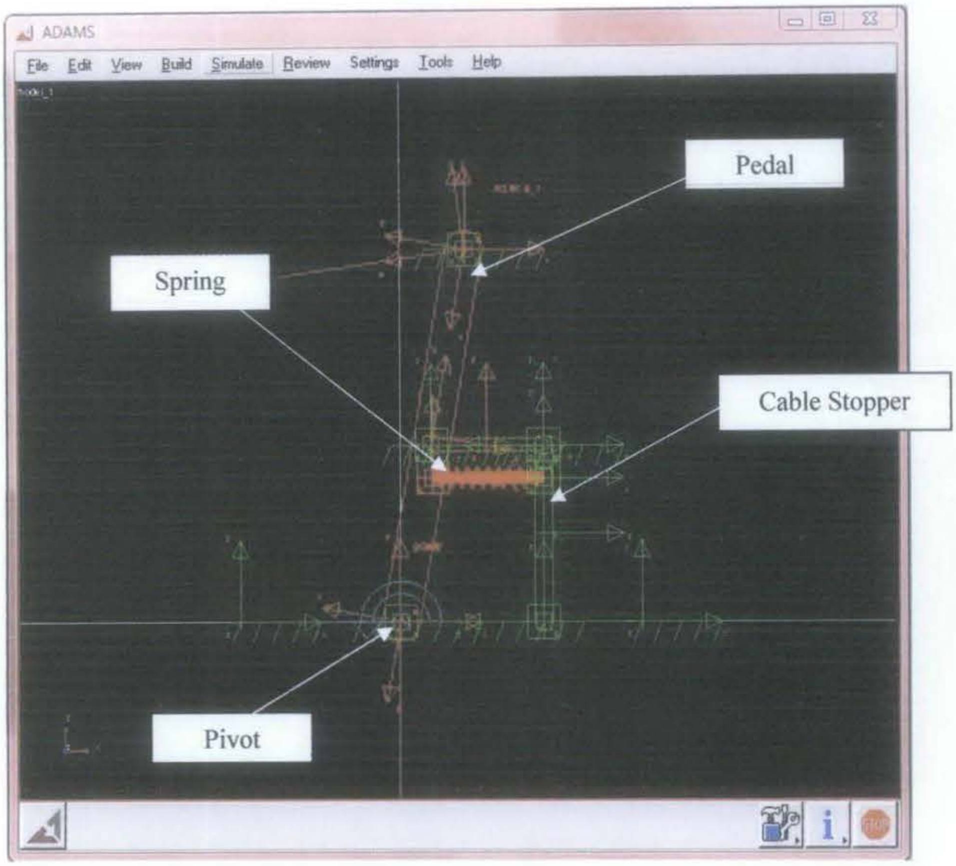


Figure 3.3: Throttle pedal simulation in ADAMS

3.3.1 Proposed Fabrication Process

Due to the time constraint, the actual fabrication process was not performed but it has been proposed in this section. The design is relatively easy to manufacture, due to the low part count. The pedal box mounting is all in the same plane making jigging is simple. The sheet metal parts for the pedal mounting is suitable to be cut on a water jet to decrease the manufacturing time and increase part quality. Then the machining of the pedal box frame members will be done by hand on mill. The holes profile on the pedal box mounting should be drilled using EDM Drilling Machine. There will be no problem to use EDM because the material that already selected for pedal box mounting is electrically conductive (refer Section 4.2).

The both levers of throttle pedal and brake pedal and the cable stopper will use the same method using water jet. The support at bias bar location will have a tube cut into desired length and bearing pockets will use lathe to fabricate. The pedal pads can be fabricating using Wire EDM. The profile first will be designing in AutoCAD because the machine only can interact with the software. The design then will be transferred to Wire EDM to cut the piece as per design. All the joints of the pedal will use TIG welding process.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Foot Pedal Operation

The operation of the foot pedal is at most ergonomically comfortable position when movement of the ankle is within 20° upwards and 30° downwards from the neutral position of the ankle, which is when the base of the foot is perpendicular to the lower leg [5]. Base on the ergonomic data, the average Asian male shoe size is nine which is the total length is 267mm [15]. The pedal length from the pivot is about 190mm. The distance of the foot from the pedal should be around 110mm which is the appropriate and comfort distance for the driver. From the diagram analysis shown in Figure 4.1 below, the drivers foot may travel 30.804° before it may slip from the throttle pedal at 'point of foot slippery' while the pedal will move until 43.247° before the foot may slip. From the diagram analysis, foot and pedal have an ample travel length to lock up all four wheels if there is no problem with the master cylinder.

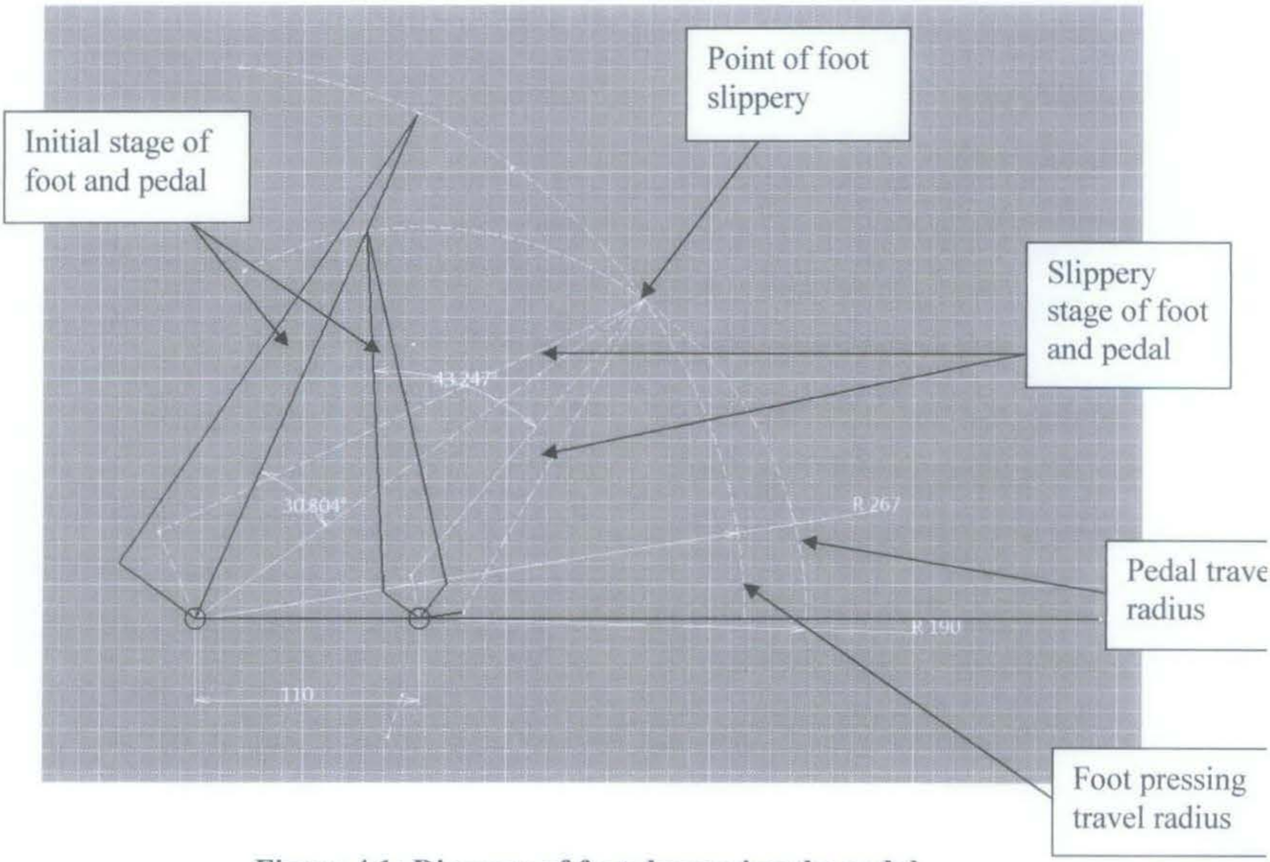


Figure 4.1: Diagram of foot depressing the pedal

The pedal contact surface area, to which the force is applied by the foot, is considered optimal for occasional use when it has a length of 80mm [5].For the height of the pedal pad, it is best to adjust it 8 inches from the floor [2].

The possible foot force applied to a foot pedal can be generated by using two different muscle functions, ankle generated force or a leg generated force. Ankle generated force gives the driver greater control over the applied force, and will be utilized for normal operation in the design of the brake pedal. The maximum force generated from ankle rotation is about 600 N, however as the pedal angle will begin at about 65° and finish at 45°, the percentage of maximum force will vary over the pedal motion from 100% at beginning to 83% at after 20° of pedal motion [16]. As this is a maximum value, the design of the brake pedal must take into account the muscle fatigue that would occur if this required for each brake pedal application. The design of the brake pedal must also take into account the possible effects of leg force being applied, under a driver panic situation. The maximum force possible when the entire leg is used is at least 2100 N; however this requires the correct seating and pedal position. For the angle of knee bend and thigh angle that the driver will have when in the Formula SAE-A race car, the force from a brake application using the entire leg will approximately be 38% of the maximum possible [17].This results in the pedal being design to with stand infinite cyclic loading of eighty lbs (≈356 N) which is the ideal leg force requirement for race application [12]. This means that the distributed force is equal to 4.45 N/mm. For the worst condition, which is the leg force being applied, an amount of 800 N force will be taken into consideration.

To correctly design a brake pedal, the basic moment calculations are used to determine the increase in the force applied by driver's foot to a magnitude required for effective brake system efficiency. The specifications given by the brake system designer, approximated that a total force of 600 N would sufficiently operate the car's brake system. This force will be achieved through mechanical advantage, placing the point to which the total force is applied (point B) closer to the foot pad (C), than the mean distance of the pedal pivot (point A). To calculate the required distance from the pedal pivot to achieve the required total force, the moment is taken about point A see Figure 4.2 and Equation 4.1.

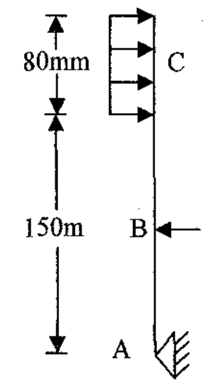


Figure 4.2: Free body diagram of brake pedal forces

Taking the Moment about A +ve

$\sum M_A = 0$

$= F_c \times l_c - F_b \times l_b$

$= (40\text{mm} \times 4.45\text{N/mm})(150\text{mm} + 40\text{mm}) - (600\text{N} \times l_b)$

$l_b = 65860/600$

$l_b = 109.8\text{mm}$

Equation 4.1: Moment calculation to find the required height of pushrod force applied

Base on previous section, it is already discussed that there are three types of placement that master cylinders can take. The result shows that the pushrod force should be applied at the middle of the pedal. This leaves no choice except to put the master cylinders at an angle. This design is made possible through the relatively recent advent of spherical bearing mounted cylinders.

There are several challenges with the design. First of all, the geometry is much more complex than in either of the two designs. In order to calculate the pedal ratio, the change in master cylinder length and pedal angle will have to be determined through calculations. Then, in order for the pushrod to be pressed normally, a slider must be installed at the end of the master cylinders and slots at the mounting for the slider. The design also contains the typical moving frame design that allows the driver with different size to move the pedal box to a comfortable driving position.

4.2 Material Considerations

The main objective of the material selection process is to find the material that can provide the lightest weight yet will not break when an amount of stress have been applied. The two types of material that have high potential for the design are steel and aluminium. These two materials are used extensively in general fabrication as they both have a reasonably low cost and good workability.

Steel would be a suitable material to manufacture a brake pedal, as it has good strength and good fatigue properties; however for application in the Formula SAE-A race car, weight considerations make is undesirable for use, as a lower strength light weight material would be sufficient. Aluminium has a lower strength than steel; however it still has good properties for implementation in a brake pedal. The major benefit of aluminium over steel is the reduced weight as it is aluminium has a density of 2.8 Mg/m3, compared to steel with a density of 7.7 Mg/m3, making aluminium 64% lighter than steel [18]. It has been decided that the part will use 6061-T6 aluminium, to create a low weight product and it has the following properties shown in Table 4.1. The endurance limit will be used in the analysis of the aluminium brake pedal as a fatigue failure could possible occur in this part. The part will be design for infinite life, and the Fatigue strength for this situation was calculated using Equation 4.2 [19] to find a value of 88 MPa. Also included into the design of this part is a factor of safety of 4, which will be used for the design of key braking parts.

$$S_n = S'_n C_L C_G C_S$$

For Aluminium

$$S = 110 \times 1 \times 1 \times 0.8$$
$$= 88 \text{ Mpa}$$

Equation 4.2: Fatigue Strength [19]

The material selected for the bias bar is a high carbon steel 4340 (properties shown in Table 4.1), as it is considered a high stress area and failure of this part would result in a total system failure.

Table 4.1: Material Properties for materials to be used in brake pedal design [19

Material	Aluminium 6061-T6	High Carbon Steel 4340
Yield Stress (σ_{yield})	275 Mpa	1020 Mpa
Stress for Infinite Life (S'_n)	110 Mpa	510 Mpa
Young's Modulus (E)	27 Gpa	207 Mpa
Poisson's Ratio (v)	0.32	0.30
Density (ρ)	2.8 Mg/m3	7.7 Mg/m3
Fatigue Stress (S_n)	88 Mpa	408 Mpa

4.3 Finite Element Analysis of Brake Pedal

An initial model of the braking pedal was created using CATIA software to create a piece of flat plate with corresponding holes and foot pad angles as previously determined. The model was considerably de-featured, with only critical elements of the component remaining. The model was constrained in all degrees of freedom on the bottom mounting hole, and the pedal pivot hole was constrained in x-direction to simulate the reactive force applied at this point by brake master cylinders. Pivot joint and clamp restraint is set at the turning hole. Force was applied to the model by converting it to a pressure applied to the front face of the pedal where the pedal foot pad would usually be placed, refer Figure 4.3.

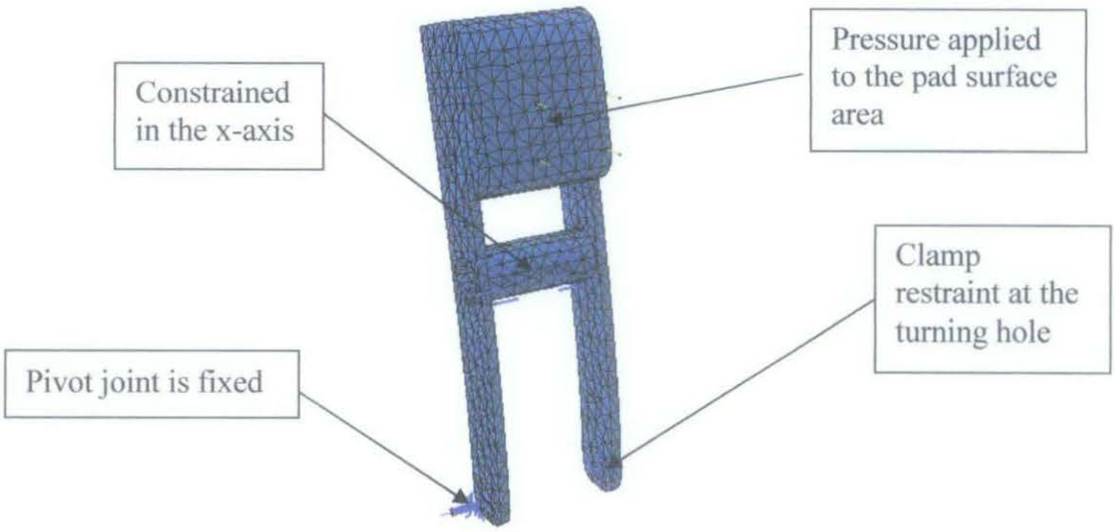


Figure 4.3: Constraints and force applied to the brake pedal model in finite element analysis

The range of the input force required from the driver's leg is from 40 to 120 pounds (178N to 534N). Typically on a street car effort is at or below 178N. In high performance vehicle and race car application, the ideal input force from the leg is 80lbs (356N) and 534N can be considered as the maximum amount of input force [9]. There are two types of analysis has been done in this section which is Translational Displacement analysis and Pedal Stress analysis and in both analysis, the load of 356N and 534N (maximum) will be applied onto the pad surface area of the model.

4.3.1 Translational Displacement Analysis

The initial brake pedal design is tested with maximum and normal load for translational analysis. Refer Figure 4.4 for the results obtained from the analysis.

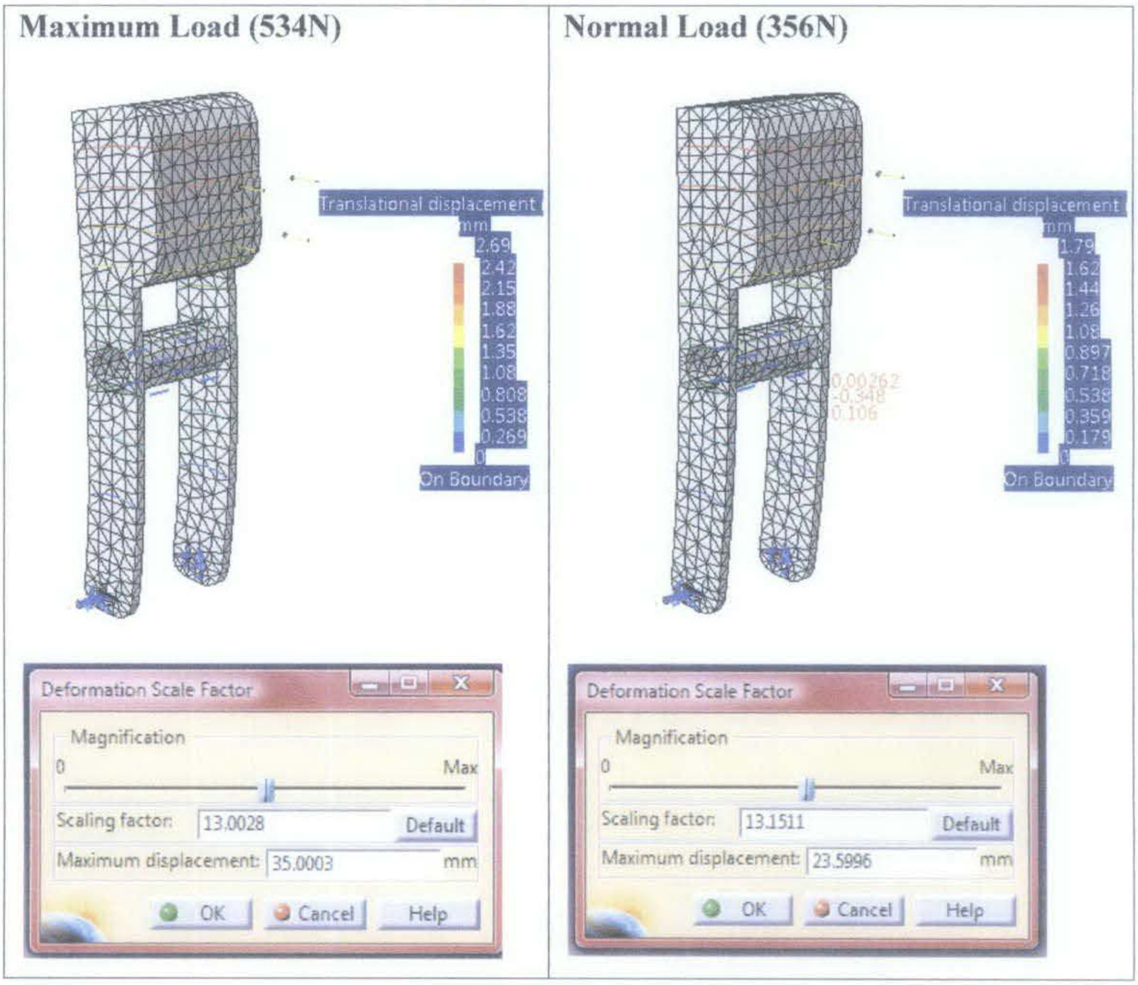


Figure 4.4: Translational Displacement analysis

For the maximum load (534N), the maximum displacement is 35.0003mm with scaling factor of 13.0028, resulting the translational displacement of 2.692mm. For the normal load (356N) applied, the maximum displacement and scaling factor are 23.5996mm and 13.1511. This produced the translational displacement of 1.7944mm. This small values of displacement of 3.00mm is acceptable because in thickness of 0.250 inch (6.35 mm) or less, it has elongation of 8% while the elongation at break is at 12% [20][21]. Furthermore, the pedal will be pivoted at the turning point and it will move in the same direction of the displacement. Thus the displacement will be neglected.

4.3.2 Pedal Stress Analysis

In this section, a certain amount of load will be applied to the pedal in FEA and Von Mises stress parameters will be used to determine whether the model can sustain the amount of loads or the model will break. Figure 4.5 and Figure 4.6 shows the results when maximum load and normal load are applied respectively. The material is set to be Aluminium 6061-T6.

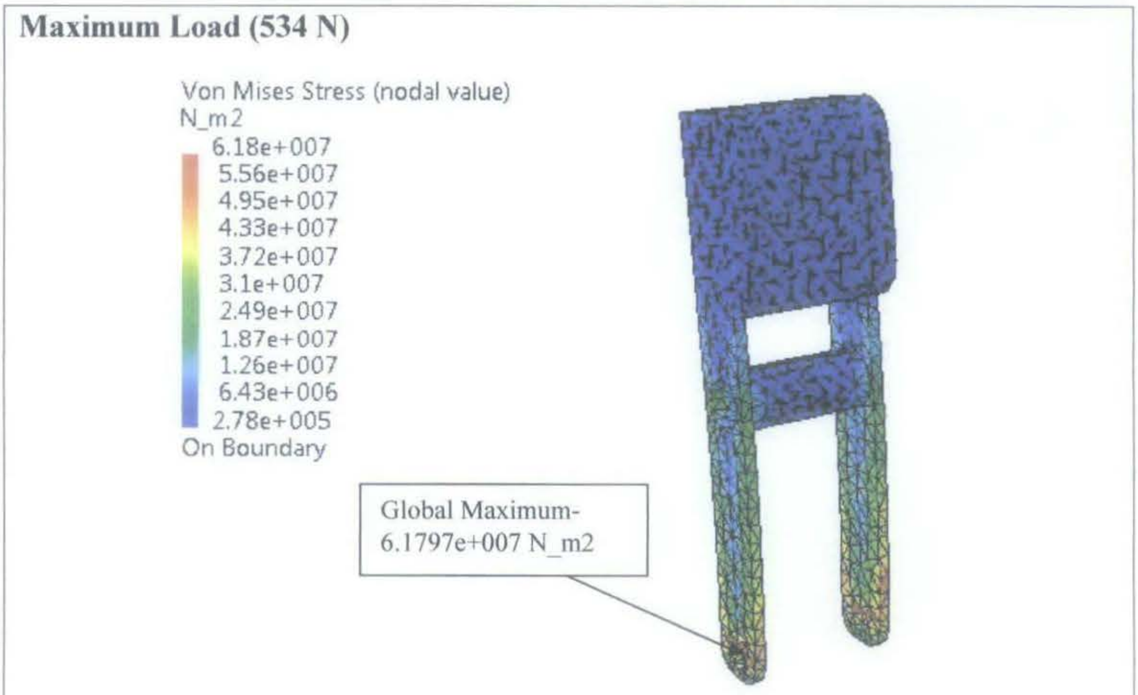


Figure 4.5: Pedal stress analysis on initial pedal design for maximum load (534 N)

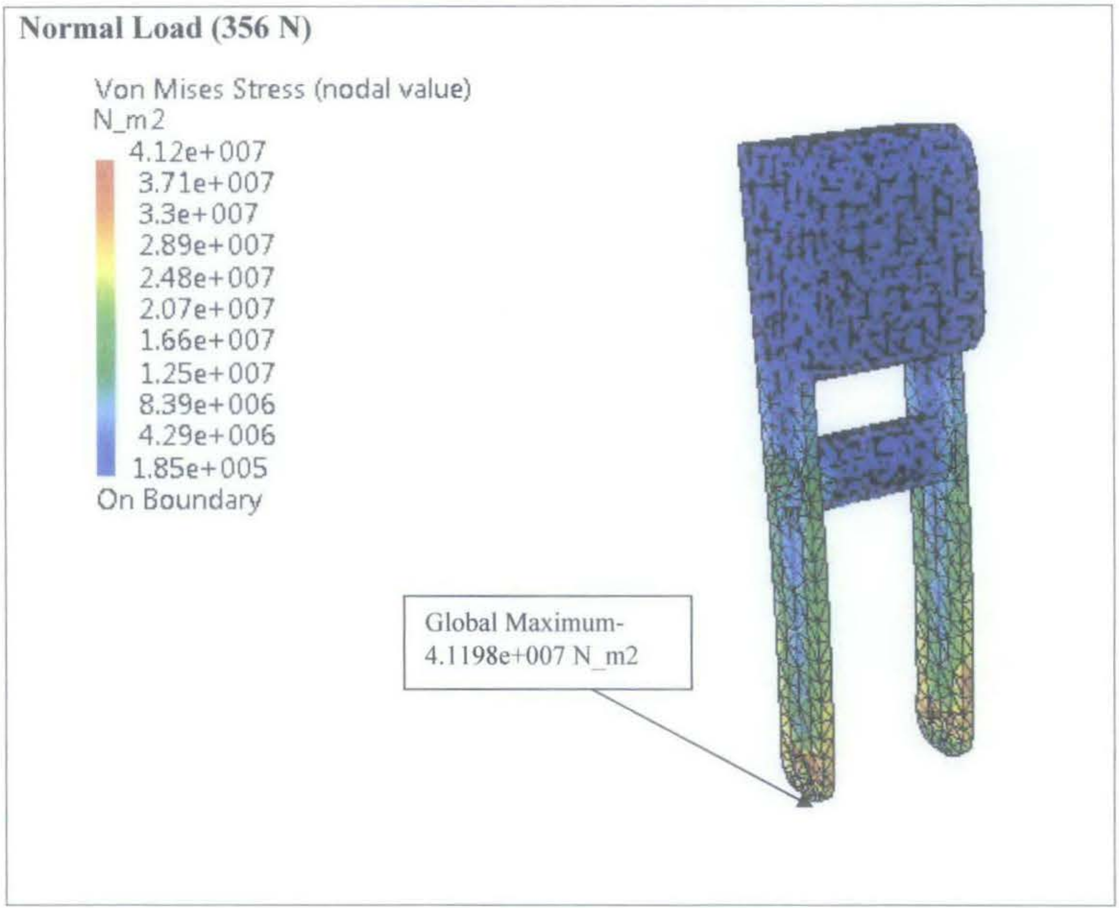


Figure 4.6: Pedal stress analysis on initial pedal design for normal load (356 N)

The global maximum stress for maximum load (543 N) is 61.8 MPa and for the normal load (356N) is 41.2 MPa. From the **Appendix** 7, the Tensile Yield Strength value for Aluminium 6061-T6 is 275 Mpa. This value is way too high compared to the maximum stress for both conditions. Therefore, it is concluded that the brake pedal design is safe although worst condition load is applied.

From the analysis result, it is found that most of the location of this initial design was under little or no stress, especially at the upper part of the pedal which is farther than clamped turning hole. This was expected as it complies with the stress distribution for beams in bending, which states that the outer most fibres from the neutral axis of a material will under goes the maximum stress when placed in bending. Due to this case, the pedal need to undergo a little changes in order to obtain the utmost efficient design.The revised design is that hole has been made at the both side of the plate at surface pad. The benefit of this modification is the associated weight reduction with the removal of materials at low stress area. Refer Figure 4.7 below.

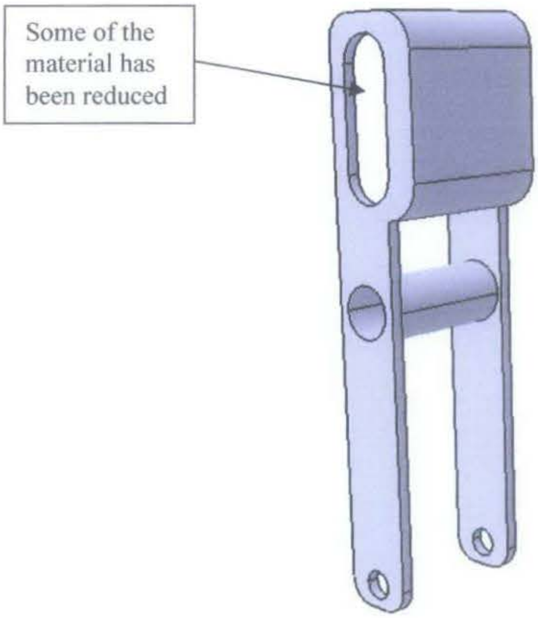


Figure 4.7: Redesigned brake pedal upright with reduced weight

The modified part was then constrained using the same method and analysed. The result was a considerably more even stress distribution and the global stress maximum is lower when load is applied. Refer Figure 4.8.

Figure 4.8: Redesigned brake pedal after FEA analysis showing the reduction in global maximum stress

4.4 Final Design

In every system proposed, it is required to study the ergonomics concept first. Likewise, this new proposed pedal box design has been referred to ergonomics data involving the driver's posture, restraint, accessibility and comfort to avoid any severe strain and injury to the driver. Besides, the new design also complies with the FSAE 2009 Rules and Regulations. Finalized pedal box design that will be equipping the FSAE 2010 car is illustrated in Figure 4.9 below. All the components such as pedals, pedal box mounting and master cylinders were assembled together. Screws and bolts were used in the assembly process. The design is made prior the study from previous design for optimized analysis result. Refer **Appendix 8** for detail design.

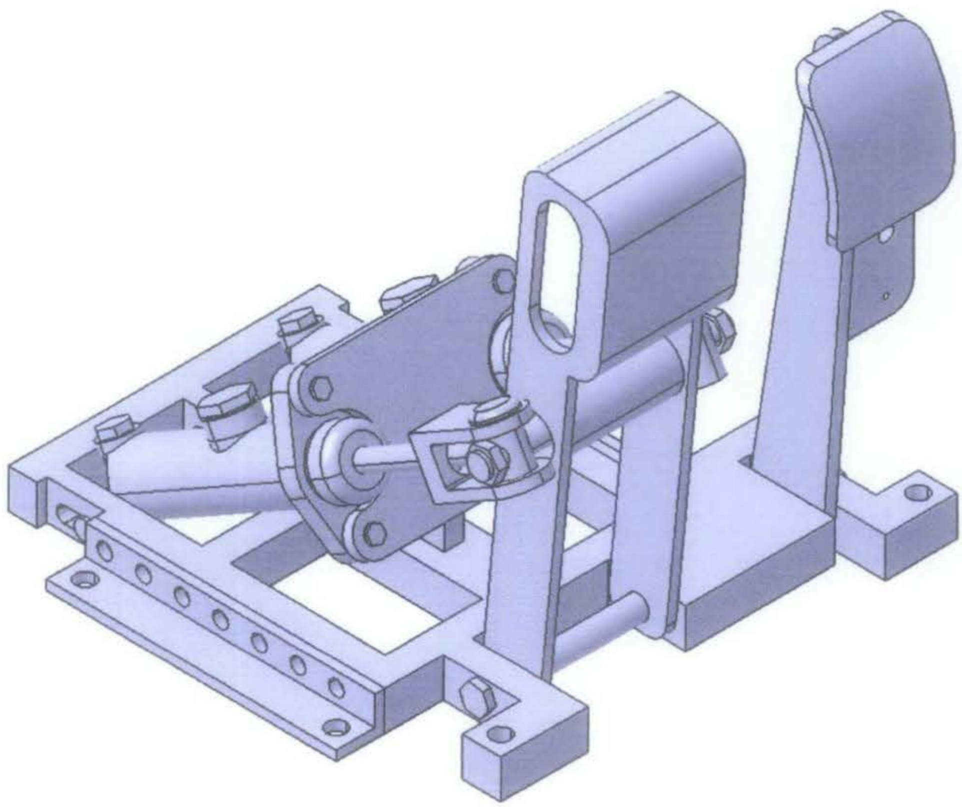


Figure 4.9: Finalized pedal box design

The detail design in **Appendix 8** shows the dimension of the finalized pedal box. The final dimension is 255 x 260 x 246 mm. From previous design by Zarizambri (2008) the dimension is 309 x 320 x 210 mm [8]. The new design gives the size reduction about 33 percents from the previous design based on the length and width of the pedal box. The major contribution to this result is because the clutch pedal was put at the shifter. This will reduces the complexity of the pedal box and the throttle pedal and brake pedal will be more spaced out. The drivers with bigger foot will have no problem in depressing the pedals. The smaller size in pedal box also gives more option to the chassis designer to locate the pedal box in the front bodywork as the pedal box location must not beyond the front bulkhead of the car. But the increasing in the height is because of the pedal length. Base on the ergonomic data, the average Asian male shoe size is nine which is the total length is 267 mm. Therefore the pedal pad's midpoint to the pedal pivot must be around 190 mm in length to comply with this data.

4.5 Brake Pedal Geometry

The first step to the design was to determine how the master cylinders and the brake pedal should interact. The two most important aspects of the brake pedal are pedal ratio and pedal travel. The pedal pad's midpoint must be about 8 inches in length as it is the average length of the male foot from heel to ball [2]. Figure 4.10 below shows the geometry of the brake pedal.

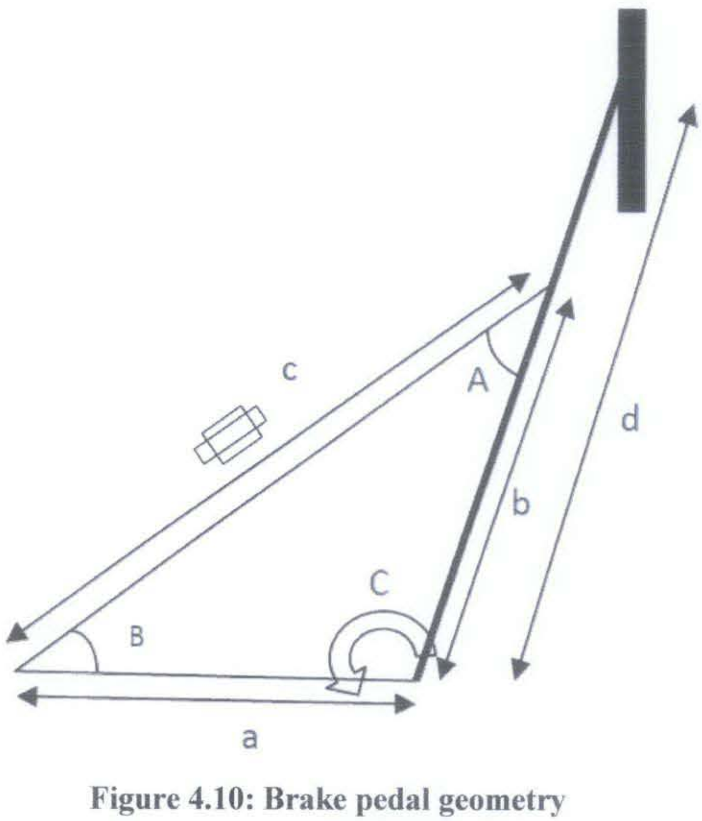


Figure 4.10: Brake pedal geometry

It is important to define all of the lengths and what distances can change before the equations governing the system are formulated. The distance from the pivot to the pushrod, ***b***, is a fixed length that has been discussed in Section 4.1. The master cylinder length, ***c***, goes from the pushrod location of the brake pedal where the spherical bearings are mounted, to the balance bar located near angle ***B***. This length changed as the brake pedal angle, ***C***, changes. It can also be increased or decreased by changing the location on the push rod thread of the female rod end that attaches the push rod to balance bar. The length between the brake pedal pivot and the balance bar, ***a***, does not change in the system when the car is in use. ***d*** is the length of the brake pedal.

The goal of the geometry analysis is to give the driver a choice of pedal ratios in the range of 4:1 to 5.5:1 which desirable for high performance and race car application. The travel should be ranged from 3 to 8 cm following that same pedal ratio pattern. To determine the pedal travel and the pedal ratio for the system, initial values for the length of the brake pedal, the distance from the pivot to the pushrod, the distance between the pedal pivot and balance bar, and the initial brake pedal angle were set as input. The law of cosines determines the master cylinder length by

$$c = \sqrt{a^2 + b^2 - 2ab \cdot \cos(C)} \qquad \textbf{(Equation 4.3)}$$

Equation 4.3 calculates the master cylinder initial and final length as the driver causes the initial pedal angle to change. The pedal travel is equal to the arc length times the pedal length using

$$PedalTravel = (C_{initial} - C_{final}) \cdot d \qquad \textbf{(Equation 4.4)}$$

with the initial and final pedal angles in radians. The pedal ratio at each given point along the path of the brake pedal equals the pedal travel divided by the change in master cylinder length,

$$PedalRatio = \frac{PedalTravel}{\Delta c} \qquad \textbf{(Equation 4.5)}$$

The next step is to determine where the pedal stops which is the point at which all four wheels lock up and the input force required by the driver at the point. The force from a master cylinder that enters the pedal box is

$$F = \Delta c \cdot A \cdot P \qquad \textbf{(Equation 4.6)}$$

where A is the cross-sectional area of the master cylinder and P is the pressure in the master cylinder at lock up. The total force entering the pedal box frame from the master cylinders is

$$F_{total} = F_{front} + F_{rear} \qquad \textbf{(Equation 4.7)}$$

where both unit front and rear system must be taken into consideration. The input force required by the driver's foot is

$$F_{input} = \frac{F_{total}}{PedalRatio} \qquad \textbf{(Equation 4.8)}$$

The calculations to determine the optimal pedal box geometry were done in Microsoft Excel. Initial values for the length of the brake pedal, the distance from the pivot to the pushrod, the distance between the pedal pivot and balance bar, and the initial brake pedal angle were inserted. The function simulates the pedal moving through an arc and computes pedal ratio and input force required.

Figure 4.11 shows a wide range of pedal angles from the start at 90 degrees all the way to zero degrees. Because the use of the law of cosines, the master cylinder length varies with pedal travel according to a cosine wave.

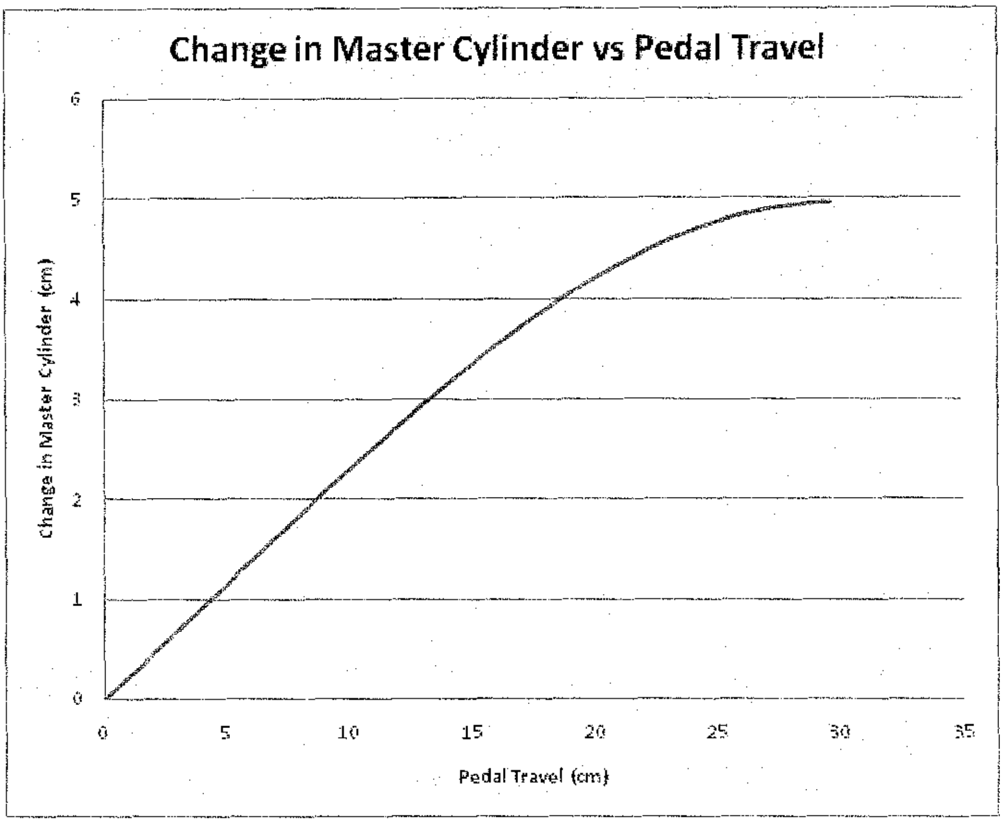


Figure 4.11: Change in Master Cylinder vs Pedal Travel for 90 to 0 degrees

4.6 Kinematics and Dynamics Analysis of Brake Pedal

By choosing an initial pedal angle and other geometry, one can choose where on the curve to operate. Only a slight increase in pedal ratio as the driver presses down on the pedal can be beneficial to driver feel, so working region that is relatively linear is beneficial. An initial angle of 20 degrees forward of vertical, a pedal length of 20 cm (≈8 inches) and a distance of 10.98 cm from the pedal pivot to the pushrod location yields Figure 4.12 below. Figure 4.12 shows the region of the graph that the pedal box works in.

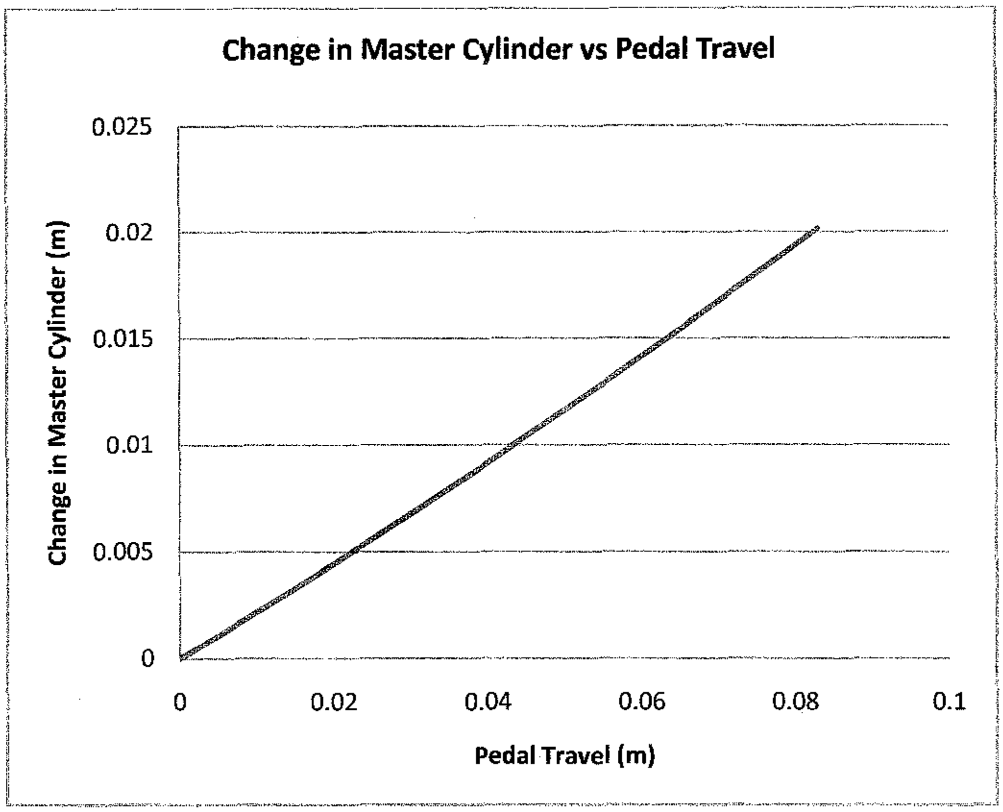


Figure 4.12: Change in Master Cylinder vs Pedal Travel

The required pushrod travel in the master cylinders is about 2 cm to lock all the tires during braking. From the graph, the pedal will travel 8 cm at most to achieve this condition. Literally the objective of pedal travel analysis is to ensure that the movement of the ankle is within 20 upwards and 30 downwards from neutral position of the ankle, which is when the base of the foot is perpendicular to the lower leg. This condition is considered as ergonomically comfortable for the driver [5]. This means that a long pedal travel is not feasible. The reason because this will give some delay for the pedal box system to send the signal to the brake to activate which is not helpful during critical situation. Even though a long pedal travels is not practical, a very short pedal travels also not desirable because driver's feel also must be taken into consideration.

Simulation has been done in ADAMS to obtain the exact angle that the pedal travels. Figure 4.13 and Figure 4.14 below shows the results from the simulation.

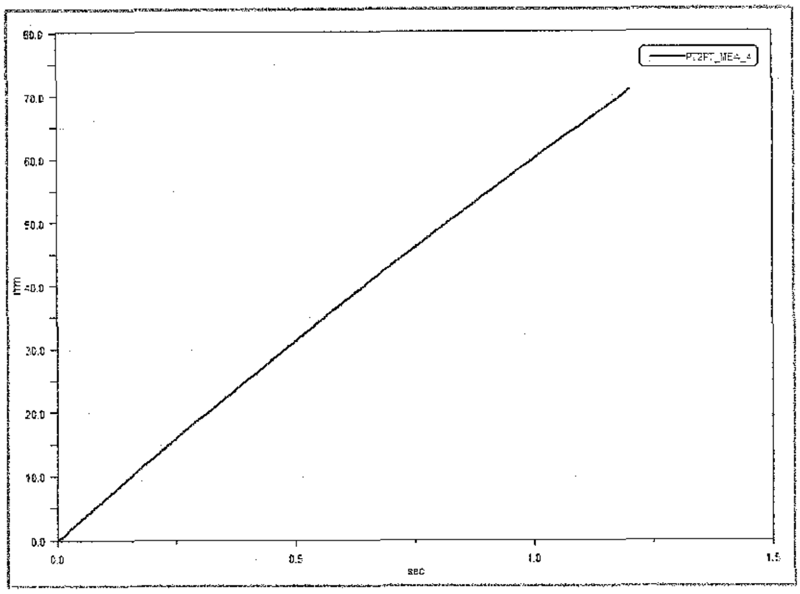


Figure 4.13: Pedal Travel vs Reaction Time

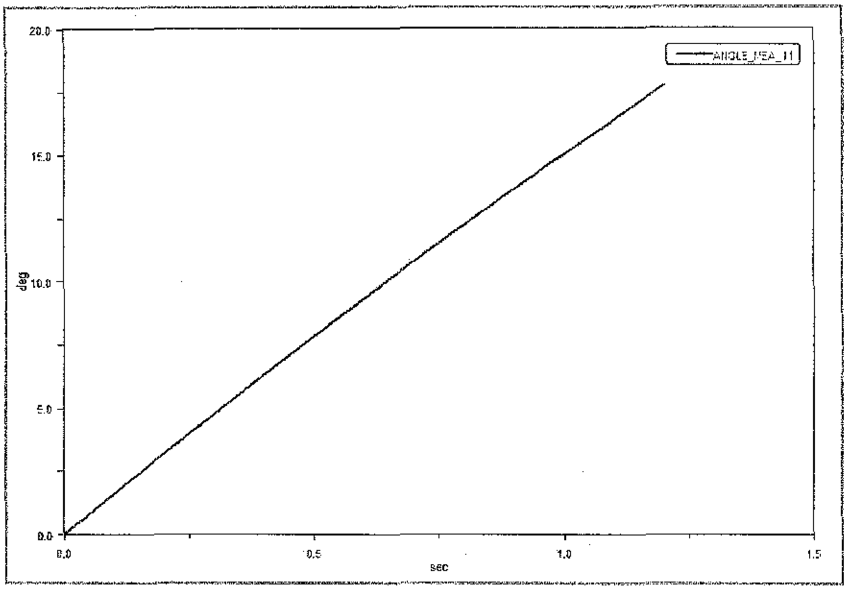


Figure 4.14: Pedal Travel Angle vs Reaction Time

The graphs above show the relations between pedal travel and pedal travel angle. When the pedal travels at 7 cm, the pedal travel angle is less than 20 degrees. This result follows the ergonomic requirement that state that the movement of the ankle is within 30 downwards from normal position. This value also satisfied the requirement from Section 4.1. The constraint provides the maximum of 43 degrees for pedal travel angle before it reaches the point of slippery.

Figure 4.15 shows the Pedal Ratio vs Pedal Travel graph. The graph shows the range of the pedal ratio when pedal travel is changing. It indicates that the pedal ratio is below 4.7:1 and above 4.1:1 as it sweeps out its arc which is in line for the desired value. Schiller (2007) from his design claimed that the pedal ratio varied from 4.3:1 to 4.6:1 [2].The greater range of pedal ratio compared to the previous design will gives more variation in operation for drivers. The region is relatively linear with only a slight decrease in pedal ratio as the driver moves the pedal through different angles. This range of pedal ratio is the ideal range for most high performance vehicle and race car application.

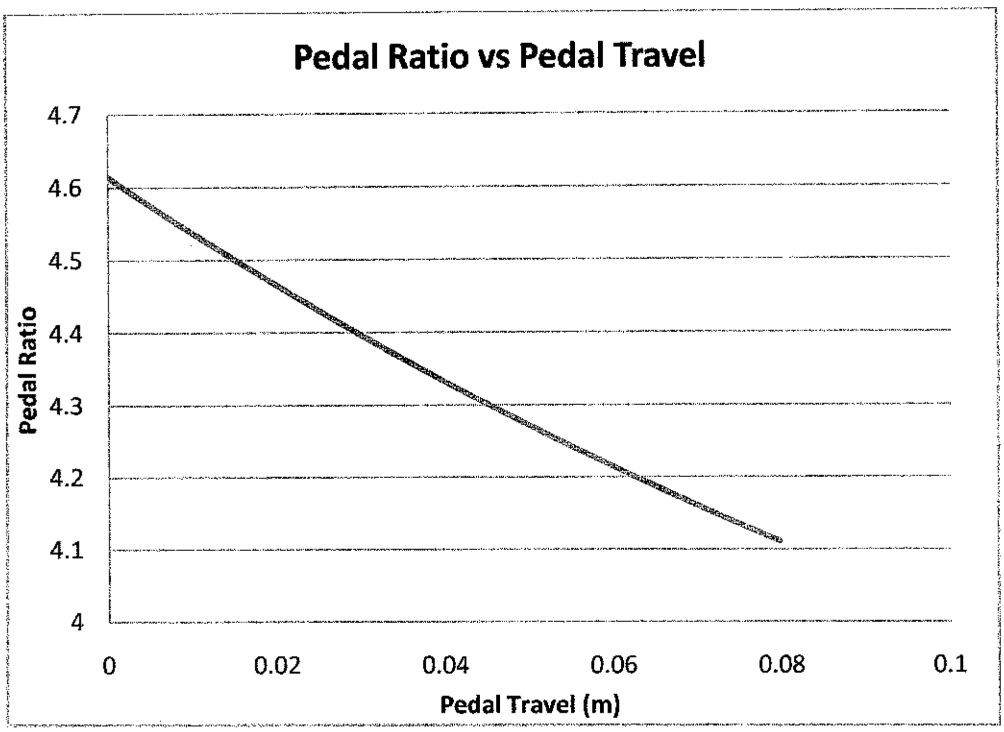


Figure 4.15: Pedal Ratio vs Pedal Travel

Pedal ratio can be defined as the mechanical advantage in order to reduce the amount of foot force applied by the driver. The pedal ratio value is differs regarding the usage of the vehicles. For the normal application such as passenger cars, the ideal value of pedal ratio value is 6.2:1 while the permissible range is from 6.0:1 to 7.0:1 [12]. For the performance car and race car application, the range is from 4.:1 to 5.5:1 [13]. The larger the pedal ratio, the greater the force multiplication [22]. The reason why the passenger cars have high pedal ratio is because the driver for passenger cars from various background and ability. Therefore it is desired to have greater force multiplication. But the higher value of pedal ratio also resulted in higher pedal travel [22]. For the race car application, this condition is not desired because the driver tends to push the pedal abruptly during high speed to slower down the car.

The next important parameter in designing the pedal box system is the required pedal force input. For this, Equation 4.4 to Equation 4.6 is used. The input force is obtained at each angle to determine the amount of force to lock-up all the wheels. Braking effort usually range between 334 and 534 N for wheels lock-up while the ideal value is 356 N [12]. In panic situation, the drivers tend to exert up until to 1779 N on the brake pedal [14]. From Figure 4.16, it can be seen that the input force required to lock up all four wheels is within the desired range of 300-400 N when the pedal travel is about 6.0 to 7.5 cm. These values then will be compared to previous design to prove the improvement of the new design.

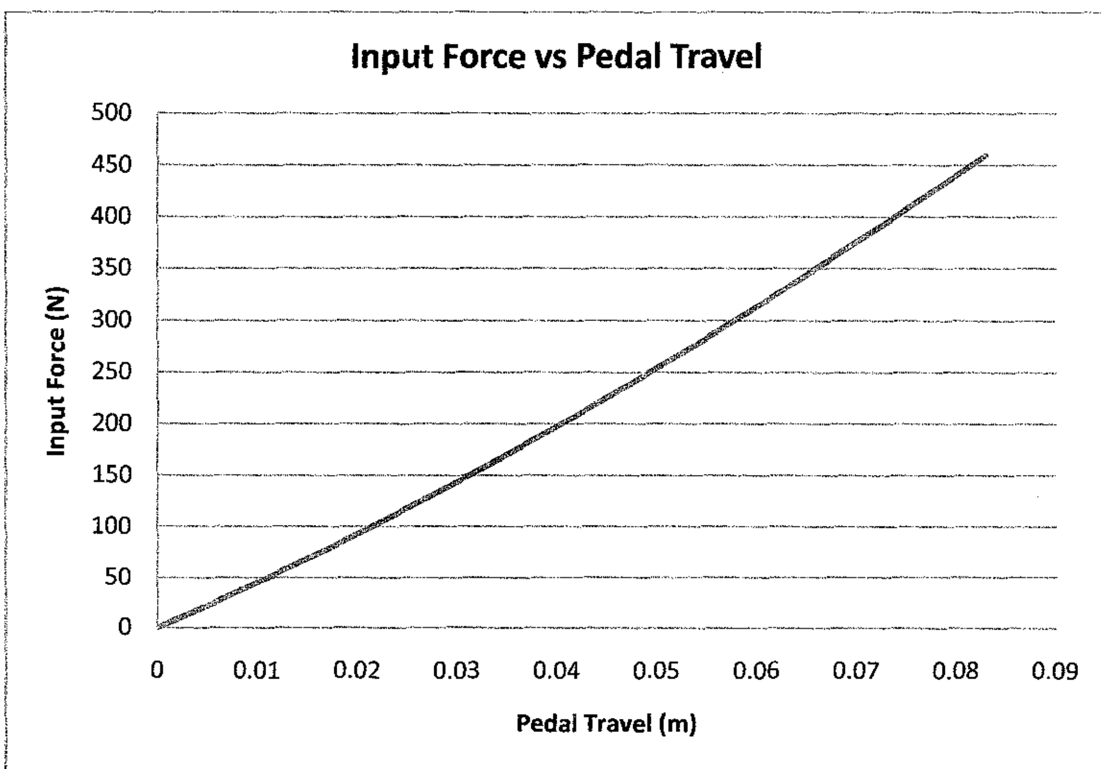


Figure 4.16: Force Input vs Pedal Travel

Figure 4.17 below shows the graph of force input versus pedal travel for previous design by Schiller (2007) [2].

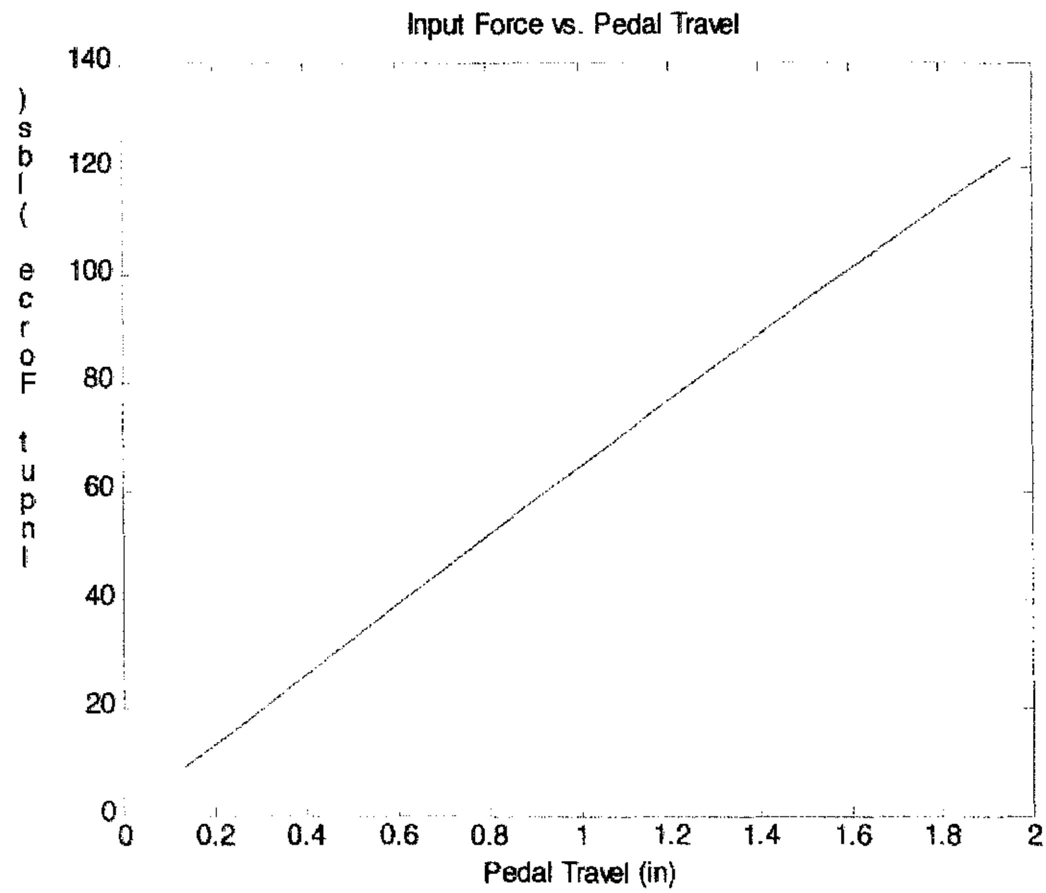


Figure 4.17: Graph of Input Force vs Pedal Travel of pedal box design by Schiller (2007) [2]

The input force required to lock up all the wheels is in the permissible range of 445-534 N (100-120 pounds) but it is not concludes the ideal value which is 356 N. It is desirable if the ideal value is lie in the designed range because the lesser the force input to depress the pedal, the better. It proves that the new pedal geometry is better and it will works nicely for the large majority of drivers.

Basically, the input force required by the driver depends on the bore size and pressure inside the master cylinders and the geometry of the brake pedal. After finalizing the pedal geometry, the master cylinders can be selected base on desired pressure and bore size.

4.7 Kinematics and Dynamics Analysis of Throttle Pedal

The objective of this section is to analyze the geometry of the throttle pedal to ensure it follows the requirement. Basically the throttle pedal design is not as critical as the brake pedal, as it is not a safety concern because the drivers do not depress the throttle pedal abruptly. It will utilise the previously discussed ergonomic considerations relating to pedal angular displacement and foot contact height. The purpose of the throttle pedal is to open the throttle of the engine's carburettor. This requires a minimal pedal force as the only resistance is supplied by the throttle return spring, which closes the throttle when no pedal force is supplied.

Figure 4.18 below is the desired geometry of the throttle pedal that will be verified in the simulation process. The dotted line is the initial position of the pedal which is inclined 10 degrees from vertical direction. The cable is fixed at the pedal located 110 cm from the pedal pivot and the cable stopper located 70 cm from the pedal. From previous research (Zarizambri, 2008) the maximum distance of throttle cable that required to fully open the throttle valve is about 45 mm [8]. Therefore, the cable needs to stretch about 115 cm.

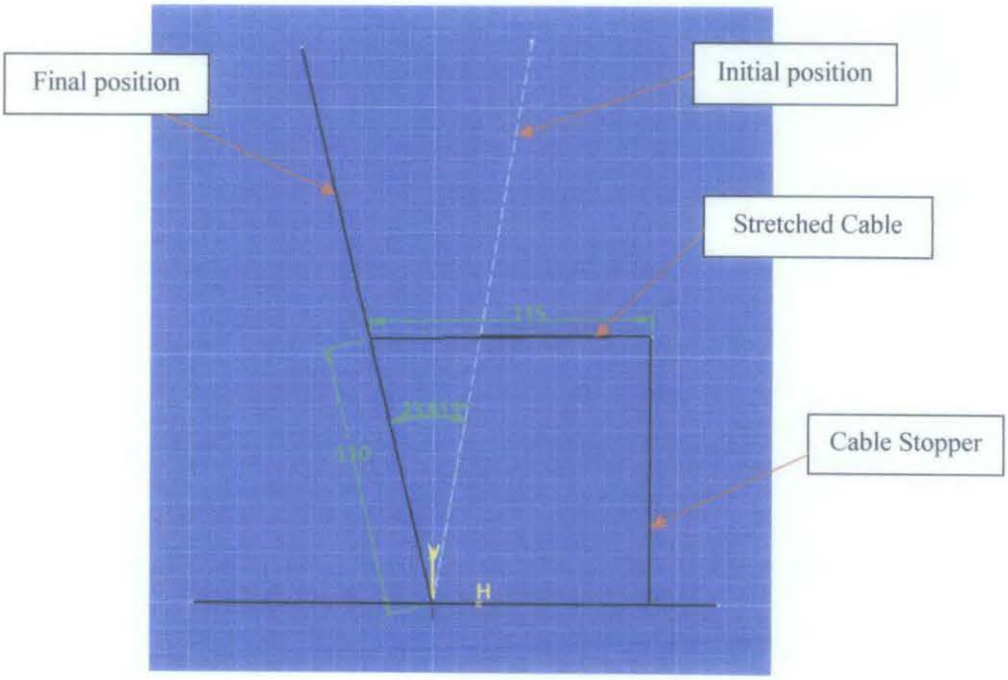


Figure 4.18: Throttle pedal geometry diagram

Simulation for throttle pedal has been done using ADAMS to verify the geometry proposed in the above figure. In ADAMS, all dimensions have been followed exactly to ensure accurate results are obtained. Figure 4.19 and Figure 4.20 below shows the results from the simulation.

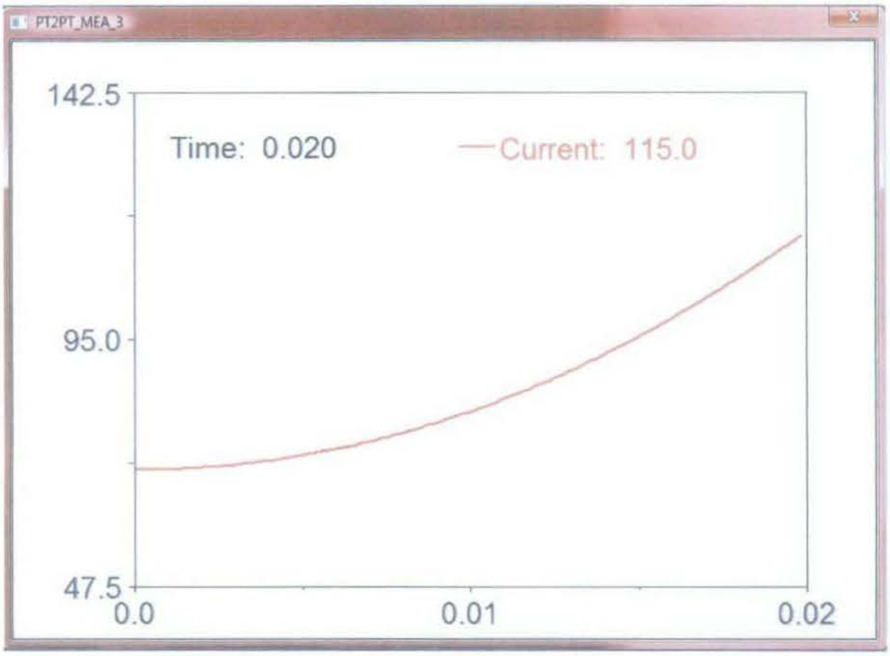


Figure 4.19: Stretched cable length vs Reaction time

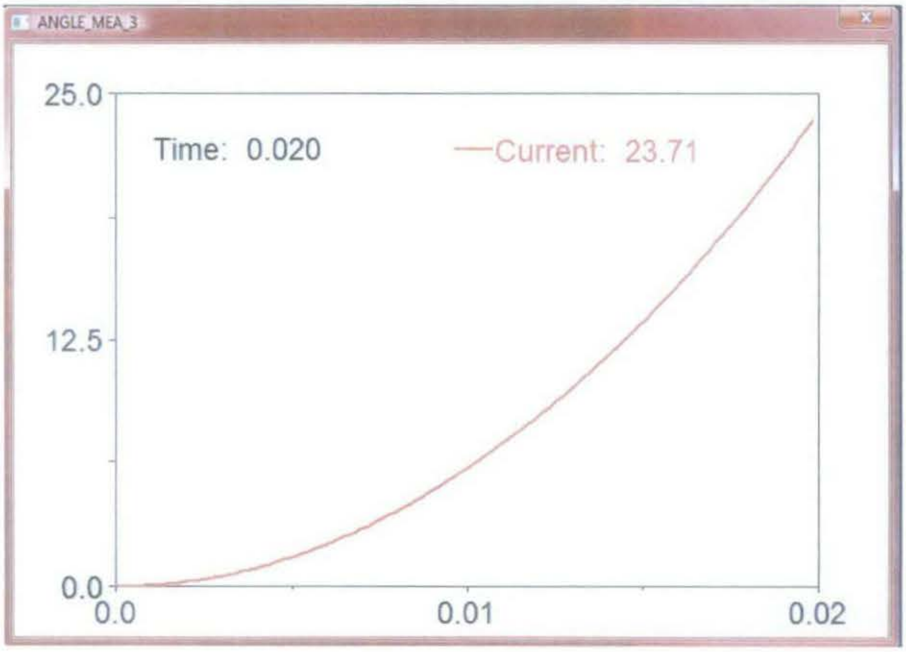


Figure 4.20: Throttle pedal angle vs Reaction time

The first graph shows the length of the stretched cable when the pedal is depressed. With specified spring stiffness coefficient and force input, it follows the desired value of cable stretched length. The second graph shows the angle, 23.71 degrees that the pedal traveled when the cable is stretched to 115 cm. This is similar to the value from the desired geometry figure which is 23.61 degrees. This verified that the throttle will be working well as desired.

It is required to ensure all the components are working well with other. Therefore everything that will be in the pedal box must be designed and selected properly. Figure 4.21 and Figure 4.22 below shows the spring elongation graph and spring force graph that were obtained from simulation process. The function of the spring is to retract back the pedal to initial position after have been depressed. The graphs will help the designer to select the suitable spring for the system.

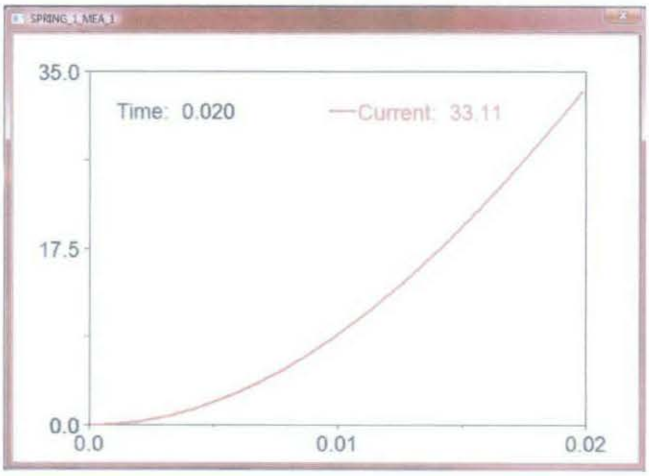


Figure 4.21: Graph of spring elongation for throttle pedal

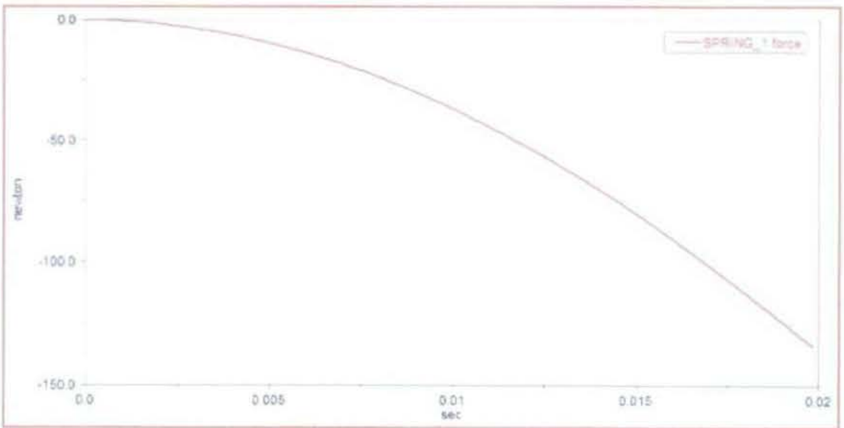


Figure 4.22: Graph of spring force for throttle pedal

4.8 Summary of Improvements for the New Pedal Box Design Compared to Previous Design

Table 4.2 below shows the summary of the improvements for the new pedal box design compared to previous design. This verified that the new proposed design is more desirable.

Table 4.2: Summary of Improvements for the New Pedal Box Design Compared to Previous Design

Parameters	Previous Design	New Design	Description
Pedal Box Dimension	309x320x210 mm (Zarizambri, 2008) [8]	255x260x246 mm The reduction of 30% in size for pedal box base	The smaller the size, the more desirable
Pedal Travel	5.08 cm (Schiller, 2007) [2]	7.0-8.0 cm	Pedal travel is designed by taking consideration the driver's feel
Pedal Ratio	3:1 (Zarizambri, 2008) [8] 4.3:1 to 4.6:1 (Schiller, 2007) [2]	4.1:1 to 4.6:1	Ideal Range (4.0:1 – 5.5:1) (Mavrigian, Carley, 1998)
Force Input	445-534 N (Schiller, 2007) [2]	300-400 N	Maximum: 534 N Ideal: 356 N (Ruiz, 2005)
Clutch Pedal Status	On the shifter (Moody, 2005) [5] Included (Previous UTP FSAE car, 2007)	On the Shifter	Depends on the designer based on complexity or familiarity

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The new design shows the reduction of the pedal box size about 30 percents from previous design. The pedal range is 7.0-8.0 cm. The result is obtained after taking considerations of driver's feel. Pedal ratio range is 4.0:1 to 4.7:1 which is in the ideal range. The input force to lock up all four wheels is within desired range 400-500 N. After taking consideration the complexity of the pedal box, the clutch is located on the shifter. These results prove the improvement of the new pedal box system. Therefore the objectives are achieved.

The implementation of the new pedal box system design will improve the performance of UTP FSAE. The project concluded for design decisions, part selection, material selection, as well as provide the all of the analysis to back up the decisions. The project that involves ergonomic study, material selection, stress-strain analysis, kinematic and dynamic analysis and fabrication process will optimized the design for the FSAE car usage.

5.2 Recommendations

The technology is expanding; therefore there is always room for improvement. It is important that the design contains several innovations as it is important to stay ahead of the curve in the competition and if the design will be used as a building block for future year's designs, it must contain new concepts that will take a while for other teams to catch up. For future development, the pedal box system should be fabricated and implemented on the car itself to see the efficiency of the system. The testing of the car should be done early to give time for improvise from the feedbacks.

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APPENDICES

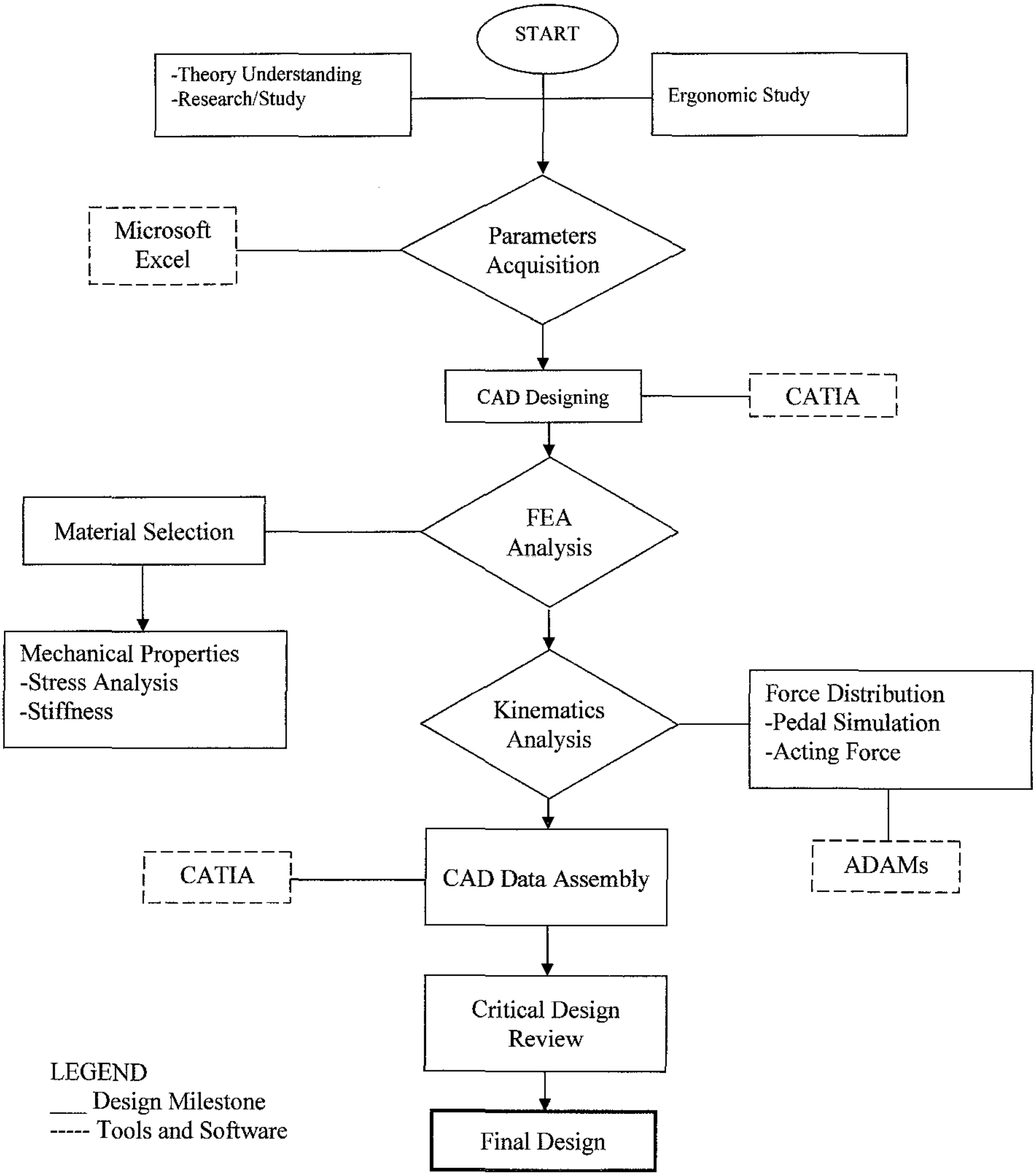
APPENDIX 1 – Ergonomic Data

Table 1 Physique (Absolute measurements) of young Japanese and American males

Characteristics	Japanese	American	Difference	P
Height (cm)	171.8 ± 5.4	180.6 ± 5.7	8.8	0.0001
Weight (kg)	62.1 ± 5.8	78.6 ± 9.1	16.5	0.0001
Sitting Height (cm)	92.7 ± 3.8	94.5 ± 2.7	1.8	0.0948
Shoulder-Elbow length (cm)	37.6 ± 1.5	42.4 ± 1.5	4.8	0.0009
Elbow-Wrist length (cm)	26.2 ± 1.0	29.3 ± 1.9	3.1	0.0001*
Thigh length (cm)	36.0 ± 1.5	40.9 ± 3.1	4.9	0.0001
Calf length (cm)	33.8 ± 1.6	40.0 ± 2.6	6.2	0.0001
Foot length (cm)	25.8 ± 1.4	28.2 ± 1.5	2.4	0.0001
Neck girth (cm)	35.9 ± 1.3	38.6 ± 2.6	2.7	0.0002*
Chest girth (cm)	87.0 ± 4.3	100.1 ± 6.0	13.1	0.0001
Biceps girth (cm)	26.6 ± 2.0	31.2 ± 3.1	4.6	0.0001
Forearm girth (cm)	25.9 ± 1.7	29.2 ± 2.0	3.3	0.0001
Abdominal girth (cm)	72.5 ± 3.7	84.9 ± 5.1	12.4	0.0001
Buttocks girth (cm)	90.2 ± 2.8	99.8 ± 5.0	9.6	0.0001*
Thigh girth (cm)	53.1 ± 3.6	56.7 ± 4.7	3.6	0.0094
Calf girth (cm)	36.5 ± 2.9	38.8 ± 2.8	2.3	0.0118

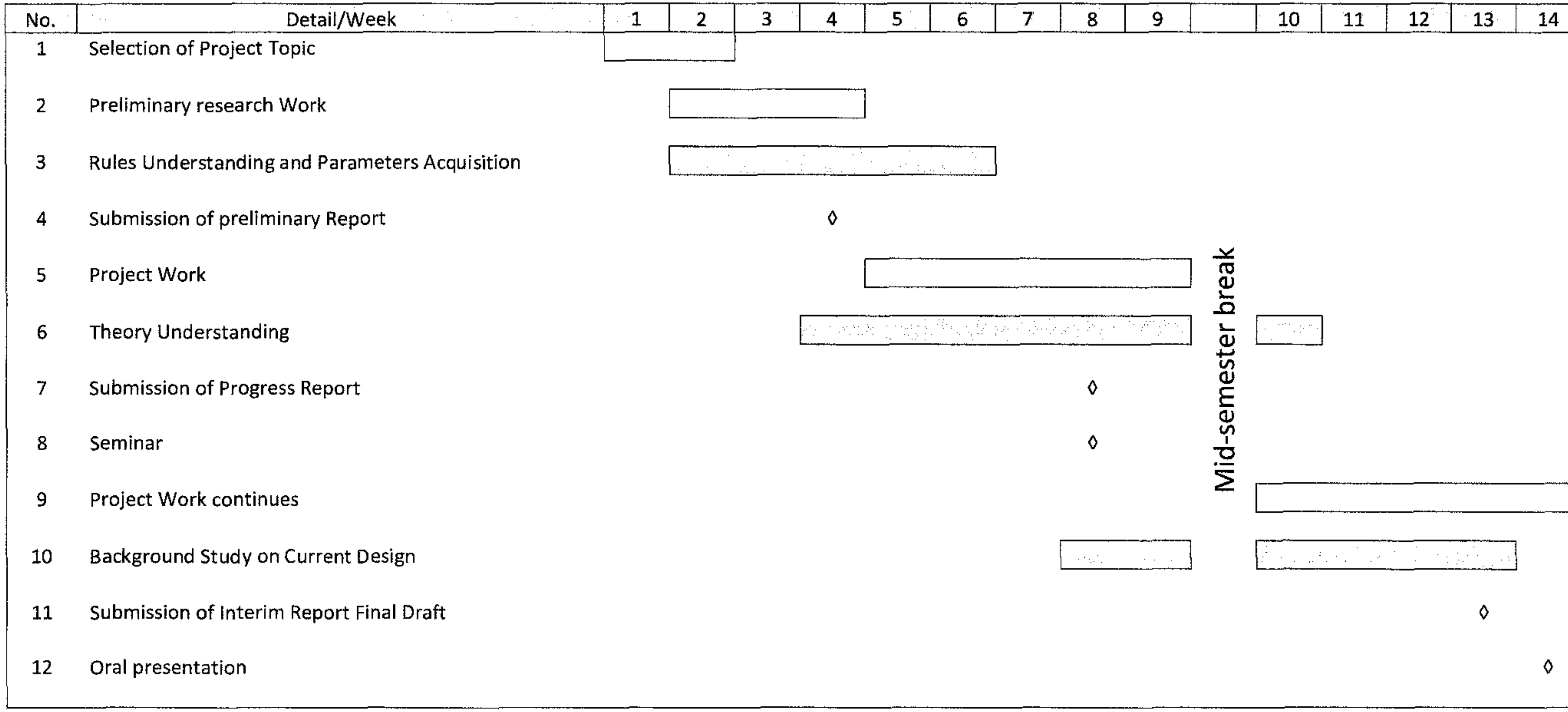
Mean ± SD. *Mann-Whitney's U tests were used.

APPENDIX 2 – Process Flow Chart



APPENDIX 3 – Gantt Chart

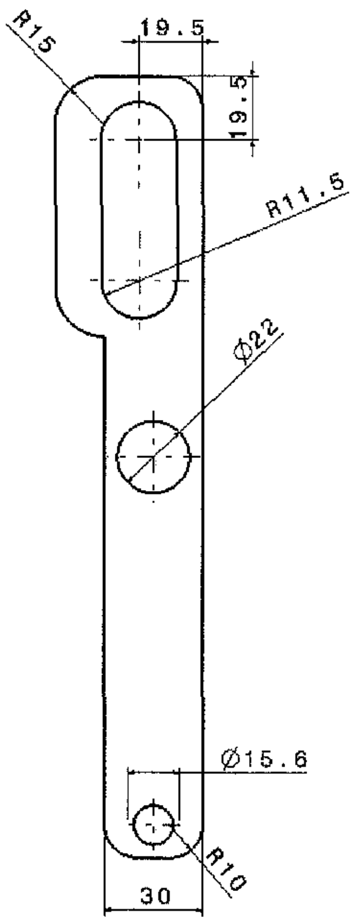
First Semester



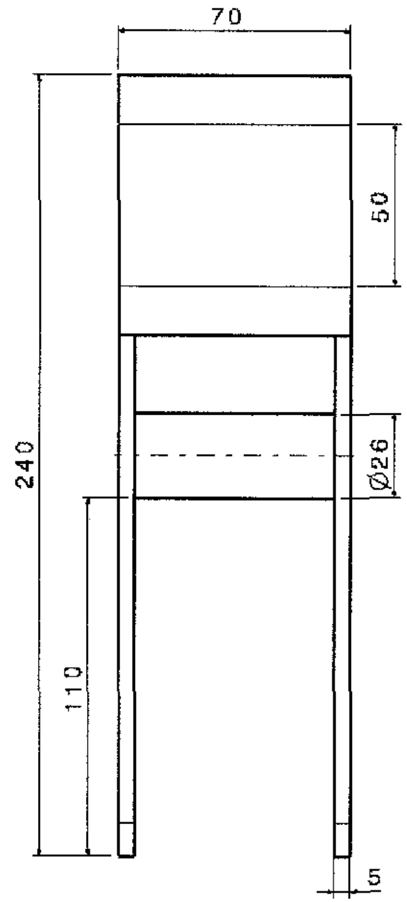
Second Semester

No.	Detail/Week	1	2	3	4	5	6	7	8	9		10	11	12	13	14	SW
1	Project Work continues										Mid-semester break						
2	CAD Designing																
3	FEA Analysis																
4	Submission of Progress Report I				◊												
5	Project Work continues																
6	Kinematics and Dynamics Analysis																
7	Submission of Progress Report 2								◊								
8	Seminar									◊							
9	Project Work continues																
10	Poster Exhibition											◊					
11	Submission of Dissertation (soft bound)															◊	
12	Oral Presentation																◊
13	Submission of Project Dissertation (hard bound)																◊

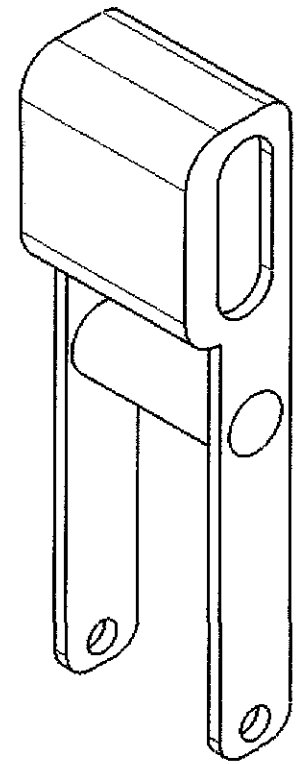
APPENDIX 4 – Brake Pedal Detail Design



Side View

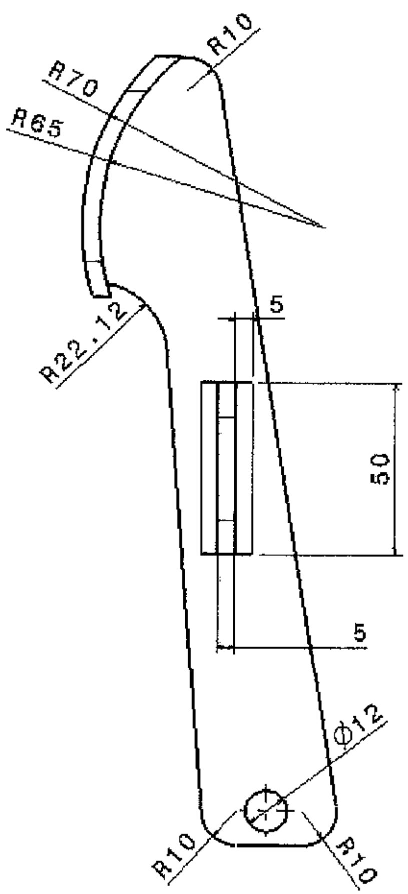


Front View

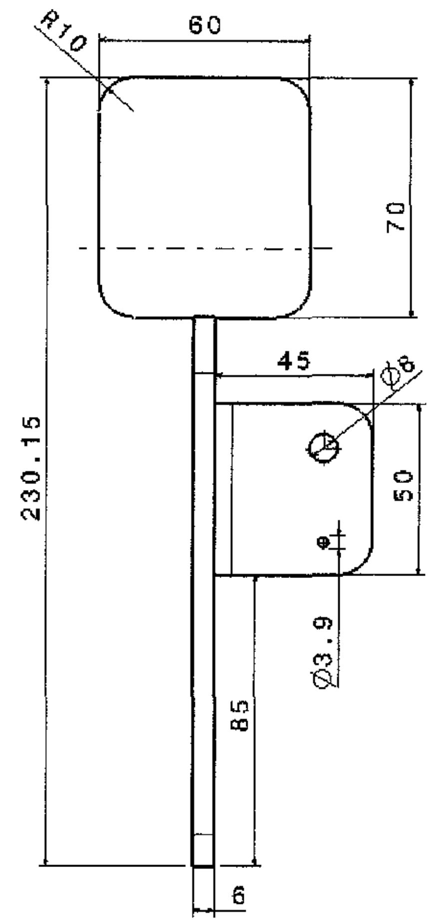


Isometric View

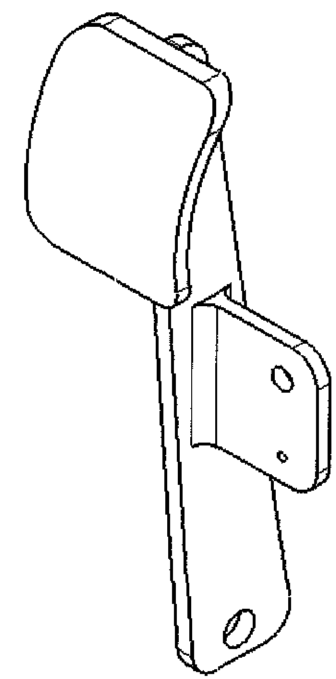
APPENDIX 5 – Throttle Pedal Detail Design



Side View

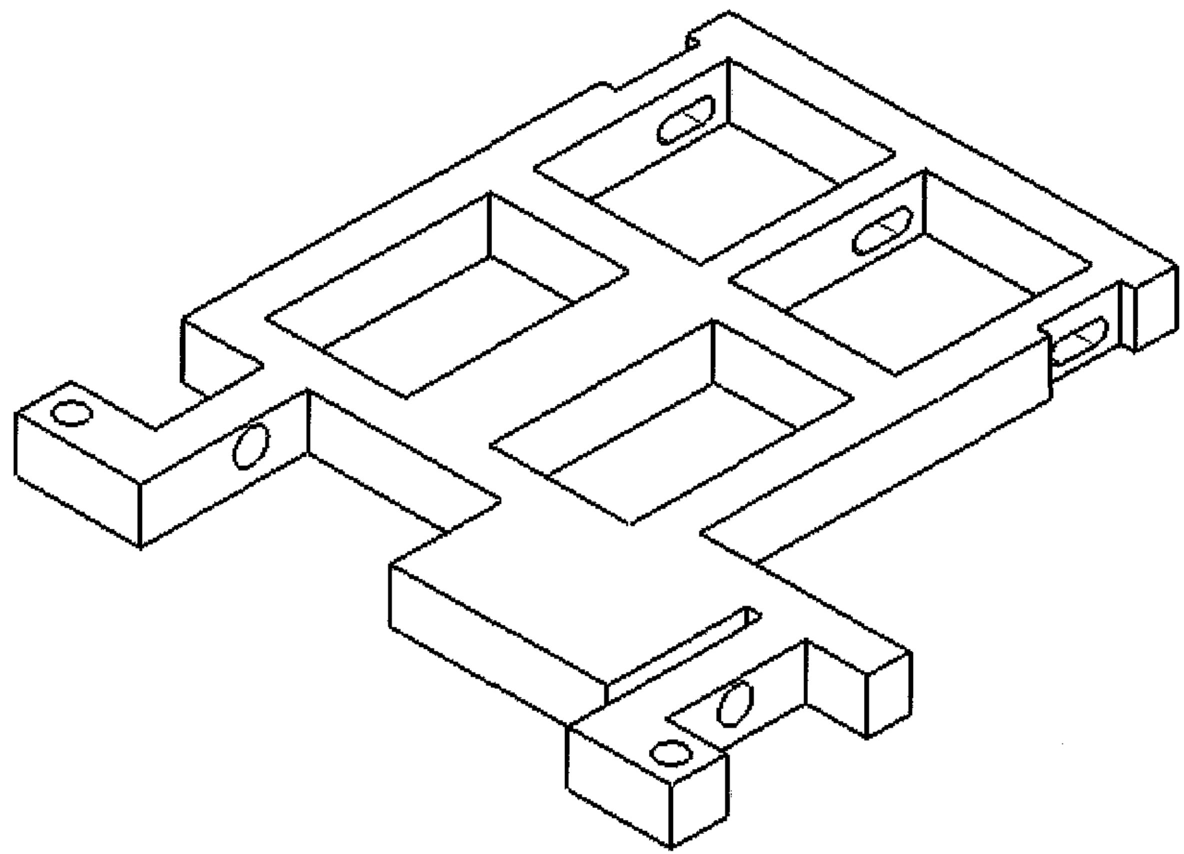


Front View

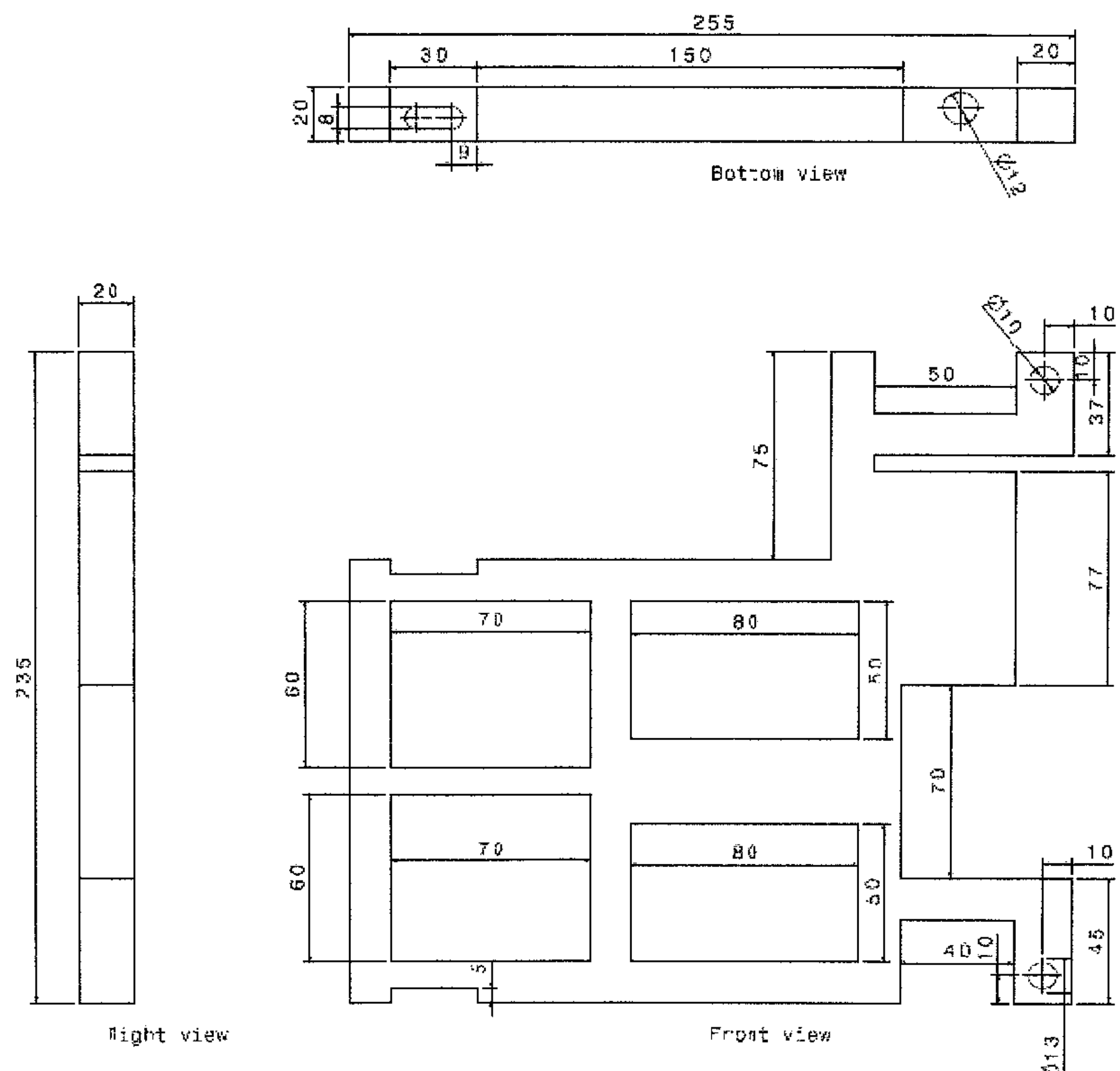


Isometric View

APPENDIX 6 – Pedal Box Mounting Detail Design



Isometric view

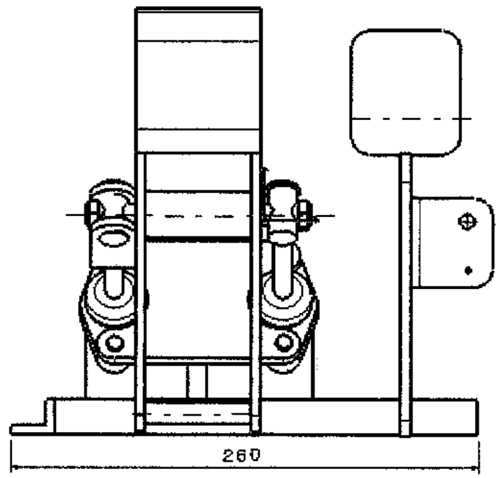

255
30
150
20
20
8
9
Bottom view
Ø12
20
235
Right view
Ø10
10
50
10
37
75
77
70
80
60
50
70
80
60
50
70
10
40
10
45
5
Ø13
Front view

APPENDIX 7 – Material Data

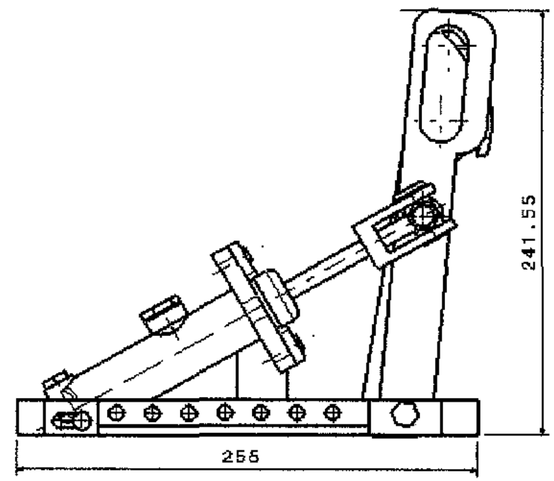
ALUMINIUM 6061-T6

Property	Value	
STRUCTURAL		
	Mpa	**ksi**
Ult. Tensile Strength	310	62.4
Yield Strength	275	46.4
Modulus of Elasticity	72000	10442
Shear Strength	205	30
	kg/m^2	
Mass Density	2850	
Hardness	95	
	1/16" thick	**1/2" thick**
% Elongation	12	17
ELECTRICAL		
	Volume	**Weight**
Electrical Conductivity	43	142
	nano ohm * m	**ohm/cir mil/ft**
Electrical Resistivity	40	24
THERMAL		
	W/m * K	**Btu/ft * n * F**
Thermal Conductivty	167	97
	10^-6/deg C	
Coeff. of Thermal Expansion	22.1	

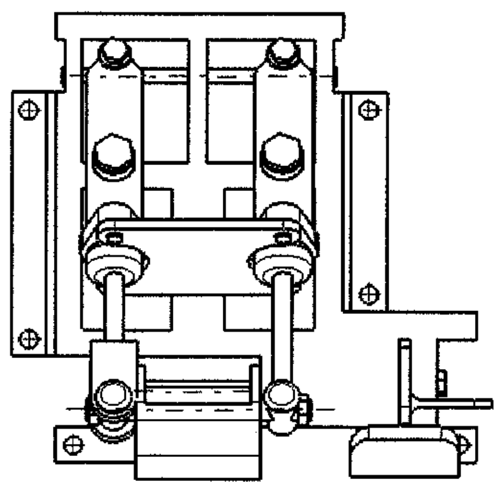
APPENDIX 8 – Pedal Box System Projection View



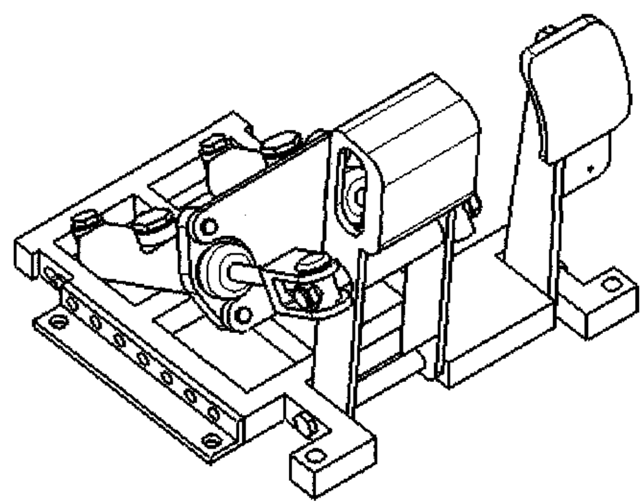
Front view



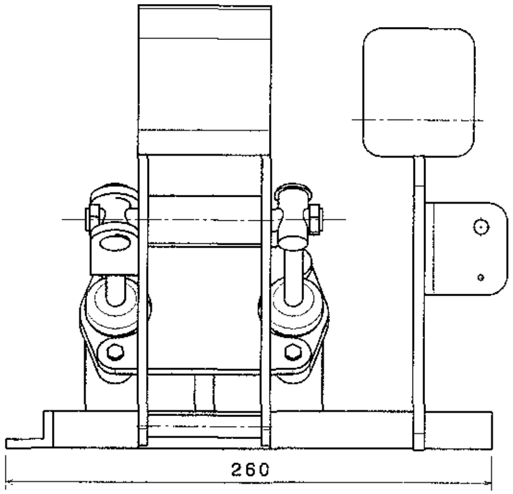
Left view



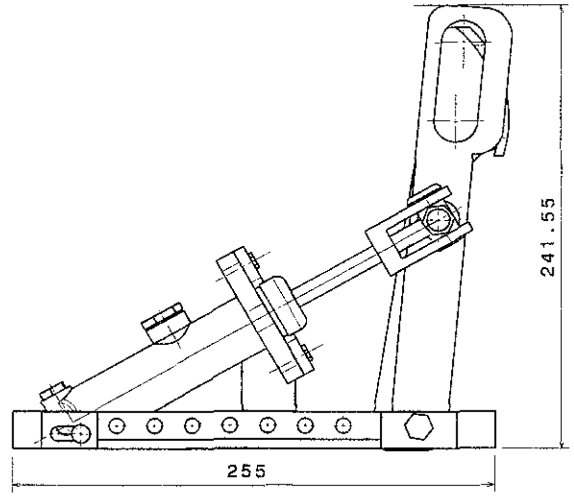
Top view



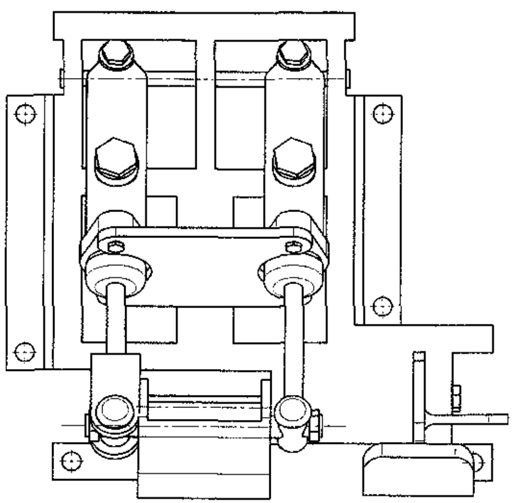
Isometric view



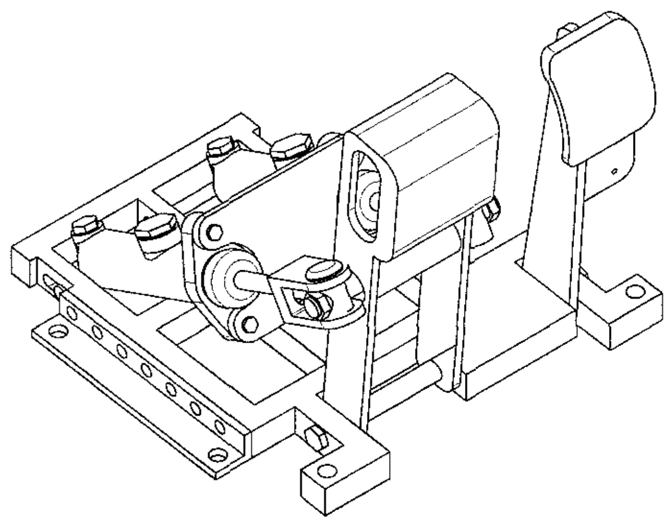
Front view



Left view



Top view



Isometric view

This drawing is our property. It can't be reproduced or communicated without our written agreement.		DASSAULT SYSTEM				
		PEDAL BOX SYSTEM PROJECTION VIEW				
DRAWN BY MAAWA	DATE 01/11/2009					
CHECKED BY	DATE xxx	SIZE A2	DRAWING NUMBER Product1			REV X
DESIGNED BY XXX	DATE xxx	SCALE 1:2	WEIGHT(kg) 0.00		SHEET 1/1	