Design of a Variable Ratio Brake Pedal

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Executive Summary

The purpose of this project was to develop a variable ratio brake pedal for future potential programs at Flex-N-Gate. The main objective was to develop a theoretical model, for a given variable ratio design, as a tool to aid in the design and development of the pedal to meet OEM requirements. A model of the pedal was generated using a spreadsheet that allowed the user to enter all the known positions of the critical points on the variable ratio pedal; the program then automatically calculates and outputs various design parameters of interest, such as the pedal travel, the path of these critical points and most importantly, the instantaneous pedal ratio throughout the entire stroke.

In order to verify and validate this theoretical model, five prototype assemblies were made and tested on Flex-N-Gate’s Pedal Ratio Tester. The results of the tests showed that the slope of the curves did correlate well with the theoretical model although an error of 40 – 48% was shown. In an attempt to improve this error, a new model was developed. The new model resulted in a theoretical curve that also matched well with the slope of the test data but yield only 20 – 32% error. The error in this new model is arguably reasonable considering all possible sources of errors that may have occurred due to design, manufacturing or testing.
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1.0 Motivation and Background

1.1 Introduction

This thesis is being written in partial satisfaction of the requirements for the degree of Bachelor of Applied Science. The purpose of this thesis project is to develop a variable ratio brake pedal for future potential programs at Flex-N-Gate. This report describes the methodology and techniques used in solving the given problem, the work that has been completed and the results of the solutions provided.

Flex-N-Gate is a Tier 1 automotive supplier whose primary customers include General Motors, DaimlerChrysler, Ford and Toyota. The Bradford facility is a 130,000 square feet plant that incorporates mechanical assembly, welding lines and metal stamping. The main products engineered in Bradford are pedals, park brakes, hinges, latches, jacks and hoists. The variable ratio pedal is an R&D project with the pedals group that is intended to capture more pedal business by providing customers with more flexibility in terms of specifying pedal ratio criteria.

The Federal Motor Vehicle Safety Standards (FMVSS) 135, section 7.11 (Appendix I), governs the operation of brake systems for personal vehicles. One of the requirements mandates brake performance for a given pedal travel in the case of panic braking. In conventional pedal designs, the rate of change of the instantaneous pedal ratio typically increases as the pedal travel increases. This implies that, for a given input load to the brake pedal, the output load of the pedal assembly increases more rapidly as the brakes are applied. However, an increasing rate of change in the brake pedal ratio results in a more aggressive brake pedal feel, which is not desirable from a human factors standpoint. The objective of this project is to design a variable ratio pedal that can control this pedal ratio such that the FMVSS requirements are met, while still providing a comfortable brake pedal feel to the driver throughout the full pedal travel.

The project was separated into five design stages and a timeline for each of these stages was assigned as is noted in Appendix A (Figure A-1). The five design stages includes: Design & Development, Prototype, Assembly, Testing and Verification/Validation & Review. This report includes a detailed description of the work done in each of these design stages and the methodology that was carried out.
1.2 Brake Pedal Mechanism

A conventional brake pedal can be regarded as a second class lever (Figure 1), where

\[ F_{\text{out}} = \frac{D_1}{D_2} F_{\text{in}} \]  

(1.2.1)

Below is a free body diagram of a conventional brake pedal (Figure 2). The pivot point on the pedal acts as a fulcrum to an applied load \( F_{\text{in}} \) at the pedal pad. The output force of the pedal, \( F_1 \), is a reaction force located at the booster pin. Both forces are perpendicular to the line joining its position to the pivot point.

\[ \sum M_{PP} = F_1 \cdot R_1 - F_{\text{in}} \cdot R_2 = 0 \]

\[ F_1 = \frac{R_2}{R_1} F_{\text{in}} \]  

(1.2.2)

where \( R_1 \) and \( R_2 \) are the distance from the output and input forces to the pivot point, respectively. The output force generated by the pedal, \( F_1 \), is larger than the input force because of the mechanical advantage.
advantage provided by the lever-type mechanism. This mechanical advantage is known as the geometric pedal ratio, or simply the pedal ratio, and is equal to $R_2/R_1$ in this case. If the positions of the applied input force and output force remain constant relative to the pivot point (i.e. $R_1$ and $R_2$ do not change), then the pedal is known as a fixed ratio pedal, as is most conventional brake pedals.

In actuality, however, the pedal arm of a fixed ratio pedal is linked to a pushrod (or booster rod) at the booster pin via a hinge-type support. The pushrod is supported radially by the booster pin and is free to rotate about the axis perpendicular to this page (Figure 3). In turn, the pushrod is attached, by a ball-and-socket joint, to the piston of the master cylinder (or brake booster) of a braking system. The pushrod is a two-force member and when it is not in line with the piston, the force exerted on the cylinder will only be a component of $F_{\text{out}}$, at an angle $\alpha$ (Figure 4). Flex-N-Gate will normally design the pedal such that $|\alpha| < 3^\circ$.

![Figure 3: Free-body diagram with pushrod](image1)

![Figure 4: Free-body diagram with pushrod at an angle $\alpha$](image2)
Now, the OEM is generally interested in the output load of the pedal that is applied directly to the master cylinder to stop the vehicle. This implies that the output force of the pedal assembly is then actually the force going into the piston of the cylinder, as opposed to $F_1$, which is shown as an internal force above. Hence, the reaction force from the master cylinder to the pushrod is denoted as $F_{out}$. Furthermore, although the ratio $F_1/F_{in}$ never changes in a fixed ratio pedal, the ratio of the force going into the piston to the applied effort will vary as the pedal travels throughout its stroke. This travel dependent ratio is recognized as the *instantaneous pedal ratio*.

### 1.3 Designing a Brake Pedal

In the design of a variable ratio pedal, the OEM must provide the pedal supplier, Flex-N-Gate (FNG), with a certain amount of information. In most cases, the OEM will initially model the brake system with a pedal having a constant geometric ratio. They will select this number based on the requirements of FMVSS 135, section 7.11 (*Appendix I*). This standard stipulates brake performance in the event of “brake power assist unit inoperative” – i.e. no assist from any of the “powered” components of the brake system (e.g. brake booster, master cylinder, etc.); this implies that all the braking effort will then be generated solely by the mechanical ratio of the pedal. Section 7.11 also states that with an applied force at the pedal pad of no more than 500 N, the vehicle must stop within 168 m, at a test speed of 100 km/h. Based on this known load at the pedal pad specified by the test, and the known geometric ratio, FNG can then calculate the amount of force transmitted to the brake booster for this panic braking scenario. For example, with a pedal load of 500 N and a 3.5 pedal ratio, 1750 N would be developed at the brake booster. The OEM will also provide a stiffness $k$ to represent the booster system. This $k$ represents the force required to stroke the brake booster pushrod to gain a certain distance. For example, if the brake booster stiffness was given as 100 N/mm, FNG could then calculate that for the FMVSS 135 test, the brake booster would be stroked 17.5 mm (1750 N ÷ 100 N/mm = 17.5 mm). When plotting an instantaneous ratio curve (pedal ratio vs. booster travel), this point is then known as the *critical point* of the curve. As long as FNG can design a pedal with a 3.5 ratio at 17.5 mm booster travel, the pedal will meet regulatory requirements, as per FMVSS 135, and the OEM can ensure that the brake system is properly sized to comply with government regulations.

### 1.4 The Variable Ratio Pedal

The purpose to designing a *variable ratio* pedal is to provide the driver with a less aggressive brake feel. Below is a typical graph of the instantaneous ratio plotted against pedal travel for a conventional *fixed ratio pedal* (*Figure 5*). Note that the curve shown represents the pedal travel only as it is being
applied and does not include the travel of the pedal when it is being released. In a conventional fixed ratio pedal, the instantaneous ratio will continue to increase after the critical point. When designing a variable ratio pedal, however, the goal is to maintain the desired ratio at the critical point (i.e. 3.5 ratio at 17.5 mm booster travel) such that the instantaneous ratio drops after that point. This will result in a pedal that meets both the regulatory criteria as well as the OEM’s subjective feel requirements.

From the graph, it is apparent that as pedal travel increases, the slope of the curve increases. This increase in slope corresponds to an increase in the rate of change in the instantaneous ratio. In other words, for a constant input force, the increment at which the output force increases becomes larger as pedal travel increases. This “accelerating” pedal ratio will cause the driver to be more inclined to propel forward as the brakes are applied, which can be displeasing for the driver and any passengers in the vehicle. By contrast, the variable ratio pedal is designed to give the driver a less aggressive brake feel by decreasing the rate of change of the instantaneous ratio with pedal travel. Ideally then, the slope of the curve, which is positive when the brakes are applied, should gradually decrease as the pedal ratio increases. The main objective of this project is to develop a model for a given variable ratio pedal design and to be able to use this model to design to OEM specifications.
2.0 Design and Development

2.1 The Model

The model was developed, using Microsoft Excel, to create a spreadsheet in an attempt to plot the instantaneous pedal ratio versus pedal travel for a variable ratio pedal. The model allows the engineer or designer to simply input the positions of all the key points of the pedal assembly on an x-y coordinate system; the program then outputs (plots) the path of these points as the brakes are applied and generates a graph of the instantaneous pedal ratio versus booster travel. More specifically, the most significant outputs of the model are: the positions of the rocker pivot, cam contact point and booster pin; the distance from the pedal pivot point to the booster pin, the pushrod stroke and pedal travel (chord); $\theta$, $\beta$, and the pushrod angularity ($\alpha$); and of course the instantaneous pedal ratio. Table 1 lists all the abbreviations used in the variable ratio model along with their definitions.

Figure 6 shows the initial positions of the 7 critical points of the variable ratio design. Please note that the point $BP$ and the line $R1$ are not on the line $R2$ (the distance from the pivot point to the centre of the pedal pad); $R1$ is the distance from the pivot point to the booster pin. In a variable ratio pedal, $R1$ (the distance between the pivot point and booster pin) is no longer constant as it was in conventional fixed ratio pedals. The reason $R1$ varies is because the booster pin is now fixed to a rocker plate, whose motion is governed by a cam bracket, rather than the pedal arm itself.

![Figure 6: Key points on the variable ratio pedal](image-url)
## Definitions:

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>PP</td>
<td>Pedal pivot</td>
</tr>
<tr>
<td>COC</td>
<td>Centre of cam</td>
</tr>
<tr>
<td>CCP</td>
<td>Cam contact point</td>
</tr>
<tr>
<td>RP</td>
<td>Rocker pivot</td>
</tr>
<tr>
<td>BP</td>
<td>Booster pin</td>
</tr>
<tr>
<td>PAD</td>
<td>Pedal pad</td>
</tr>
<tr>
<td>PN</td>
<td>Pushrod shaft nose</td>
</tr>
<tr>
<td>theta</td>
<td>Angle of pedal arm at RP relative to line through pedal pivot parallel to m/cyl C/L.</td>
</tr>
<tr>
<td>Pushrod Stroke</td>
<td>Displacement of PN along the m/cyl. C/L.</td>
</tr>
<tr>
<td>Instant. Pedal Ratio</td>
<td>The output force generated at the p/rod nose along the m/cyl C/L divided by the input force at, and perpendicular to, pedal pad.</td>
</tr>
<tr>
<td>Pushrod Angularity (alpha)</td>
<td>Angle between m/cyl. C/L and pushrod shaft C/L.</td>
</tr>
<tr>
<td>beta</td>
<td>Angle of booster point relative to line through pedal pivot parallel to m/cyl C/L.</td>
</tr>
<tr>
<td>R1</td>
<td>Distance from PP to m/cyl pushrod connecting point on pedal assembly (BP).</td>
</tr>
<tr>
<td>R2</td>
<td>Distance from PP to the centre of the pedal pad.</td>
</tr>
<tr>
<td>HP</td>
<td>Perpendicular distance from pedal pivot to line through m/cyl C/L.</td>
</tr>
<tr>
<td>PL</td>
<td>Length along pushrod shaft C/L from the nose of the pushrod (PN) to the eye (at BP).</td>
</tr>
</tbody>
</table>

### Table 1: Variable ratio pedal abbreviations and definitions
2.2 Derivations

2.2.1 The Instantaneous Pedal Ratio

Assume that the applied force to the centre of the pedal pad ($F_{in}$) is always perpendicular to the line joining the pivot point to the centre of the pedal pad ($R_2$). Also recall that beta is defined as the angle between $R_1$ and the horizontal (parallel to the master cylinder centerline).

\[
\text{Inst. Ratio} = \frac{F_{out}}{F_{in}} \quad (2.2.1)
\]

\[
\text{Pedal Ratio} = \frac{F_1}{F_{in}} = \frac{R_2}{R_1} \\
\frac{1}{F_{in}} = \frac{R_2}{R_1} \cdot \frac{1}{F_1} \quad (2.2.2)
\]

Substituting \((\text{Equation } 2.2.2 \text{ into } 2.2.1)\),

\[
\text{Inst. Ratio} = \frac{R_2}{R_1} \cdot \frac{F_{out}}{F_1} \quad (2.2.3)
\]

From the diagrams,

\[
F_m = F_1 \cos(90^\circ - \beta - \alpha) \quad (2.2.4)
\]

(see 3 cases)

Also,

\[
F_{out} = F_m \cos \alpha \quad (2.2.5)
\]

where $F_m$ is the force along the pushrod shaft centerline (two-force member).

Substituting \((\text{Equations } 2.2.4 \& 2.2.5 \text{ into } 2.2.3)\),

\[
\text{Inst. Ratio} = \frac{R_2}{R_1} (\cos \alpha)(\cos(90^\circ - \beta - \alpha))
\]

Note that $R_2$ in the above equation is the only variable that remains constant; all other variables vary with pedal travel.
2.2.2 Rocker Point (RP)

Define $|PPRP|$ as the distance between the pivot point (PP) and the rocker point.

Therefore, the two boxed equations $(x_n, y_n)$ are the coordinates of the rocker point, calculated by utilizing the point preceding it $(x_{n-1}, y_{n-1})$. 

\[
\begin{align*}
\text{Case 1: } \theta &> 90^\circ \\
&
\begin{align*}
x_n &= x_{n-1} - |PPRP| \left[ \sin(\theta_n - \theta_0) - \sin(\theta_{n-1} - \theta_0) \right] + x_{n-1} \\
y_n &= y_{n-1} - |PPRP| \left[ \cos(\theta_n - \theta_0) - \cos(\theta_{n-1} - \theta_0) \right] + y_{n-1}
\end{align*}
\end{align*}
\]

\[
\begin{align*}
\text{Case 2: } \theta &< 90^\circ \\
&
\begin{align*}
x_n &= x_{n-1} - |PPRP| \left[ \sin(180 - \theta_n) - \sin(90 - \theta_n) \right] \\
y_n &= y_{n-1} - |PPRP| \left[ \cos(180 - \theta_n) - \cos(90 - \theta_n) \right]
\end{align*}
\end{align*}
\]
2.2.3 Cam contact point (CCP)

The cam contact point is essentially the position of the moulded roller as it rolls along the groove of the cam bracket.

Two points exist for the intersection of the two circles. \( CCP(x,y) \) was chosen such that a continuous path was plotted.
2.2.4 Booster Pin (BP)

The positions (path) of the booster pin were found similarly to the position of the CCP derived on the previous page.

The intersecting point that creates a continuous curve with respect to the initial position of BP was chosen.
2.2.5 Pushrod Stroke

The pushrod stroke, or booster travel, is defined as the displacement [mm] of the pushrod shaft nose in the x-direction. Therefore, it is equal to the travel of the piston as the brakes are applied.

Define $|BPPN|$ as the distance between the booster pin ($BP$) and the pushrod nose ($PN$).

As anticipated, the pushrod stroke is dependent on the position of the booster pin and the angle $\alpha$. 
2.2.6 Pushrod Angularity

The pushrod angularity is defined as the angle [degrees] between the master cylinder centerline and the pushrod shaft centerline. Below is the derivation of $\alpha_n$ in 3 different scenarios for ($\alpha$ and $\beta$).

In all 3 cases,

$$\alpha_n = -\sin^{-1} \left[ (R_j - \frac{HP}{\cos(90^\circ - \beta_n)}) \frac{\cos(90^\circ - \beta_n)}{PL} \right]$$  \hspace{1cm} (2.2.6.1)
2.2.7 Pedal Travel (chord)

The pedal travel refers to the displacement of the pedal pad. The distance is measured as the chord of the curved path, rather than the sector length. Recall that theta is the angle between the line joining the rocker pin to the pivot point, and the line through the pivot point parallel to the master cylinder centerline.
2.2.8 Beta ($\beta$)

Beta is defined as the angle of the booster point relative to the line through the pedal pivot parallel to the master cylinder centerline.

If $BP_x > 0$ (i.e. $BP$ is to the right of $PP$ on the $xy$-plane), then:

$$\beta_n = 90^\circ - | \tan^{-1} \left( \frac{BP_y}{BP_x} \right) |$$  \hspace{1cm} (2.2.8.1)

otherwise,

$$\beta_n = 90^\circ + | \tan^{-1} \left( \frac{BP_y}{BP_x} \right) |.$$  \hspace{1cm} (2.2.8.2)

2.2.9 Theta ($\theta$)

Theta is defined as the angle of the pedal arm (rocker pivot) relative to the line through the pedal pivot parallel to the master cylinder centerline.

If $RP_x > 0$ (i.e. $RP$ is to the right of $PP$ on the $xy$-plane), then:

$$\theta_i = 90^\circ - | \tan^{-1} \left( \frac{RP_y}{RP_x} \right) |$$  \hspace{1cm} (2.2.9.1)

else,

$$\theta_i = 90^\circ + | \tan^{-1} \left( \frac{RP_y}{RP_x} \right) |.$$  \hspace{1cm} (2.2.9.2)

For this spreadsheet, theta is used to control the increment size of the pedal travel. Thus, the resolution of the instantaneous ratio curve to pedal travel can be controlled and is directly related to the increment size. For example, if the increment is chosen to be half a degree, then:

$$\theta_n = \theta_{n-1} + 0.5^\circ.$$  \hspace{1cm} (2.2.9.3)

2.2.10 $R_1$

Finally, $R_1$ is defined as the distance between the pedal pivot and the pushrod connection point on the pedal assembly (booster pin).

$$\left( R_1 \right)_n = \sqrt{(BP)^2_{x,n} + (BP)^2_{y,n}}$$  \hspace{1cm} (2.2.10.1)
2.3 Variable Ratio Pedal Model

Attached in Appendix B is the spreadsheet developed for the prototype variable ratio pedal. The model in the top-right corner tracts the path of the triangular rocker plate and its three critical points ($BP$, $RP$ and $COP$). The instantaneous ratio is plotted against the pushrod stroke displayed in the bottom-right corner. It is apparent from this graph that the rate of change in the instantaneous ratio for this variable ratio pedal is decreasing as pedal travel increases, as desired.

The row that is highlighted in yellow represents the approximate typical maximum pushrod stroke required by the OEM. Thus, all the data below this row would be irrelevant for a max pushrod stroke criterion of $21.63 \text{ mm}$. The row highlighted in orange represents the travel at which testing was performed. Furthermore, the column of numbers that are highlighted near the bottom represents all pushrod angularities ($\alpha$) greater than 3 degrees. As discussed earlier, Flex-N-Gate typically designs the pedal such that this angle is smaller than 3 degrees relative to the master cylinder centerline.
3.0 Prototype

3.1 Methodology

The second stage of the project involved designing prototype parts for testing in order to verify the model created in the first stage. A solid model of the variable ratio pedal assembly was provided by the UG designer in the pedals group. Engineering prints were created for all the prototype parts from this model and were provided to the prototype department to fabricate. The prints specified the materials to be used and the tolerances on the parts. A review of the general assembly of the pedal was also conducted. The drawings were created using CADKEY and sent to all parties involved in the project, including the senior product engineer, the tooling manager, the prototype department supervisor and the machinists (Appendices C & D).

The prototype variable ratio brake pedal was designed and based off the current production C-Segment brake assembly. All common parts that could be used from the existing C-segment pedal were employed in the prototype variable ratio assemblies. This included the Housing, Pedal Pad Plate, Pedal Pad, Pivot Tube, Pivot Pin and Pedal Bushings. Figures 7, 8 and 9 display some of the components required in the assembly. A list of all the parts is shown in the Bill of Materials (BOM) and can be found in Appendix E. Included in the BOM are the quantity per assembly, material specification and whether it is a production, prototype or purchased part.

![Solid CAD model of the variable ratio brake pedal](image)

**Figure 7:** Solid CAD model of the variable ratio brake pedal
Figure 8: Front-left side of the variable ratio brake pedal

Figure 9: Close up of the cam mechanism
3.2 Material Selection

In general, all the pins were made from cold-rolled steel (CRS 1045) and the plates made of high-strength, low-alloyed (HSLA) annealed steel (XLF 050). Both these materials were chosen for their excellent stiffness and wear-resistant properties to prevent deformation during testing of the parts. These materials are also easily machinable as the processes for fabricating these parts were also taken into consideration during material selection. Another factor that was considered when choosing an appropriate material for these prototypes was cost. In order to reduce cost, stock materials were used to avoid ordering custom-sized material and to cut down on machining costs.

3.3 Dimensioning and Tolerancing

One way to minimize the sources of error when comparing the actual test data to the theoretical model is to specify very tight tolerances to all the parts to be made. This will avoid any excess slack between mating components and reduce the error in the pedal ratio, since it is based on relative positions. There is a limit, however, to the amount of accuracy that can be achieved because different machining processes will have different tolerance capabilities. For instance, knowing that the cam bracket was to be laser cut, the prototype supervisor was consulted to obtain the smallest tolerance possible for the laser cutter to be used prior to specifying the tolerances for the cam bracket on the drawing.

Notice in the drawings of Appendix D that dimensions and tolerances were only specified for critical features (e.g. pin diameter, hole diameter, etc.); all other features were made to the general tolerance of the drawing and the dimensions from the solid model. The tolerances for mating parts were established using existing similar mating components in current production pedal assemblies. For example, the current production C-Segment booster pin and pedal arm were used to find the tolerance for all spun pin-hole fits, where the pin was to be spun like a rivet. The diameter of the booster pin and the corresponding hole size on the pedal arm were reviewed in order to find the minimum and maximum tolerance between the two mating features. Once this tolerance range was obtained (0.1 - 0.5 mm), the dimensions for all the pins in the prototype assembly that were riveted were calculated based on this tolerance. This method in calculating tolerances and holes sizes was used for all other mating components as well, which included rotating fits, slide fits and parts being welded together. Table 2 shows a chart of all the pin-hole type dimensions and tolerances that were considered between mating components.
### Table 2: Tolerances used in mating components

<table>
<thead>
<tr>
<th>PART NAME</th>
<th>DIMENSIONS [mm]</th>
<th>TOLERANCE RANGE [mm]</th>
<th>MATING TYPE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Production Parts</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Booster Pin</td>
<td>Ø10.7 Ø10.9</td>
<td>0.1 - 0.5</td>
<td>Spun</td>
</tr>
<tr>
<td>Pedal Arm hole</td>
<td>Ø11.0 Ø11.2</td>
<td>0.05 - 0.35</td>
<td>Weld</td>
</tr>
<tr>
<td>Pivot Tube</td>
<td>Ø19.05 Ø19.15</td>
<td>0.0 - 0.4</td>
<td>Rotate (metal on metal)</td>
</tr>
<tr>
<td>Pedal Arm hole</td>
<td>Ø19.20 Ø19.40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Metal Pin</td>
<td>Ø8.0 Ø8.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Metal Roller ID</td>
<td>Ø8.2 Ø8.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Prototype Parts</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pedal Arm hole</td>
<td>Ø11.96 Ø12.16</td>
<td>0.1 - 0.5</td>
<td>Spun</td>
</tr>
<tr>
<td>Rocker Pin</td>
<td>Ø8.3 Ø8.5</td>
<td>0.1 - 0.4</td>
<td>Rotate (metal on metal)</td>
</tr>
<tr>
<td>Rocker Plate</td>
<td>Ø8.6 Ø8.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Booster Pin</td>
<td>Ø10.7 Ø10.9</td>
<td>0.1 - 0.5</td>
<td>Spun</td>
</tr>
<tr>
<td>Rocker Plate</td>
<td>Ø11.0 Ø11.2</td>
<td>0.1 - 0.5</td>
<td>Spun</td>
</tr>
<tr>
<td>Cam Pin</td>
<td>Ø9.6 Ø9.8</td>
<td>0.1 - 0.5</td>
<td>Spun</td>
</tr>
<tr>
<td>Rocker Plate</td>
<td>Ø9.9 Ø10.1</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

#### 3.3 Manufacturing and Functionality

The following section contains images of the prototype parts, brief descriptions on how they were processed and the functionality of the part. There were a total of 7 different prototype parts machined by the prototype department.

##### 3.3.1 Pedal Arm

The input force from the driver as he/she is braking is essentially applied to the base of the pedal arm as it pivots about the pivot pin. The smaller hole on the arm holds the rocker plate via the rocker pin (*Figures 10a and 10b*).

![Figure 10a: Pedal arm (side view)](image1) ![Figure 10b: Pedal arm (front)](image2)

The pedal arm was created using Flex-N-Gate’s Universal Pedal Arm Fixture. The offset in the pedal does not have an effect on the pedal ratio.
The flat portion of the arm is securely clamped to the fixture.

**Figure 11a:** Flex-N-Gate’s Universal Pedal Arm Fixture (*front*)

The top half of the fixture presses down on the pedal arm, creating an offset. The pedal arm is placed in two different positions in the same fixture to create the two bends in the arm.

**Figure 11b:** Pedal Arm Fixture (*close-up*)

**Figure 11c:** Pedal Arm Fixture (*back*)
3.3.2 Cam Bracket

The cam bracket is the key component in creating a variable ratio for the pedal (*Figure 12*). The cam bracket guides the rocker along a prescribed path which forces the booster pin (rod) to follow a path that is directed related to it. As the booster pin moves along the path dictated by the cam bracket, the distance between the pin and the pivot point is no longer fixed as it is in conventional pedals, thus, creating a variable ratio.

![Cam Bracket](image1.png)  
*Figure 12: Cam Bracket*

The cam bracket is welded to the housing using a weld fixture (*Figure 13*) that was custom made for this bracket.

![Weld fixture](image2.png)  
*Figure 13: Weld fixture*
The cam brackets were outsourced to a local laser cutter and formed in-house. Above are some images of the hydraulic press that was used (Figures 14 and 15) and below are the two die halves used to form the bracket (Figure 16).
3.3.3 Cam Pin

The cam pin (Figure 17) is used to hold the molded roller to the rocker plate. It was turned using a lathe. Notice that there is a groove located on one end of the pin; this groove fits a retaining ring that is used to keep the molded roller on the pin. The other side of the pin is spun onto the rocker plate.

3.3.4 Rocker Pin

The rocker pin (Figure 18) is spun onto the pivot arm and is used as a pivot point for the rocker plate. The rocker plate sits on the second smallest diameter while a washer is fitted on the smallest diameter (step). The step is then spun against the washer allowing the rocker plate to rotate freely about the second smallest diameter.

3.3.5 Booster Pin

This pin is spun onto the rocker plate and holds the pushrod that actuates the master cylinder on the other end (Figure 19). The pushrod is constrained in all directions and is only free to rotate about the axis that is parallel to the length of the booster pin.

3.3.6 Rocker Plate

The rocker plate (Figure 20) essentially pivots about the rocker pin and links the motion of the cam roller to the booster pin (or pushrod). It was produced using an in-house CNC wire cutter.
3.3.7 Moulded Roller Assembly

The roller (Figure 21) is used to guide the variable ratio mechanism along the path of the cam bracket. The roller insert is made of 30% glass-filled nylon with 15% PTFE (Polytetrafluoroethylene), which provides hardness and lubrication respectively.

Figure 21: Moulded roller assembly

This roller was modified from a production part that is normally used in the hinge (Figure 22). One side of the flange was requested to be machined off (Figure 21) in order to be able to fit the roller into the cam bracket.

Figure 22: Current production hinge
4.0 Assembly

4.1 Dimensional Layout

Prior to assembly, all completed prototype parts were checked for their dimensional accuracy. A brief dimensional layout was performed to determine if the parts were made within specifications as per drawings. Only key dimensions were measured (i.e. mating features) using a digital caliper and height gage. The results of the layout are not included in this report but all key dimensions did indeed meet all tolerances specified, thanks to the prototype department.

4.2 Process

Below are the 7 prototype parts fitted together prior to assembly (Figure 23). As discussed in the previous section, the assembly of the brake pedal consisted of a variety of processes that included welding, riveting and a retaining ring. To best illustrate the use of these means of assembly, the locations of these various processes are shown in the completed final assembly displayed in Figures 24 – 28. Furthermore, a marked up drawing of the pivot tube is attached in Appendix F showing the appropriate positions for welding of the pedal plate and pivot tube relative to the pedal arm.

Figure 23: Parts fitted together prior to assembly
Figure 24: Completed final assembly (*isometric view*)

Figure 25: Completed final assembly (*front-left close-up*)

Retaining ring

riveted
Figure 26: Completed final assembly (right side close-up)

Figure 27: Completed final assembly (bottom of pedal pad plate)

Figure 28: Completed final assembly (pivot tube close-up)
4.3 Modifications

Although all the prototype components were made as per drawing and within specifications, there were a few modifications required during assembly. The following is a list of these modifications described in more detail.

1. There were essentially two main sub-assemblies that were assembled prior to the final assembly: the first included the pedal arm with all three pins, the roller, pivot tube and rocker plate attached; the other was the housing with the cam bracket welded to it. In order to integrate the two sub-assemblies together, the front of the housing was cut off to allow the roller to slip into the cam bracket (Figure 29 & 30).

2. The current production C-segment brake does not include any sort of roller/cam mechanism. Therefore, a second section of the housing was removed to provide clearance for the path of the roller in the variable ratio assembly (Figure 29 & 30).

3. The roller-to-cam fit on assemblies #2, 3 and 4 seemed a little tight after assembly. This was not a surprise as a very tight tolerance was specified on the mating features of the two parts in an attempt to gain maximum positioning accuracy. In order to relieve some of the friction between the two parts, the bottom surface of the cam profile was grinded until the roller rolled smoothly as intended. This modification has no effect on the dynamics of the pedal because the force exerted on the roller by the cam bracket is only from the top surface. In fact, the bottom of the cam bracket may be omitted all together to save material and cost for future builds.

4. The booster pins were prototyped off a marked up production drawing. Although there were no changes made to any of the diameters on the part, the booster pin had to be modified as the pin diameter that fits into eye of the booster rod on the tester was too large. The reason why there was interference was because the diameter on the current production C-segment booster pins was made below the nominal value while the prototype booster pins were machined above the nominal. The eye (hole) of the booster rod, however, was fabricated at the nominal value.
**Figure 29:** Housing (before modifications)

**Figure 30:** Housing (after modifications)
5.0 Testing

5.1 Equipment

All five pedal assemblies were tested on Flex-N-Gate’s Pedal Ratio Tester in order to obtain actual data that can be compared to the theoretical model developed. Shown below are the tester (Figures 31a and 31b) and the controller (Figure 32). The tester is mainly composed of two pneumatic cylinders, a potentiometer, two load cells and a displacement gage. One cylinder is used to drive the pedal pad while the other is used as a resistive load on the booster rod. The cylinder used to drive the pedal is actually connected to a swing bar that rotates about the same axis as the pedal arm (i.e. the swing bar’s axis of rotation passes through the pivot pin); the actuator used to physically press the pedal pad is also fixed onto the swing bar. A potentiometer is used to measure the rotation angle of the swing bar as it is driven by the cylinder. The pedal assembly is simply bolted onto a custom fixture used for C-segment brakes and the eye of the booster rod is fitted onto the booster pin. One of the two load cells is located near the nose of the booster rod, in-line with the resistive load, and measures the output force of the pedal (Figure 33); the other is place at the actuator to measure the input force (Figure 34).

Figure 31a: Flex-N-Gate’s Pedal Ratio Tester (front view)
Figure 31b: FNG’s Pedal Ratio Tester (*iso view*)

Figure 32: Controller

Figure 33: Booster rod connection

Figure 34: Actuator

The controller is essentially used to actuate the two pneumatic cylinders and read the measurements off the load cells, displacement gage and potentiometer. The controller also allows the test to be carried out automatically by driving the actuator at a slow and steady interval while recording all the data onto the CPU. A procedure for this automatic cycle test is described in a later section (5.3).
5.2 Test Setup

Flex-N-Gate’s Pedal Ratio Tester was designed and built as a universal tester to test all of Flex-N-Gate’s present and future pedal assemblies. The main adjustment that had to be made to the tester was a change in the custom-designed fixture plates specific for each assembly type. There are three fixture plates in total: One plate is used to mount the pedal housing while the other two are used to support the arms of the swing bar (Figure 35). One of the main reasons the variable ratio pedal was designed off the current production C-Segment assembly was to purposely make use of the already existing fixture plates for this tester. Note that the plates that support the arms of the swing bar are assembly-specific because they ensure that the rotation axis of the swing bar passes through the pivot pin and thus, the pedal’s axis of rotation.

![Figure 35: Fixture plates changeover](image)

The model that was developed for the variable ratio pedal assumed that the input force on the pedal pad remained perpendicular throughout the entire travel. Several additional adjustments were made during the test setup to ensure that this stayed true (Figure 36a and 36b).
5.3 Test Procedure

In an effort to test all five pedal assemblies consistently, four of the five assemblies (assembly #1 excluded) were tested on the same day to avoid any variations in the test setup. Due to time limitations, each assembly was tested a minimum of two cycles during automatic cycling and a minimum of three cycles during manual testing. In an automatic test, the controller is programmed to apply a set input load of 50 lbf to the pedal pad by keeping the assembly static and balancing the resistive output load at the booster rod. This load is maintained constant as the booster output travel is increased at the default input speed of 0.3 mm/s (Figure 37). The test is completed when it reaches the set input angle that must be determined prior to testing by examining the travel limits of the pedal assembly. The controller will then generate a data file that records the input and output loads relative to the output travel, time and input angle at regular intervals.
Three additional manual cycles, per assembly, were also required in order to verify the validity of the data generated from the automatic cycles. The tester was presumed unreliable due to a faulty potentiometer. However, if one was only interested in the loads and the output travel, then the input angle, measured by the potentiometer, may be disregarded by performing a manual test that is independent of the input angle. The following is the procedure that was used for the manual testing of the variable ratio pedals on the pedal ratio tester:

1. Determine the output travel distance for the assembly.
2. Bring the input actuator to the pedal pad so that a preload of less than 2 lbf is read.
3. Zero all three readings on the controller screen (Output Travel, Input Load and Output Load).
4. Step back on the input actuator until the input load is 50 lbf.
5. *Record all 3 measurements (Output Travel, Input Load and Output Load).
6. Step back on the output actuator 3 mm to increment the output travel of the booster (the input and output loads should drop as this step is carried out).
7. Repeat steps 4-6 until the output travel reaches 32 mm.

*The input angle was actually recorded as an additional source of verification to the data generated by the tester during automatic cycling. The angles recorded did correlate with the data produced during automatic cycling.
6.0 Results & Analysis

6.1 Instantaneous Pedal Ratio vs. Booster Travel Graph

The results of all the testing are plotted on two graphs (Appendix G): One graph incorporates all the results of testing done in the automatic mode (Figure G-1) and the other displays all those that were tested manually (Figure G-2). The theoretical curve that was generated using the model was also plotted on the two graphs for comparison.

In general, the curves for the trials of each assembly are fairly consistent in the tests ran automatically. By inspection, assembly #4 had the largest difference between its two trials (~15%); the other four assembly’s trials were within 5% of each other. Assembly #3 yield the largest pedal ratio: on average, the ratio started at about 1.7 at the initial position and ended at 2.5 after roughly 30mm of booster travel. Assembly #2 ranked second in terms of highest ratio output (~1.5 – 2.1), followed by #5 (~1.5 – 2), #4 (~1.4 – 1.9) and #1 (~1.2 – 1.5). The curves of assemblies #1, 2, 3 and 4 did follow the general profile of the theoretical model developed (with the exception of assembly #5, which was unusually high for the first 6mm of its travel). The highest curve was generated by trial #1 of assembly #3 as it came within 33% of the theoretical curve. The worst curve was produced by trial #5 of assembly #1 with an error of 63%. The average curve can be approximated using trial #1 of assembly #4, which yield a percentage error of 48.

In Figure G-2, the curves for the trials of each assembly are again reasonably consistent. Assembly #3 remained at the top, in terms of generating the highest pedal ratio curve, starting this time at about 1.8 (on average) and ending at 2.3 throughout its pedal travel. Assembly #5 followed with 1.9 – 2.1, assembly #2: 1.7 – 2.2 and #4: 1.5 – 2.0. Unfortunately, a manual test was not done for assembly #1. The highest curve generated from the manual tests was trial #1, assembly #3, which managed to stay within 33% of the theoretical. With the absence of assembly #1, assembly #4 produced the worst curve with an error of 50% (trial #3). On average, the manual testing of the assemblies generated a curve that came within 40% of the theoretical and is best represented by trial #3 of assembly #3.

In summary, assembly #3 was consistently the “best” assembly, yielding results that came within 33% of the theoretical curve developed. The consistency amongst the trials of each assembly generally stayed within 5% of one another in both the automatic and manual tests performed. The order in which the assemblies ranked in terms of the highest pedal ratio curve generated also remained the same in both manual and automatic cycling (assembly #3 > assembly #2 > assembly #4). This result is based on the fact that assembly #1 was not manually tested and assembly #5 was deemed inconsistent and thus jumped around in the rankings.
6.2 Sources of Error

There are many possible sources of error that may have contributed to the discrepancy between the actual test data and the theoretical model developed. Errors could have arisen from any one of the previous four design stages (e.g. design and development, prototype, assembly or testing). The following section describes some of those sources that may have attributed to the difference between the actual results and the analytical prediction.

The prototyping and assembly of parts is one possible source of error. Although it was mentioned that all prototype parts were in fact made to specification, the assembly of these parts were not checked in terms of their relative positioning in assembly. Any deviation from the nominal position of the design will alter the geometry of the pedal and result in a different ratio plot. For example, assemblies #2, 3 and 4 required the removal of material from the bottom surface of the cam bracket profile. This implies that either the position of this profile relative to the pivot point was not consistent amongst the five assemblies (although a weld fixture was used to position the cam bracket against the housing during welding) or the location of the profile on the part, which was not checked, varied.

In the design of the pedal assembly, improvements could have been made to the fit between the rocker plate and the rocker pin. A substantial amount of slack can be seen between the washer that was spun onto the rocker pin, and the rocker pin itself. The assembly was designed to have a minimum clearance of 0.05mm and a maximum clearance of 0.45mm (Figure D-4 and D-7) in order to avoid an interference fit and minimize the friction between the parts. The actual value of this clearance is unknown and is not easily predicted since the process of riveting these assemblies was uncontrolled and the amount of deformation in the material is unknown. The result of this unknown excess slack is of course a divergence in the relative positions of the geometry from the design as the assembly is cycled, as well as un-prescribed stress concentrations in the parts.

The testing of the assemblies could also yield a number of sources of error. For example, the variable ratio model assumed that the applied force to the pedal pad stayed consistently perpendicular to the line joining the pivot point to the pedal pad centre throughout the entire travel. However, the actuator may not have been perfectly 90 degrees to the pedal pad, or it could have deviated from this angle as the pedal is pressed, if the rotation of the arm on the tester is not concentric with the rotation of the pedal arm. Furthermore, if in fact there is any variance between the axes of rotation of the pedal arm and tester arm, the distance between the point of actuation at the pad and the pivot point will vary and/or change with rotation depending on which axis (or axes) is off. Moreover, the value of $R^2$ was not measured for each assembly during testing and could have been inaccurate from the start of the test.
Finally, the length of the pushrod was indeed longer than what was modeled in the design. The pushrod that was modeled in CAD is used for the testing of current production C-segment brakes; the location of the booster pin on the variable ratio pedal, however, is different from the current production assemblies though it is fitted on the same C-segment housing. The reason why the pushrod used on the variable ratio pedals was longer was because it required an extension in order to reach the connection point at the booster pin. This will definitely have an effect on the pedal ratio as the location of the nose of the pushrod was changed. The location of the nose of the pushrod is directly related to the angle of inclination of the rod with respect to the centre line of the master cylinder. This pushrod angularity is used in the calculation of the component of the force seen through the pushrod, and thus, the output force of the assembly. Evidently, the actual ratio of the pedal will differ from the model developed since the original pushrod length was used. If the actual length of the rod was measured, the coordinates of the pushrod nose can easily be changed in the model developed to recalculate a new theoretical curve for the longer pushrod.
7.0 Remodel

7.1 New Variable Ratio Model

Although the slope of the curves from the test data do correlate well with the theoretical model developed, the error between the two is substantial (approximately 45% on average). Therefore, a new model for the variable ratio pedal was developed in an attempt to reduce this percentage error. Drawn in Figure 38 is the variable ratio pedal system with the pushrod attached. The output force, $F_{\text{out}}$, is again the reaction force exerted by the master cylinder onto the pushrod. The nose of the pushrod is modeled as a fixed slider mechanism and translates only in the horizontal direction. The housing of the assembly is also fixed as shown. The input force exerted at the pedal pad by the driver is denoted as $F_{\text{in}}$.

![Figure 38: Variable ratio pedal system](image)
To derive an expression for the instantaneous pedal ratio, the pedal system above is broken down into three components: the pushrod, rocker plate and pedal arm. Figures 39 and 40 show a set of free-body diagrams for the variable ratio system. The cam bracket and housing are treated as one subassembly (labeled as body 1) and can be regarded as a fixed wall. The pedal arm is labeled as body 2, and the rocker plate as body 3. The following four assumptions were made for this new model:

1. Friction is neglect for the slider mechanism.

2. Although the actual nose of the pushrod is free to rotate in all directions, it is treated as a 2D problem and is assumed to rotate only about the axis perpendicular to the paper.

3. The moulded roller assembly is assumed to ‘roll’ against the top surface of the cam bracket profile (i.e. there is no tangential force on the roller as friction is neglected). Therefore, the only force exerted on the roller is a force normal to the surface of the bracket whose line of action passes through COC at all times.

4. The input force is assumed perpendicular to the line joining the pedal pad and the pivot point (R2) at all instances.

There are two cases presented in this new variable ratio model: The first case (Figures 39a and 39b) represents all points in time when CCP is to the right of COC in terms of Cartesian coordinates. The second case (Figures 40a and 40b) represents all instances when CCP is to the left of COC. There are also four new angles introduced in this new model: $\theta_2$, $\varphi$, $\zeta$ and $\rho$. In each case, $\theta_2$, $\varphi$ and $\zeta$ are defined with respect to the horizontal and $\rho$ with respect to the vertical. A force triangle is also drawn for each of the four figures including all necessary angles to solve the system. Each force triangle represents a static equation where the sum of all forces in the triangle is equal to zero. Note that all forces in these figures are drawn to scale relative to one another. The quantity $F_{23}$ represents the force of body 2 on 3, and $F_{13}$ represents the force of body 1 on 3. The force $F_{23}$ is also equal in magnitude but opposite in direction to $F_{32}$. Figures 39b and 40b show the individual free-body diagrams of the rocker plate and pedal arm merged into one. The red arrows represent the forces on the rocker plate and the blue arrows signify the forces on the pedal arm. The rocker plate in Figures 39b and 40b have two known lines of action: $F_m$ and $F_{13}$. In a three-force member, the line of actions of all three forces will always meet at one point. Hence, the line of action of $F_{23}$ crosses the intercept of the two known lines of action of $F_m$ and $F_{13}$. Having solved for the line of action of $F_{23}$, the pedal arm then assumes two known lines of action: $F_{in}$ and $F_{32}$. As a result, the line of action of $F_{12}$ is then revealed as it crosses the intercept of these two known lines of action.
Case 1: CCP_x > COC_x

Figure 39a: Free-body diagram of pushrod (case 1)

Figure 39b: Combined individual free-body diagrams of rocker plate and pedal arm (case 1)
Recalling that the instantaneous pedal ratio is defined as the ratio of the output force exerted by the pushrod into the master cylinder, to the input force at the pedal pad:

\[
\text{Instantaneous pedal ratio} = \frac{F_{\text{out}}}{F_{\text{in}}} \quad (7.1.1)
\]

From Figure 39a,

\[
F_{\text{out}} = F_{\text{in}} \cdot \cos \alpha \quad (7.1.2)
\]

Using the sine law on the rocker plate force triangle in Figure 39b yields:

\[
\frac{F_m}{\sin(90^\circ - \rho - \varphi)} = \frac{F_{13}}{\sin(\varphi - \alpha)} = \frac{F_{23}}{\sin(90^\circ + \alpha + \rho)} \quad (7.1.3)
\]

\[
\Rightarrow F_{23} = F_m \cdot \frac{\sin(90^\circ + \alpha + \rho)}{\sin(90^\circ - \rho - \varphi)} \quad (7.1.4)
\]

Using the sine law on the pedal arm force triangle in Figure 39b yields:

\[
\frac{F_{\text{in}}}{\sin(\zeta - \varphi)} = \frac{F_{12}}{\sin(90^\circ + \varphi - \theta_2)} = \frac{F_{32}}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.5)
\]

\[
\Rightarrow F_{\text{in}} = F_{32} \cdot \frac{\sin(\zeta - \varphi)}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.6)
\]

Since,

\[
F_{23} = F_{32} \quad (7.1.7)
\]

Combining (7.1.7) and (7.1.4),

\[
F_{32} = F_{23} = F_m \cdot \frac{\sin(90^\circ + \alpha + \rho)}{\sin(90^\circ - \rho - \varphi)} \quad (7.1.8)
\]

Now, substituting (7.1.8) into (7.1.6),

\[
F_{\text{in}} = F_m \cdot \frac{\sin(90^\circ + \alpha + \rho)}{\sin(90^\circ - \rho - \varphi)} \cdot \frac{\sin(\zeta - \varphi)}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.9)
\]

Finally, substituting (7.1.9) & (7.1.2) into (7.1.1),

\[
\text{Instantaneous pedal ratio} = \frac{\cos \alpha \cdot \sin(90^\circ - \rho - \varphi) \cdot \sin(90^\circ - \zeta + \theta_2)}{\sin(90^\circ + \alpha + \rho) \cdot \sin(\zeta - \varphi)} \quad (7.1.10)
\]
Case 2: \( \text{CCP}_x < \text{COC}_x \)

Figure 40a: Free-body diagram of pushrod (case 2)

Figure 40b: Combined individual free-body diagrams of rocker plate and pedal arm (case 2)
Again, the instantaneous pedal ratio is defined as the ratio of the output force exerted by the pushrod into the master cylinder, to the input force at the pedal pad (Equation 7.1.1):

From Figure 40a,

\[ F_{out} = F_m \cdot \cos \alpha \quad (7.1.11) \]

Using the sine law on the rocker plate force triangle in Figure 40b yields:

\[ \frac{F_m}{\sin(90^\circ + \rho - \varphi)} = \frac{F_{13}}{\sin(\varphi - \alpha)} = \frac{F_{23}}{\sin(90^\circ + \alpha - \rho)} \quad (7.1.12) \]

\[ \Rightarrow F_{23} = F_m \cdot \frac{\sin(90^\circ + \alpha - \rho)}{\sin(90^\circ + \rho - \varphi)} \quad (7.1.13) \]

Using the sine law on the pedal arm force triangle in Figure 40b yields:

\[ \frac{F_{in}}{\sin(\zeta - \varphi)} = \frac{F_{12}}{\sin(90^\circ + \varphi - \theta_2)} = \frac{F_{32}}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.14) \]

\[ \Rightarrow F_{in} = F_{32} \cdot \frac{\sin(\zeta - \varphi)}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.15) \]

Since,

\[ F_{23} = F_{32} \quad (7.1.16) \]

Combining (7.1.16) and (7.1.13),

\[ F_{32} = F_{23} = F_m \cdot \frac{\sin(90^\circ + \alpha - \rho)}{\sin(90^\circ + \rho - \varphi)} \quad (7.1.17) \]

Now, substituting (7.1.17) into (7.1.15),

\[ F_{in} = F_m \cdot \frac{\sin(90^\circ + \alpha - \rho)}{\sin(90^\circ + \rho - \varphi)} \cdot \frac{\sin(\zeta - \varphi)}{\sin(90^\circ - \zeta + \theta_2)} \quad (7.1.18) \]

Finally, substituting (7.1.18) & (7.1.11) into (7.1.1),

\[ \text{Instantaneous pedal ratio} = \frac{\cos \alpha \cdot \sin(90^\circ + \rho - \varphi) \cdot \sin(90^\circ - \zeta + \theta_2)}{\sin(90^\circ + \alpha - \rho) \cdot \sin(90^\circ - \zeta + \varphi)} \quad (7.1.19) \]
7.1.1 Solving for the intercept $T_1$

Recall that the rocker plate in Figures 39b and 40b have two known lines of action: $F_m$ and $F_{13}$. To find the intercept of these lines ($T_1$), a two-point formula for the equation of a straight line is used to define the line of action for each of the two forces. The following derivation shows how the intercept point, $T_1$, is calculated in the new variable pedal ratio model shown in Appendix H (Figure H-1).

Line of action of $F_m$:
\[
\frac{y - P N_y}{B P_y - P N_y} = \frac{x - P N_x}{B P_x - P N_x}
\]
\[
y = \frac{B P_y - P N_y}{B P_x - P N_x} \cdot (x - P N_x) + P N_y
\]
\[(7.1.1.20)\]

Line of action of $F_{13}$:
\[
\frac{y - C C P_y}{C O C_y - C C P_y} = \frac{x - C C P_x}{C O C_x - C C P_x}
\]
\[
x = \frac{C O C_x - C C P_x}{C O C_y - C C P_y} \cdot (y - C C P_y) + C C P_x
\]
\[(7.1.1.21)\]

Substituting (7.1.1.21) into (7.1.1.20),
\[
y = \frac{B P_y - P N_y}{B P_x - P N_x} \cdot \left\{ \frac{C O C_x - C C P_x}{C O C_y - C C P_y} \cdot (y - C C P_y) + C C P_x \right\} - P N_x + P N_y
\]
\[(7.1.1.22)\]

Let $m_1 = \frac{B P_y - P N_y}{B P_x - P N_x}$
\[(7.1.1.23)\]

and $m_2 = \frac{C O C_y - C C P_y}{C O C_x - C C P_x}$
\[(7.1.1.24)\]

From (7.1.1.22),
\[
ym_1 m_2 y = m_1 (-m_2 C C P_y + C C P_x - P N_x) + P N_y
\]
\[
y = \frac{m_1 (-m_2 C C P_y + C C P_x - P N_x) + P N_y}{1 - m_1 m_2}
\]
\[(7.1.1.25)\]

Equation 7.1.1.21 is a function of $y$. By calculating Equation 7.1.1.25, both the $x$ and $y$ coordinates of $T_1$ are then solved simultaneously throughout the entire travel of the pedal. As a result, the line of action of $F_{23}$ is then known as it crosses this intercept, $T_1$, and $RP$. 


7.1.2 Solving for the intercept $T_2$

As the line of action of $F_{23}$ is now known, the pedal arm now possesses two known lines of action: $F_{in}$ and $F_{32}$, which is the same as $F_{23}$. Since the line of action of $F_{32}$ has two known points, $T_1$ and $RP$, the two-point formula for the equation of a line is used once more. Only one known point (PAD) is known for the line of action of $F_{in}$. The slope of $F_{in}$, however, is known since it is perpendicular to the line joining the pivot point and pedal pad. Therefore, a point-slope formula for the equation of a line is used, with the slope of $F_{in}$ being the negative reciprocal of $R2$. The following derivation shows how the intercept point, $T_2$, is calculated.

Line of action of $F_{32}$:

$$\frac{y - T_{1y}}{RP_y - T_{1y}} = \frac{x - T_{1x}}{RP_x - T_{1x}}$$

$$y = \frac{RP_y - T_{1y}}{RP_x - T_{1x}}(x - T_{1x}) + T_{1y} \quad (7.1.2.26)$$

Line of action of $F_{in}$:

$$y = PAD_y - \left(\frac{PP_y - PAD_y}{PP_x - PAD_x}\right)(x - PAD_x) \quad (7.1.2.27)$$

Let $m_3 = \frac{RP_y - T_{1y}}{RP_x - T_{1x}} \quad (7.1.2.28)$

and $m_4 = \frac{PP_y - PAD_y}{PP_x - PAD_x} \quad (7.1.2.29)$

Comparing $(7.1.2.26)$ to $(7.1.2.27)$, and substituting in $(7.1.2.28)$ and $(7.1.2.29)$,

$$m_3(x - T_{1x}) + T_{1y} = PAD_y - (m_4)(x - PAD_x)$$

$$x(m_3 + m_4) = m_3 T_{1x} - T_{1y} + PAD_y - m_4 \cdot PAD_x$$

$$x = \frac{m_3 T_{1x} - T_{1y} + PAD_y - m_4 \cdot PAD_x}{m_3 + m_4} \quad (7.1.2.30)$$

By calculating the $x$ coordinate from Equation 7.1.2.30, and solving for Equation 7.1.2.27, which is a function of $x$, the Cartesian coordinates of $T_2$ are determined and thus, the line of action of $F_{12}$ is known since it passes through $T_2$ and the pivot of the pedal.
7.2 The introduction of 4 new angles: $\theta_2$, $\varphi$, $\zeta$ and $\rho$

7.2.1 $\theta_2$

Figures 39b and 40b show $\theta_2$ being defined as the angle between the horizontal and $R2$ (the line connecting $PP$ to $PAD$). The angle is defined at the horizontal in the clockwise direction.

7.2.2 $\varphi$

$\varphi$ is defined in Figures 39b and 40b as the acute angle between the force at the rocker pin ($F_{23}$ or $F_{32}$) and the horizontal. $\varphi$ can not be solved unless the line of action of $F_{23}$ is known (i.e. intercept $T_1$ must first be solved).

7.2.3 $\zeta$

$\zeta$ is defined in Figures 39b and 40b as the acute angle between the line of action of $F_{12}$ and the horizontal. $\zeta$ can not be solved until the line of action of $F_{12}$ is known (i.e. intercept $T_2$ must first be solved).

7.2.4 $\rho$

$\rho$ is defined in Figure 39b as the acute angle between the line of action of $F_{13}$ and the vertical. It is defined at the vertical in the counterclockwise direction. In Figure 40b, $\rho$ is also the acute angle between $F_{13}$ and the vertical but is defined in the clockwise direction from the vertical.

7.3 Results of the new model

Figures H-2 and H-3 again show the plot of the instantaneous pedal ratio versus booster travel of the test results. An additional curve generated by the new model was also added to these graphs. The new curve shows that the pedal should theoretically start with a ratio of 2.25 and end at 2.95 after 30.21 mm of travel. The slope of this curve is similar to the original model's but is much improved in terms of the percentage error. On average, there is an error of 32% from the automatic trials and only 20% from the manual tests. This is a significant improvement of 16 – 20%, depending on the type of test. In comparison to the original curve, an improvement of 23% in the error was obtained. Note that these comparisons can be made because of the slope of these curves were successfully predicted in both models.
8.0 Conclusion

The purpose of this thesis project was to develop a variable ratio brake pedal for future potential programs at Flex-N-Gate. The main objective of the project was to develop a model for a given variable ratio pedal design and to be able to use this model to design to OEM specifications. A model was created for a particular variable ratio design using Microsoft Excel. The spreadsheet contained input variables for the critical points of the variable ratio pedal and provided a number of outputs including the pedal travel, the path of the critical points and most importantly, the instantaneous pedal ratio.

In order to verify and validate the model, five prototype assemblies were made and tested on Flex-N-Gate’s Pedal Ratio Tester. The results of the tests showed that the slope of the curves did correlate well with the theoretical model although there was an error of 40 – 48% in comparison. In an attempt to improve upon this error, a new model was developed using force triangles on three-force member bodies. This new model resulted in a theoretical curve that also matched well with the slope of the test data but yield only 20 – 32% error. The error in this new model is arguably reasonable considering all the possible sources of errors that may have occurred due to manufacturing, design or testing.

The model developed for this variable ratio brake pedal design can now be used to aid the engineers and designers at Flex-N-Gate to design the pedal to OEM specifications, predict the path of the assembly and most importantly, calculate the instantaneous pedal ratio (with a 20 – 32% error). For the future, Flex-N-Gate may choose to produce a second build to revalidate this model, taking into account the previous sources of error. As a continuation of this thesis, a future project may focus on a feasibility study of the manufacturing of this pedal in high volumes. This would of course include a complete cost analysis, materials selection and process design.
Appendix A
### Appendix A: Project Timeline

**Figure A-1: Project Timeline**

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<td>Verification/Validation &amp; Review</td>
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Appendix B
### Appendix B: Variable Ratio Pedal Model

#### Inputs
- Pedal pivot to rocker pivot: 50.00 mm
- rocker pivot to pushrod nose: 69.13 mm

#### Deflections
- Rocker: 35.34 in.
- Pushrod: 2.30 in.
- Pushrod (Nose): 0.98 in.
- Pushrod: 47.95 in.
- Rocker pivot: 13.06 in.
- Pedal pivot: 27.43 in.

#### Outputs
- Foot pedal: 8.43 in.
- Lever arm: 3.90 in.
- Lever arm: 0.83 in.
- Lever arm: 3.68 in.
- Lever arm: 6.60 in.
- Lever arm: 0.76 in.
- Lever arm: 0.38 in.
- Lever arm: 0.09 in.
- Lever arm: 0.08 in.

#### Table

<table>
<thead>
<tr>
<th>Pushrod Instant. Pedal Ratio</th>
<th>Pushrod Instant. Lever arm</th>
<th>Pedal Instant. Lever arm</th>
<th>Lever arm</th>
<th>Instant. Cam Contact</th>
<th>Instant. Cam Contact</th>
<th>Instant. Cam Contact</th>
<th>Instant. Cam Contact</th>
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#### Instantaneous Pedal Ratio
- The output force generated at the pushrod nose along the m/cyl C/L divided by the input force at, and perpendicular to, pedal pad.

#### Instantaneous Pushrod Stroke
- The output stroke generated at the pushrod nose along the m/cyl C/L.

#### Pushrod Stroke
- Angle of pushrod strut relative to line through pedal pivot parallel to m/cyl C/G.

#### Pushrod (Nose) (to CCP)
- Angle between m/cyl C/L and pushrod strut C/L.

#### Pushrod (Nose) (to CCP)
- Distance from RP to CCP/pushrod connecting line parallel to (m/cyl C/L).

#### Pushrod (Nose) (to CCP)
- Distance from RP to the centre of the pedal pad.

#### Perpendicular distance from pedal pivot to line through m/cyl C/L.

#### PL
- Length along pushrod strut C/L from the nose of the pushrod (PN) to the eye (JP).
Appendix C: Prototype Request Correspondences

John Nicol

From: Joe Mok
Sent: Thursday, August 31, 2006 8:58 AM
To: John Nicol
Cc: Andrew Power; Nick Porco
Subject: RE: Prototype Request - Variable Ratio Pedal

John,

I've attached a cadkey file of the entire assembly including the housing; the cam bracket will need to be welded to this housing.

Also attached are the prints for all the required parts; there are some differences in dimensions between the math data and the prints so please use the prints as masters for pin and hole diameters and tolerances.

The third attachment is a print of the roller cam. This is a production part for the hinge and will need to modified at a later date, hence, you will not need to machine this part from scratch.

When these parts are complete, please return them to Nick Porco so that I can be present for the actual assembly process.

Thanks,
Joe

-----Original Message-----
From: Joe Mok
Sent: Monday, August 28, 2006 1:41 PM
To: Andrew Power
Cc: Steven Dembo; Nick Porco
Subject: Prototype Request - Variable Ratio Pedal

Andrew,

As for Joe, 314006 needs comp. only do not spin
work to marked up prints
Joe to follow up with weld feasibility

Attached is a prototype request and a model of the updated cam bracket (205419-0205-0.x.t). Note that the cam in the previous assembly model (205419-0200-0.x.t) is obsolete.

<< File: Prototype Sample Request - 8-28-06.doc >> << File: 205419-0205-0.x.t >>

Thanks,

Joe Mok

Product Engineering
FLEX|N|GATE (Ventra Group Co.)
Product Development Centre
(905) 778 9900 ext. 769
(905) 778 9509 fax
joe.mok@ventra.com
Appendix D
Appendix D: Marked-up Prototype Engineering Prints

Figure D-1: Pedal Arm drawing

Figure D-2: Cam Bracket drawing
Figure D-3: Cam Pin drawing

Figure D-4: Rocker Pin drawing
Figure D-5: Booster Pin drawing

Figure D-6: Moulded Roller Assembly drawing
Figure D-7: Rocker Plate
**Appendix E**: Prototype Variable Ratio Brake Pedal Assembly Bill of Materials (BOM)

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Appendix F
Appendix F: Pivot Tube drawing with weld instructions
Appendix G
Variable Ratio C-Segment Modified Prototype Brake Pedal

Figure G-1: Plot of all the results with the theoretical curve (automatic mode)

Note: All negative value data have been omitted from this graph (e.g. negative travel and negative loads).
Figure G-2: Plot of all the results with the theoretical curve (manual mode)
Appendix H
Variable Ratio C-Segment Modified Prototype Brake Pedal

Figure H-2: Plot of all the results with the new theoretical curve (automatic mode)

Note: All negative value data have been omitted from this graph (e.g. negative travel and negative loads).
Figure H-3: Plot of all the results with the new theoretical curve (manual mode)
Appendix I
Appendix I: FMVSS 135 – section 7.11

§ 571.135

(b) Stopping distance for reduced test speed: \( S \leq 0.10V + 0.0158V^2 \).

§7.11. Brake power unit or brake power assist unit inoperative (System depleted).

§7.11.1. General information. This test is for vehicles equipped with one or more brake power units or brake power assist units.

§7.11.2. Vehicle conditions. (a) Vehicle load: GVWR only.

(b) Transmission position: In neutral.

(c) Test conditions and procedures.

(a) IBT: \( \leq 65 \, ^\circ C \) (149 \, ^\circ F), \( \leq 100 \, ^\circ C \) (212 \, ^\circ F).

(b) Test speed: 100 \, km/h (62.1 \, mph).

(c) Pedal force: \( \leq 65 \, N \) (14.6 \, lbs), \( \leq 500 \, N \) (112.4 \, lbs).

(d) Wheel lockup: No lockup of any wheel for longer than 0.1 seconds allowed at speeds greater than 15 \, km/h (9.3 \, mph).

(e) Number of runs: 6 stops.

(f) Test surface: PFC of 0.9.

(g) Disconnect the primary source of power for one brake power assist unit or brake power unit, or one of the brake power unit or brake power assist unit subsystems if two or more subsystems are provided.

(h) If the brake power unit or power assist unit operates in conjunction with a backup system and the backup system is automatically activated in the event of a primary power service failure, the backup system is operative during this test.

(j) Exhaust any residual brake power reserve capability of the disconnected system.

(k) Make each of the 6 stops by a continuous application of the service brake control.

(l) Restore the system to normal at completion of this test.

(l) For vehicles equipped with more than one brake power unit or brake power assist unit, conduct tests for each in turn.

(m) For vehicles with electrically-actuated service brakes (brake power unit), this test is conducted with any single electrical failure in the electrically-actuated service brakes instead of a failure of any other brake power or brake power assist unit, and all other systems intact.

§7.11.4. Performance requirements. The service brakes on a vehicle equipped with one or more brake power assist units or brake power units, with one such unit inoperative and depleted of all reserve capability, shall stop the vehicle as specified in §7.11.4(a) or §7.11.4(b).

(a) Stopping distance from 100 \, km/h test speed: \( \leq 138 \, m \) (515 \, ft).

(b) Stopping distance for reduced test speed: \( S \leq 0.10V + 0.0158V^2 \).


§7.12.1. Vehicle conditions. (a) Vehicle load: GVWR only.

(b) Transmission position: In neutral.

(c) Parking brake burnish: (1) For vehicles with parking brake systems not utilizing the service friction elements, the friction elements of such a system are burnished prior to the parking brake test according to the published recommendations furnished to the purchaser by the manufacturer.

(2) If no recommendations are furnished, the vehicle's parking brake system is tested in an unburnished condition.

(d) Parking brake applications: 1 application and up to 2 reapplications, if necessary.

§7.12.2. Test conditions and procedures.

(a) IBT:

(1) Parking brake systems utilizing service brake friction materials shall be tested with the IBT \( \leq 100 \, ^\circ C \) (212 \, ^\circ F) and shall have no additional burnishing or artificial heating prior to the start of the parking brake test.

(2) Parking brake systems utilizing non-service brake friction materials shall be tested with the friction materials at ambient temperature at the start of the test. The friction materials shall have no additional burnishing or artificial heating prior to or during the parking brake test.

(b) Parking brake control force: Hand control \( \leq 400 \, N \) (88.9 \, lbs); foot control \( \leq 500 \, N \) (112.4 \, lbs).

(c) Hand force measurement locations: The force required for actuation of a hand-operated brake system is measured at the center of the hand grip area or at a distance of 40 \, mm (1.57 \, in) from the end of the actuation lever as illustrated in Figure 8.

(d) Parking brake applications: 1 application and up to 2 reapplications, if necessary.

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