Penn Electric Racing Mechanical Challenge: Brake Pedal Design

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Design Requirements

FSAE Important and Relevant Rules to Braking System

I will be referencing the following rules frequently in justifications for my design choices.

Rule T.3.1.2
Brake system must act on all four wheels, be operated by a single control, and be capable of locking all four wheels.

Rule T.3.1.3
Brake system must have two independent hydraulic circuits, and a leak or failure anywhere must still maintain effective braking on at least two wheels.

Rule T.3.1.4
Each hydraulic circuit must have its own separate fluid reservoir, or it can also have an internally separated original equipment style reservoir.

Rule T.3.1.6
Brake by wire systems are prohibited; this means the pedal must be mechanically and hydraulically linked to the calipers, not electronically.

Rule T.3.1.7
Unarmored plastic brake lines are prohibited.

Rule T.3.1.9
Any part of the brake system mounted on the sprung side of the car must not hang below the lowest surface of the chassis, when viewed from the side.

Rule T.3.1.10, requirements detailed in Rule T.8.2
Every fastener in the brake system is classified as a Critical Fastener, which pulls in the

fastener requirements below.

Rule T.3.2.1

The brake pedal must be either fabricated from steel or aluminum, or machined from steel, aluminum, or titanium.

Rule T.3.2.2

The brake pedal and everything supporting it, including the pedal box, chassis mounting, and any adjustment mechanism, must survive a two thousand Newton force with zero failure. *Clarification: this exact number is not enforceable, since no human can realistically apply 2000 N through a single leg press against a pedal (see the discussion in Performance Targets). A more realistic ceiling of 1500 N will be used in the actual structural design load throughout this document.*

Rule T.3.2.3

Failure of any non load bearing component in the brake system or pedal box must not interfere with pedal operation.

Rule T.3.2.4 (EV only)

The first 90% of pedal travel *may* be used to regenerate energy without actuating the hydraulic brake system; the remaining pedal travel *must* directly operate the hydraulic brake system. Brake energy regeneration may stay active. Note that 90% is a permissive ceiling, not a mandatory allocation. *This rule does not constrain the present design, since regenerative braking is not implemented through brake pedal travel — see the Total Pedal Travel subsection below.*

Rule T.3.3.1 through Rule T.3.3.5

A Brake Over Travel Switch is required, must be a purely mechanical analog single pole single throw switch, must not be resettable by the driver, and must open the car's shutdown circuit when tripped.

Rule T.4.3.1 and Rule EV.4.6

A Brake System Encoder is required, meaning a sensor or switch that measures brake pedal position or brake system pressure.

Rule T.3.4.1 and Rule T.3.4.2

A brake light is required, must be red, a minimum size, and mounted in a specified location range.

Rule T.8.2

Every brake system fastener must meet at minimum Society of Automotive Engineers Grade five or Metric Class eight point eight, must be hex head or socket head cap screw only, must not be countersunk, must use a visible positive locking mechanism, and must have at least two full threads projecting past any lock nut.

Rule IN.14.3
During the dynamic brake test, the system must demonstrate full lockup of all four wheels with the tractive system switched off.

Rule EV.4.7 and Rule EV.7.7 (note for awareness only)
The Brake Pedal Plausibility Check and the Brake System Plausibility Device both rely on the Brake System Encoder signal to cut motor power during simultaneous acceleration and braking.

Performance Targets

The design must provide at least 1200 psi to the front brake lines and at least 400 psi to the rear brake lines. The following derivations are going to use these targets along with the vehicle's physics to calculate additional numbers the pedal design must hit including the pedal ratio, master cylinder bore size, pedal travel, pushrod force, and brake bias adjustment range.

Assumed design inputs

The following values are assumptions I made based off of a typical FSAE electric vehicle which I use in my derivations. Each is justified individually and all of them are used consistently throughout the numeric examples below. All variables are defined at the place of their first use in the derivations.

- Total mass with driver, $m = 260$ kg
- Static front weight fraction, 48% front / 52% rear: typical for an EV with rear-mounted motor and battery mass, giving $L_r = 0.744$ m, $L_f = 0.806$ m for a wheelbase $L = 1.55$ m.
- Center of gravity height, $h = 0.30$ m: estimate for a low to ground FSAE chassis.
- Tire-road friction coefficient, $\mu_{\text{tire}} = 1.5$: estimate for FSAE racing slicks.
- Target deceleration, $a_x = 1.5g = 14.715$ m/s^2: since a car cannot decelerate faster than $\mu_{\text{tire}} \times g$.
- Tire rolling radius, $r_{\text{tire}} = 0.254$ m (10 in): common FSAE wheel/tire package.

- Caliper piston area, $A_{\text{piston}} = 1.5\ \text{in}^2$ per caliper (one side): a typical small FSAE two-piston caliper.
- Pad friction coefficient, $\mu_{\text{pad}} = 0.45$: for a semi-metallic brake pad.
- Effective rotor radius, $R_{\text{eff}} = 4$ in: common size for an FSAE rotor.
- Comfortable maximum foot force (design target), $F_{\text{foot,design}} = 70$ lbf.
- Pedal ratio (design choice), $PR = 6.0$
- Pad knock-back clearance, $c_{kb} = 0.010$ in per pad.
- Number of calipers per circuit: 2 (one per wheel on that axle), giving a total circuit piston area of $3.0\ \text{in}^2$ for the stroke calculation.
- Structural design load, $F_{\text{foot,structural}} = 1500$ N (see Rule T.3.2.2 note above).

Total vehicle weight

Newton's second law, $F = ma$, applied to gravity in place of a general acceleration:

$$W = mg \tag{1}$$

where W is total vehicle weight, m is total mass (car plus driver), and g is gravitational acceleration ($9.81\ \text{m/s}^2$).

Design Calculation: $W = (260)(9.81) = 2550.6$ N.

Static weight per axle

For the following calculations I am placing the origin at the contact point of the rear wheel to make finding the moment easier. The front contact point is a horizontal distance L away and the center of gravity a horizontal distance L_r away, both at ground level for the contact points. With zero deceleration, only the weight and the front normal force produce moments about the origin (both act vertically, so each moment equals force times horizontal distance):

$$\begin{aligned}
\sum M_{\text{rear}} &= 0 \\
L\,W_{f,\text{static}} - W L_r &= 0 \\
L\,W_{f,\text{static}} &= W L_r
\end{aligned}$$

Dividing both sides by L isolates $W_{f,\text{static}}$:

$$W_{f,\text{static}} = \frac{WL_r}{L}$$

The rear static weight follows from vertical force equilibrium, $W_{f,\text{static}} + W_{r,\text{static}} = W$:

$$W_{r,\text{static}} = W - \frac{WL_r}{L} = \frac{W(L-L_r)}{L} = \frac{WL_f}{L}, \quad \text{since } L - L_r = L_f$$

$$W_{f,\text{static}} = \frac{WL_r}{L}, \qquad W_{r,\text{static}} = \frac{WL_f}{L} \tag{2}$$

where $W_{r,\text{static}}$ is the static rear axle weight, $W_{f,\text{static}}$ is the static front axle weight, L_r is the horizontal distance from the rear axle to the center of gravity, L_f is the horizontal distance from the front axle to the center of gravity, and $L = L_r + L_f$ is the wheelbase.

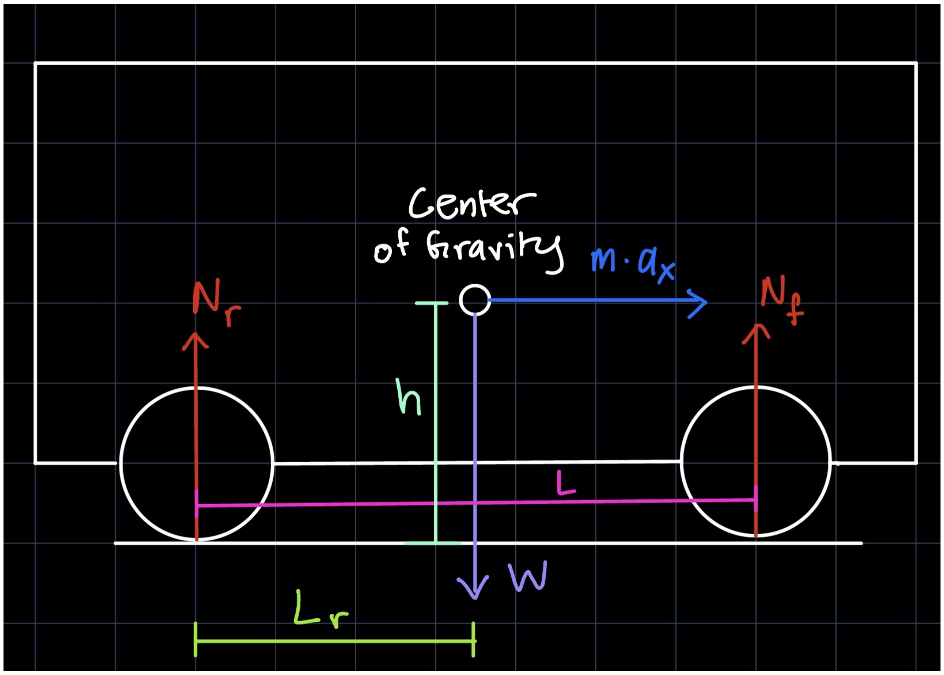

Figure 1: Free body diagram of the vehicle, used for both the static weight and weight transfer derivations.

Design Calculation: Using $L_r = 0.744$ m, $L_f = 0.806$ m, $L = 1.55$ m: $W_{f,\text{static}} = 2550.6 \times \frac{0.744}{1.55} = 1224.3$ N, and $W_{r,\text{static}} = 2550.6 \times \frac{0.806}{1.55} = 1326.3$ N (check: $1224.3 + 1326.3 = 2550.6$ N $= W$).

Weight transfer during braking

See Figure 1; the same diagram applies with $a_x > 0$.

According to D'Alembert's principle braking is modeled as a fictitious forward-pointing inertial force ma_x acting at the center of gravity, which lets the decelerating system be solved with static equilibrium. Using the general 2D moment formula for a force at point (x, y) with components (F_x, F_y) about the origin, $M = xF_y - yF_x$, the moment contributed by each force acting on the car is:

$$
\begin{aligned}
M_{\text{weight}} &= L_r(-W) - h(0) = -WL_r \\
M_{\text{inertial}} &= L_r(0) - h(ma_x) = -h\,ma_x \\
M_{N_f} &= L(N_f) - 0(0) = LN_f \\
M_{\text{friction}} &= x(0) - 0(F_x) = 0 \quad \text{(both terms vanish since friction acts at } y = 0 \text{ and is horizontal)}
\end{aligned}
$$

Since this is static we can sum all moments to zero and solving for N_f:

$$
\begin{aligned}
-WL_r - h\,ma_x + LN_f = 0 \\
LN_f &= WL_r + h\,ma_x \\
N_f &= \frac{WL_r}{L} + \frac{ma_x h}{L}
\end{aligned}
$$

Subtracting the static result from Equation 2, $N_f - W_{f,\text{static}} = N_f - \frac{WL_r}{L}$, isolates the change in weight during braking:

$$
\Delta W = \frac{ma_x h}{L} \tag{3}
$$

where a_x is the target deceleration and h is the center of gravity height above the ground.

Design Calculation: $\Delta W = \dfrac{(260)(14.715)(0.30)}{1.55} = 740.5$ N.

Dynamic axle loads

Since the wheels are always in contact with the ground there is always a vertical force equilibrium where the sum of both normal forces have to equal total weight at every instant, regardless if we are braking or not:

$$
N_f + N_r = W \quad \Rightarrow \quad N_r = W - N_f
$$

Substituting $N_f = W_{f,\text{static}} + \Delta W$ from Equation 3:

$$\begin{aligned} N_r &= W - (W_{f,\text{static}} + \Delta W) \\ &= (W - W_{f,\text{static}}) - \Delta W \\ &= W_{r,\text{static}} - \Delta W, \quad \text{since } W - W_{f,\text{static}} = W_{r,\text{static}} \end{aligned}$$

$$W_{f,\text{dyn}} = W_{f,\text{static}} + \Delta W, \qquad W_{r,\text{dyn}} = W_{r,\text{static}} - \Delta W \tag{4}$$

Design Calculation: $W_{f,\text{dyn}} = 1224.3 + 740.5 = 1964.8$ N, and $W_{r,\text{dyn}} = 1326.3 - 740.5 = 585.8$ N (check: $1964.8 + 585.8 = 2550.6$ N $= W$).

Maximum tire force before lockup

Friction force is always the friction coefficient multiplied by the normal force, $F_{\text{friction,max}} = \mu N$. In this case we can substitute the dynamic axle loads from Equation 4 as the normal load at each axle:

$$F_{f,\text{tire}} = \mu_{\text{tire}} W_{f,\text{dyn}}, \qquad F_{r,\text{tire}} = \mu_{\text{tire}} W_{r,\text{dyn}} \tag{5}$$

where μ_{tire} is the tire-to-road friction coefficient.

Design Calculation: $F_{f,\text{tire}} = (1.5)(1964.8) = 2947.2$ N, and $F_{r,\text{tire}} = (1.5)(585.8) = 878.7$ N.

Required braking torque per axle

Torque is force times the perpendicular distance from the axis of rotation. In this case the tire force acts at the ground contact point, a distance r_{tire} (the tire's rolling radius) from the wheel's center of rotation:

$$T = F \times r_{\text{tire}}$$

Substituting the tire forces from Equation 5:

$$T_f = F_{f,\text{tire}} r_{\text{tire}}, \qquad T_r = F_{r,\text{tire}} r_{\text{tire}} \tag{6}$$

where r_{tire} is the tire rolling radius. T_f and T_r represent the total torque demand across both wheels on the front and rear axles respectively.

Design Calculation: $T_f = (2947.2)(0.254) = 748.6$ N·m, and $T_r = (878.7)(0.254) = 223.2$ N·m.

Required hydraulic pressure: used to validate the 1200/400 psi requirement

Hydraulic pressure on a piston of area A_{piston} produces a clamping force, which is defined as the following:

$$F_{\text{clamp}} = PA_{\text{piston}}$$

The brake pad coming into contact with the rotor causes a friction (with pad friction coefficient μ_{pad}) that converts this clamping force into a tangential force resisting rotor rotation:

$$F_{\text{tangential}} = \mu_{\text{pad}} \, F_{\text{clamp}} = \mu_{\text{pad}} PA_{\text{piston}}$$

This tangential force acts at the effective rotor radius R_{eff} and produces a torque at a single caliper. Because a caliper clamps from both sides simultaneously, a single caliper's torque is always doubled:

$$T_{\text{one caliper}} = 2 \, F_{\text{tangential}} \, R_{\text{eff}} = 2 \, \mu_{\text{pad}} \, P \, A_{\text{piston}} \, R_{\text{eff}}$$

Since T_f and T_r (Equation 6) represent the combined torque demand from *both* wheels on that axle, and each wheel has its own caliper, the total torque available from the axle is n_{calipers} times a single caliper's torque, where $n_{\text{calipers}} = 2$ in this case:

$$T = n_{\text{calipers}} \, T_{\text{one caliper}} = 2 \, \mu_{\text{pad}} \, P \, (n_{\text{calipers}} A_{\text{piston}}) \, R_{\text{eff}} = 2 \, \mu_{\text{pad}} \, P \, A_{\text{piston,total}} \, R_{\text{eff}}$$

where $A_{\text{piston,total}} = n_{\text{calipers}} A_{\text{piston}}$ is the combined piston area across both calipers on the axle. Solving for pressure:

$$P = \frac{T}{2 \, \mu_{\text{pad}} \, A_{\text{piston,total}} \, R_{\text{eff}}} \tag{7}$$

where T is required axle torque (Equation 6), $A_{\text{piston,total}}$ is the combined caliper piston area across both wheels on that axle, and R_{eff} is the effective rotor radius. In order to validate the 1200/400 psi minimum performance targets we can solve for the pressure and compare the results.

Design Calculation: Converting torque to imperial units (1 N·m = 8.8507 lbf·in) to match the psi target: $T_f = 6626$ lbf·in, $T_r = 1976$ lbf·in. With $\mu_{\text{pad}} = 0.45$, $A_{\text{piston,total}} = 3.0$ in^2 (2 calipers, per the assumed design inputs), $R_{\text{eff}} = 4$ in, the denominator is $2(0.45)(3.0)(4) = 10.8$, giving $P_{f,\text{required}} = 6626/10.8 = 613$ psi and $P_{r,\text{required}} = 1976/10.8 = 183$ psi. The given targets are roughly double these minimums ($1200/613 \approx 2.0\times$ front, $400/183 \approx 2.2\times$ rear), which a good margin to ensure the target deceleration.

Master cylinder force and bore diameter

According to Pascal's law, pressure applied to a confined incompressible fluid is transmitted equally throughout. This is important because the same pressure P from Equation 7 acts at the master cylinder as well. Force equals pressure times area, applied at the master cylinder piston:

$$F_{MC} = PA_{MC}$$

The bore area of a circular master cylinder piston is $A_{MC} = \frac{\pi d_{MC}^2}{4}$. Substituting and solving for the bore diameter:

$$F_{MC} = P \cdot \frac{\pi d_{MC}^2}{4}$$
$$d_{MC}^2 = \frac{4F_{MC}}{\pi P}$$

$$F_{MC} = PA_{MC}, \qquad d_{MC} = \sqrt{\frac{4F_{MC}}{\pi P}} \tag{8}$$

where F_{MC} is master cylinder output force and d_{MC} is master cylinder bore diameter.

Design Calculation: To minimize the bill of materials, a single master cylinder bore will be used for both the front and rear circuits, with the balance bar splitting pushrod force in the ratio of the target pressures (1200 : 400 = 3:1). With $F_{\text{foot,design}} = 70$ lbf and $PR = 6.0$, total pushrod force is $F_{\text{pushrod}} = (70)(6.0) = 420$ lbf. Splitting this 3:1 gives $F_{\text{front}} = 315$ lbf and $F_{\text{rear}} = 105$ lbf. Solving for a common bore using the front condition: $A_{MC} = 315/1200 = 0.2625$ in^2, so $d_{MC} = \sqrt{4(0.2625)/\pi} = 0.578$ in.

Pedal moment balance

The pedal arm is a rigid body pinned at a single pivot, so it can only rotate. Static rotational equilibrium requires the sum of moments about the pivot to equal zero. This is okay to do because even though the pedal arm may contain a small angular acceleration, it is small enough that the resulting inertial torque ($I\alpha$) is negligible compared to the torques already being analyzed.

$$\sum M_{\text{pivot}} = 0$$
$$F_{\text{foot}}\, d_{\text{foot}} - F_{\text{pushrod}}\, d_{\text{pushrod}} = 0$$

$$F_{\text{foot}}\, d_{\text{foot}} = F_{\text{pushrod}}\, d_{\text{pushrod}} \tag{9}$$

where d_{foot} and d_{pushrod} are the perpendicular distances from the pivot to each force's line of action.

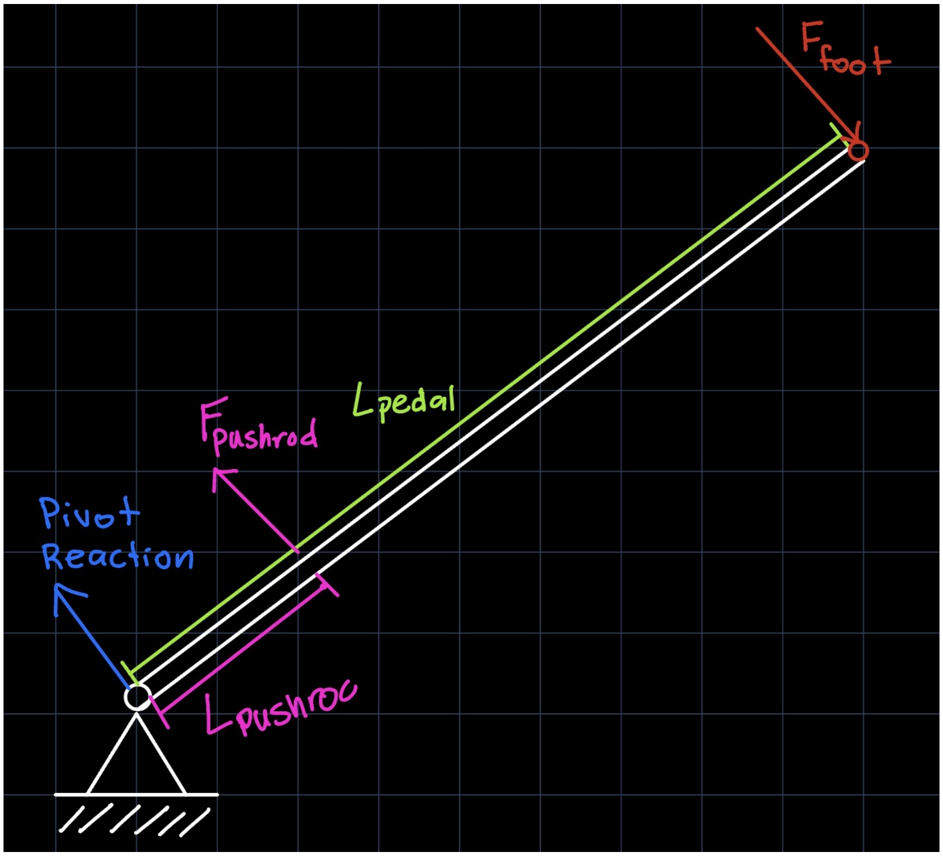

Figure 2: Free body diagram for pedal moment balance

Pedal ratio

Dividing both sides of Equation 9 by $F_{\text{foot}}\, d_{\text{pushrod}}$:

$$\frac{F_{\text{foot}}\, d_{\text{foot}}}{F_{\text{foot}}\, d_{\text{pushrod}}} = \frac{F_{\text{pushrod}}\, d_{\text{pushrod}}}{F_{\text{foot}}\, d_{\text{pushrod}}}$$
$$\frac{d_{\text{foot}}}{d_{\text{pushrod}}} = \frac{F_{\text{pushrod}}}{F_{\text{foot}}}$$

We can define this ratio as the pedal ratio PR. and noting that when both forces act approximately perpendicular to the arm the perpendicular distances reduce to the plain straight-line lengths along the arm, $d_{\text{foot}} \approx L_{\text{pedal}}$ and $d_{\text{pushrod}} \approx L_{\text{pushrod}}$:

$$PR = \frac{F_{\text{pushrod}}}{F_{\text{foot}}} = \frac{d_{\text{foot}}}{d_{\text{pushrod}}} \approx \frac{L_{\text{pedal}}}{L_{\text{pushrod}}} \tag{10}$$

where L_{pedal} is the distance from the pivot to the end of the pedal pad and L_{pushrod} is the distance from the pivot to the pushrod attachment point.

Design Calculation: Choosing $PR = 6.0$ (assumed above) and a packaging-constrained pushrod attachment distance $L_{\text{pushrod}} = 1.0$ in gives $L_{\text{pedal}} = (6.0)(1.0) = 6.0$ in. This is a good sizing because it fits nicely within the 10 in tall accelerator pedal envelope given in the design requirements.

Pushrod force

Multiplying both sides of Equation 10 by F_{foot}:

$$PR \times F_{\text{foot}} = \frac{F_{\text{pushrod}}}{F_{\text{foot}}} \times F_{\text{foot}}$$

$$F_{\text{pushrod}} = F_{\text{foot}} \times PR \tag{11}$$

From the Rule T.3.2.2's clarification above, the structural worst case uses $F_{\text{foot}} = 1500$ N (a more realistic ceiling). We can also solve with a comfortable driver force target in order to give us the operational design target.

Design Calculation: Operational case: $F_{\text{pushrod}} = (70)(6.0) = 420$ lbf (≈ 1868 N), matching Equation 8 above. Structural worst case, using $F_{\text{foot}} = 1500$ N: $F_{\text{pushrod,max}} = (1500)(6.0) = 9{,}000$ N (≈ 2023 lbf). This will be the design load carried forward into the Step 3 structural checks.

Pad knock-back and master cylinder stroke

Brake fluid is incompressible, so the volume of fluid the master cylinder piston displaces must exactly equal the volume needed to close the knock-back clearance across every piston in the circuit. The volume swept by the master cylinder piston over its stroke is:

$$V_{MC} = A_{MC}\, S_{MC}$$

The volume needed to close the clearance at each caliper piston is its area times the knock-back clearance; with n_{pistons} pistons in the circuit, the total volume needed is:

$$V_{\text{needed}} = n_{\text{pistons}}\, A_{\text{piston}}\, c_{kb}$$

Setting these volumes equal (conservation of incompressible fluid volume) and solving for the stroke:

$$
\begin{aligned}
A_{MC}\, S_{MC} &= n_{\text{pistons}}\, A_{\text{piston}}\, c_{kb} \\
S_{MC} &= \frac{n_{\text{pistons}}\, A_{\text{piston}}\, c_{kb}}{A_{MC}}
\end{aligned}
$$

$$
S_{MC} = \frac{n_{\text{pistons}}\, A_{\text{piston}}\, c_{kb}}{A_{MC}} \tag{12}
$$

where S_{MC} is the required master cylinder stroke, n_{pistons} is the number of pistons in the circuit, A_{piston} is the area of each piston, c_{kb} is the pad knock-back clearance, and A_{MC} is the master cylinder bore area.

Design Calculation: Front circuit has 2 fixed calipers, each with pistons on both sides of the rotor at 1.5 in^2 per side, giving $n_{\text{pistons}} = 4$ total pistons and a total circuit piston area of 6.0 in^2; with $c_{kb} = 0.010$ in and $A_{MC} = 0.2625$ in^2 from above: $S_{MC} = (6.0)(0.010)/0.2625 = 0.2286$ in.

Pedal travel at the pad, from work conservation

An ideal lever conserves mechanical work. This means that the work done by the driver's foot equals the work delivered to the master cylinder piston.

$$
W_{\text{foot}} = F_{\text{foot}}\, x_{\text{pedal}}, \qquad W_{MC} = F_{\text{pushrod}}\, S_{MC}
$$

Setting these equal:

$$
\begin{aligned}
F_{\text{foot}}\, x_{\text{pedal}} &= F_{\text{pushrod}}\, S_{MC} \\
x_{\text{pedal}} &= \frac{F_{\text{pushrod}}}{F_{\text{foot}}}\, S_{MC}
\end{aligned}
$$

Substituting $F_{\text{pushrod}}/F_{\text{foot}} = PR$ from Equation 10:

$$
x_{\text{pedal}} = PR \times S_{MC} \tag{13}
$$

where x_{pedal} is the pedal pad travel required to fully close the hydraulic clearance in Equation 12. Note that a higher pedal ratio, while multiplying force, simultaneously multiplies required travel by the same factor — this trade-off is inherent to the lever and cannot be avoided.

Design Calculation: $x_{\text{pedal}} = (6.0)(0.2286) = 1.371$ in.

Total pedal travel

Regenerative braking is not implemented through brake pedal travel in this design and the pedal is instead purely hydraulic across its entire stroke, so total travel is simply the hydraulic travel already derived, plus a small practical margin.

$$x_{\text{total}} = x_{\text{pedal}} + \text{margin} \tag{14}$$

where x_{total} is the total pedal travel from fully released to fully applied, x_{pedal} is the hydraulic travel required from Equation 13, and margin accounts for manufacturing tolerance and pad wear over a session.

Design Calculation: $x_{\text{total}} = 1.371 + 15\% \approx 1.577$ in. As a geometry check, this corresponds to a pedal rotation angle of $\arctan(1.6/6.0) \approx 14.9°$.

Ideal front brake bias as a function of deceleration

The ideal front bias β_{ideal} is the fraction of total braking force that should go to the front axle so that each axle uses its total available grip in proprotion to its dynamic load, $\beta_{\text{ideal}} = W_{f,\text{dyn}}/W$ (since $W_{f,\text{dyn}} + W_{r,\text{dyn}} = W$ from Equation 4). Substituting $W_{f,\text{dyn}} = W_{f,\text{static}} + \Delta W$ from Equation 4, and $\Delta W = ma_x h/L$ from Equation 3:

$$\begin{aligned}\beta_{\text{ideal}} &= \frac{W_{f,\text{static}} + \Delta W}{W} \\ &= \frac{W_{f,\text{static}}}{W} + \frac{ma_x h}{WL}\end{aligned}$$

Since $W = mg$ from Equation 1, the ratio m/W simplifies to $1/g$:

$$\frac{ma_x h}{WL} = \frac{ma_x h}{mgL} = \frac{a_x h}{gL}$$

$$\beta_{\text{ideal}} = \frac{W_{f,\text{static}}}{W} + \frac{a_x h}{gL} \tag{15}$$

where β_{ideal} is the ideal front bias fraction. Because a_x appears in this expression, the ideal bias is not a constant and it shifts forward as braking intensity increases.

Design Calculation: At the target deceleration $\beta_{\text{ideal}} = 0.48 + \dfrac{(14.715)(0.30)}{(9.81)(1.55)} = 0.48 + 0.290 = 0.770$, or 77.0% front. This is good because the actual bias will always be set to 75% to accommodate for the 3:1 ratio in the design.

Practical adjustment range

The full theoretical swing in ideal bias between coasting ($a_x = 0$) and the target deceleration is found by evaluating Equation 15 at both conditions and subtracting:

$$\begin{aligned}\Delta\beta_{\text{theoretical}} &= \beta_{\text{ideal}}(a_x = a_{x,\text{target}}) - \beta_{\text{ideal}}(a_x = 0)\\ &= \left(\frac{W_{f,\text{static}}}{W} + \frac{a_{x,\text{target}}h}{gL}\right) - \frac{W_{f,\text{static}}}{W}\\ &= \frac{a_{x,\text{target}}h}{gL}\end{aligned}$$

This is normally going to be a very large swing at racing deceleration levels, which would be impossible for a mechanical balance bar to track dynamically in real time. Because of this the bias is fixed at a nominal design point rather than continuously adjusted.

A practical target is a balance bar adjustment range of ± 5 to ± 10 percentage points around the nominal front bias, centered on the threshold-braking design point used to satisfy the dynamic brake test (Rule IN.14.3).

Design Calculation: $\Delta\beta_{\text{theoretical}} = 0.770 - 0.480 = 0.290$, a 29.0 percentage point theoretical swing between coasting and full threshold braking which confirms why a fixed design point with a small practical adjustment range is much more practical instead of trying to track the full swing.

Packaging Constraints

Material properties of the given mounting plate

The provided mounting plate is $\frac{1}{4}$ in thick 7075-T6 aluminum. Published typical properties for 7075-T6 are:

- Yield strength: $\sigma_{\text{yield}} \approx 503$ MPa (73 ksi)
- Ultimate tensile strength: $\sigma_{\text{ultimate}} \approx 572$ MPa (83 ksi)

Sizing and placement of the brake pedal

The given accelerator pedal occupies a 4 in long, 2.5 in wide, 10 in tall envelope. The brake pedal box will be positioned laterally with a 4 in gap to the left of the accelerator envelope. This is done so that it can be sized for independent left-foot and right-foot of the brakes and accelerator respectively.

The brake box's depth (front-to-back) footprint is driven by the pedal arm's orientation rather than its full length: because the arm is oriented mostly vertically and the horizontal depth swept by the pad through its travel is small (approximately 1.6 in as calculated in the Performance Targets section). Adding a 1.5 in allowance for the pivot bracket and balance bar clearance behind the pivot, the total depth footprint from the firewall to the pad at full press is approximately 3.1 in.

Ergonomics

Pedal face angle range

Using the total pedal travel from Equation 14 and the pedal arm length from Equation 10:

$$\theta_{\text{total}} = \arctan\left(\frac{x_{\text{total}}}{L_{\text{pedal}}}\right) = \arctan\left(\frac{1.6}{6.0}\right) \approx 14.9^\circ$$

The pedal face sweeps approximately 14.9° total, from rest to full press. Since regenerative braking is not implemented through the pedal no dead-zone travel is required.

Pedal pad size and shape

- **Width, 2.75 in:** chosen to comfortably support the ball of a shoe
- **Height, 3.5 in:** chosen to give the foot enough vertical surface to maintain contact through the full sweep
- **Rectangular shape :** a rectangle with rounded corners is the simplest shape to waterjet-cut accurately allowing fabrication costs to remain low

Architecture and Component Selection

Architecture

Force path through the system

There is a specific sequence in which force travels through this system: the driver's foot presses the pedal pad, the pedal pad is mounted on the pedal arm, the pedal arm rotates about a fixed pivot point, the pivot transmits force to the pushrod, the pushrod pushes the master cylinders, and the master cylinders send pressurized fluid through the brake lines to the calipers.

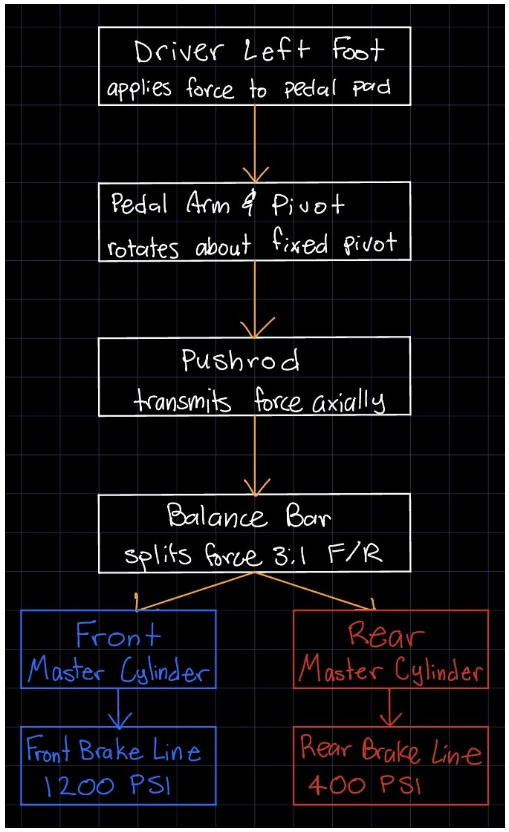

Figure 3: Force path block diagram.

Pedal type: floor-mounted

The pedal will be floor-mounted. This is chosen because a floor-mounted pivot keeps the pivot bracket close to the $\frac{1}{4}$ in aluminum mounting plate, so the load path into the plate is much more simple and avoids a tall cantilevered bracket that a hanging design would require.

Two master cylinders with a balance bar

Two separate, identical master cylinders will be used and they will be connected by a balance bar, where the front and rear hydraulic circuits fully independent of each other. This satisfies Rule T.3.1.3 (two independent hydraulic circuits) because a failure in one master cylinder or circuit should not affect the other, and it also satisfies Rule T.3.1.4 (separate fluid reservoirs) automatically, since each master cylinder is a self-contained single-circuit unit with its own dedicated reservoir, rather than requiring a separate step to add reservoirs.

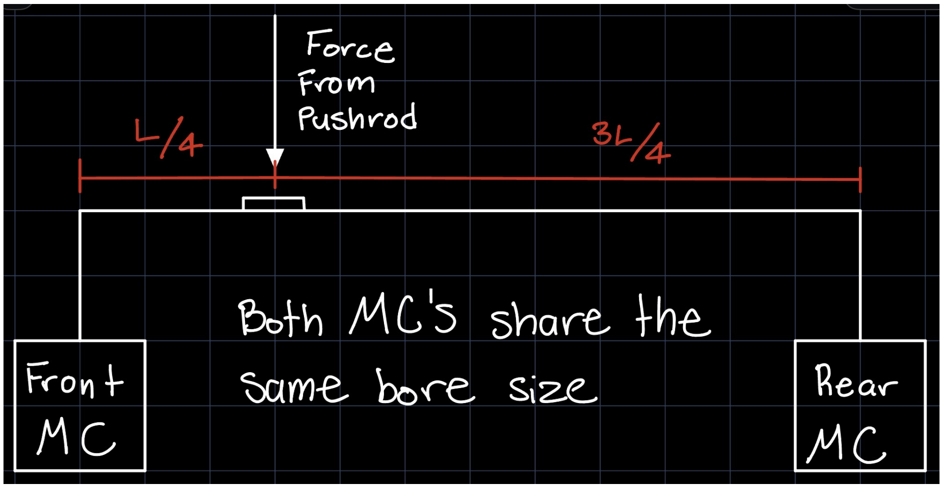

Figure 4: Master cylinder and balance bar sketch.

Balance bar pivot position

We can treat the balance bar itself as a rigid body in equilibrium, with the pushrod force F applied at the pivot and the resulting reaction forces R_{front} and R_{rear} at each end:

$$F = R_{\text{front}} + R_{\text{rear}}$$

Taking moments about the front end and about the rear end allows us to eliminate R_{front} and R_{rear}:

$$R_{\text{rear}} = \frac{Fd}{L}, \qquad R_{\text{front}} = \frac{F(L-d)}{L}$$

where d is the distance from the front end to the pivot point and L is the full length of the balance bar. Setting $R_{\text{front}} = 3R_{\text{rear}}$ to match the target 3:1 split from Equation 8:

$$\frac{F(L-d)}{L} = 3 \cdot \frac{Fd}{L} \implies L - d = 3d \implies d = \frac{L}{4}$$

The pivot should therefore be positioned one quarter of the way from the front end of the balance bar and three quarters of the way from the rear end. This makes intuitive sense that whichever end the pivot sits closer to receives the larger share of the force in a linear fashion.

Pivot: justification for bushing and not bearing

The pivot will use a flanged bronze bushing rather than a needle bearing, because a bushing is lower cost and reduces part count. The pivot in this application has a very low rotational speed so the precision and reduced friction of a bearing isn't really neccesary.

Off-the-Shelf Component Selection

Master Cylinder

#	Brand / Part	Bore	Material	Notes
1	Tilton 76-562	9/16 in	Aluminum	Below calculated 0.578 in target but it rounds toward higher pressure which is the safer direction
2	Tilton 76-625	5/8 in	Aluminum	Rounds toward lower pressure than target
3	Wilwood GS Compact	5/8 in	Die-cast aluminum	Same bore as Option 2 but different manufacturer
4	OBP Motorsport	0.625 in	Not specified	UK brand, marketed as lower-cost alternative to premium brands

Table 1: Master cylinder options considered

Selected: Option 1, Tilton 76-562. This is the only option found with a bore smaller than the calculated 0.578 in target, meaning it rounds toward a higher achieved pressure rather than lower. While OBP is a genuinely cheaper brand, its smallest available bore (0.625 in) isnt going to achieve the same safe rounding.

Balance Bar

#	Brand / Part	Type	Notes
1	Tilton 72-260	7/16"-20, steel bar, spherical bearings	Designed specifically for fixed-mount dual master cylinder systems, matching this architecture
2	Tilton 72-250	600-series, 3/8"-24	Same family but a thinner threaded rod

Table 2: Balance bar options considered

Rod Ends / Clevises

#	Brand / Part	Thread	Ult. Radial Load	Safety Factor
1	Aurora CM-4	1/4”-28	2,212 lbf	1.09
2	Aurora CM-5	5/16”-24	3,577 lbf	1.77
3	Aurora CM-6	3/8”-24	5,068 lbf	2.51
4	Aurora CM-7	7/16”-20	6,345 lbf	3.14
5	Aurora CM-8	1/2”-20	8,338 lbf	4.12
6	Aurora CM-10	5/8”-18	9,713 lbf	4.80
7	Aurora CM-12	3/4”-16	14,207 lbf	7.02

Table 3: Rod end options considered

Selected: Option 5, Aurora CM-8. CM-8 (8,338 lbf ultimate) is the smallest and lightest part on this list that clears Aurora’s recommended 4:1 safety factor, at 4.12.

Brake Over Travel Switch (BOTS)

#	Type	Approx. Cost	Notes
1	McMaster 7090K208, plunger actuator, SPST-NC (Honeywell / Omron-style)	$22.24	No dedicated commercial “BOTS” product exists and this SPST-NC switch is normally closed and opens when tripped, which matches Rule T.3.3.1–3.3.5
2	Speedway Motors Universal Mechanical Brake Light Switch	$14	Seems to be less flexible for a custom pedal over-travel trip point

Table 4: BOTS options considered

Selected: Option 1, McMaster 7090K208. Even though Option 1 is around 50% more expensive it best follows Rule T.3.3.1-3.3.5 and the possibility of mounting Option 2 seems unclear.

Brake System Encoder (BSE)

#	Type	Approx. Cost	Signal
1	Wilwood 300-11181 pressure switch (also listed as Speedway #83530011181)	$14.04	Binary
2	Speedway Motors #91603029, includes brass tee fitting	$27.99	Binary

Table 5: BSE options considered

Selected: Option 1, Wilwood 300-11181 hydraulic pressure switch. The rulebook language only requires "a sensor or switch that measures brake pedal position or brake system pressure," which a binary switch satisfies. Between the two binary options, Option 1 is cheaper and has the same functionality.

Fasteners

#	Spec	Source
1	SAE Grade 5 / Metric Class 8.8 min, hex or socket head, nyloc locknut	Generic hardware from McMaster-Carr

Table 6: Fastener specification

Custom-Made Components

Pedal arm

The pedal arm is a custom-made part. No off-the-shelf brake pedal arm can match this design's specific pedal ratio of 6:1, lever arm lengths of $L_{\text{pedal}} = 6.0$ in and $L_{\text{pushrod}} = 1.0$ in, and packaging envelope.

Combined bracket

A single bracket will hold the pivot mount, the master cylinder mount, and the balance bar mount, instead of three separate brackets, in order to reduce the number of custom parts from three to one. This bracket itself will be bolted to the given $\frac{1}{4}$ in aluminum plate.

Materials and Manufacturing

Steel versus aluminum for the pedal arm

Rule T.3.2.1 restricts the pedal arm to steel or aluminum if it is fabricated or to steel, aluminum, or titanium if it is machined. However, titanium can already be excluded here due to how expensive it is. The two realistic options can be compared directly against each other for this specific case.

Material properties. The comparison uses readily available properties for 6061-T6 aluminum and normalized 4130 chromoly steel:

Property	6061-T6 Aluminum	4130 Chromoly Steel
Young's modulus, E	10.0×10^6 psi	29.0×10^6 psi
Yield strength, σ_y	35,000 psi	63,000 psi
Density	0.098 lb/in^3	0.284 lb/in^3

$$\frac{E_{\text{steel}}}{E_{\text{al}}} = \frac{29.0}{10.0} = 2.9$$

Stiffness comparison. However, instead of just relying on the modulus ratio by itself we can also model the arm deflection directly. The section from the pushrod attachment to the pad is treated as a cantilever of length $L = 6 - 1 = 5$ in, fixed at the pushrod attachment point (since the incompressible hydraulic system anchors that point once resistance builds, as established in Performance Targets), with the foot force applied at the tip:

$$\delta = \frac{FL^3}{3EI}$$

where δ is the tip deflection, F is the applied force, L is the cantilever length, E is the material's Young's modulus, and I is the cross-section's area moment of inertia.

Cross-Section Calculation

The pedal arm has a rectangular cross-section. Looking at the pedal under a realistic load, a target safety factor of 2.5 is chosen to handle stress concentrations and manufacturing variability.

Using 6061-T6's published yield strength ($\sigma_{\text{yield}} \approx 35{,}000$ psi):

$$\sigma_{\text{allow,al}} = \frac{\sigma_{\text{yield}}}{FS} = \frac{35{,}000}{2.5} = 14{,}000 \text{ psi}$$

For a solid rectangle of width b and thickness t, $\sigma_{\text{bend}} = \dfrac{6M}{bt^2}$. Solving for the required section modulus at the critical section:

$$bt_{al}^2 = \frac{6M}{\sigma_{\text{allow,al}}} = \frac{6(1686)}{14{,}000} = 0.723 \text{ in}^3$$

Choosing a width $b = 2$ in, sized to fit within the pedal box's lateral clearance:

$$t_{al} = \sqrt{\frac{0.723}{2}} \approx 0.601 \text{ in}$$

Rounded to a standard stock thickness, $t_{al} = 0.625$ in at the critical section.

Using 4130 chromoly steel's published yield strength ($\sigma_{\text{yield}} \approx 63{,}000$ psi) and the same target factor of safety:

$$\sigma_{\text{allow,steel}} = \frac{63{,}000}{2.5} = 25{,}200 \text{ psi} \qquad \text{(vs. 14,000 psi for aluminum)}$$

Solving $bt_{steel}^2 = 6M/\sigma_{\text{allow,steel}}$ for steel at the same width $b = 2$ in:

$$bt_{steel}^2 = \frac{6(1686)}{25{,}200} = 0.4014 \text{ in}^3, \qquad t_{steel} = \sqrt{\frac{0.4014}{2}} \approx 0.448 \text{ in}$$

Stiffness comparison continued Using the derived thickness for each material($t_{al} = 0.625$ in, $t_{steel} = 0.448$ in), the area moment of inertia for a solid rectangular cross-section, $I = bt^3/12$, can be calculated separately for each:

$$I_{al} = \frac{(2)(0.625)^3}{12} = 0.0407 \text{ in}^4, \qquad I_{steel} = \frac{(2)(0.448)^3}{12} = 0.0150 \text{ in}^4$$

I measures how a cross-section's shape resists bending and it is completly independent of material.

Using the deflection formula stated above where $F = 337.2$ lbf and $L = 5$ in, deflection can be calculated for each material:

$$\delta_{al} = \frac{FL^3}{3E_{al}I_{al}} = \frac{(337.2)(5)^3}{3(10.0 \times 10^6)(0.0407)} = 0.0345 \text{ in}$$

$$\delta_{steel} = \frac{FL^3}{3E_{steel}I_{steel}} = \frac{(337.2)(5)^3}{3(29.0 \times 10^6)(0.0150)} = 0.0323 \text{ in}$$

$$\frac{\delta_{al}}{\delta_{steel}} = \frac{0.0345}{0.0323} \approx 1.07$$

Once each material is sized to its own optimal thickness, the deflections are nearly identical. Steel's higher E is almost entirely cancelled out by the thinner section its higher yield strength allows.

Mass comparison Using the resized dimensions over the full 6 in arm length:

$$m_{\text{al}} \approx (2)(0.625)(6)(0.098) = 0.735 \text{ lb}, \qquad m_{\text{steel}} \approx (2)(0.448)(6)(0.284) = 1.527 \text{ lb}$$

$$\frac{m_{\text{steel}}}{m_{\text{al}}} = 2.08$$

Steel comes out to be roughly double the mass of the aluminum once properly sized.

Additional qualitative factors.

- **Corrosion:** aluminum is naturally corrosion resistant while steel is not.
- **Weldability:** steel welds without significant local strength loss, wheras welding 6061-T6 locally significantly reduces strength right where a weld is placed.

Selected: Aluminum 6061-T6. Once both materials are correctly sized, aluminum is not that much softer than steel, while steel carries around double the mass for the same safety factor. This, along with the fact that aluminum is corrosion resistive makes it the best choice.

Structural Analysis

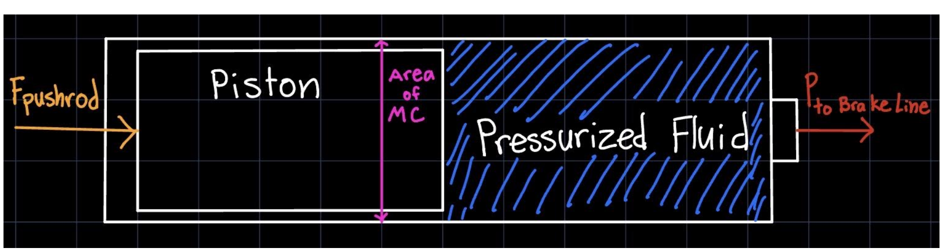

Figure 5: Master Cylinder Free Body Diagram

This diagram is the visual aid for the derviation of 8: $F_{MC} = PA_{MC}$.

Pivot and Mounting Bolt Reactions

Using the same free body diagram as Equation 9 where both the foot force, pushrod force, and pivot reaction force act approximately perpendicular to the arm, we can solve for all pivot force given the 1500 N load.

$$\sum F_t = F_{\text{foot}} - F_{\text{pushrod}} + F_{\text{pivot},t} = 0$$

$$F_{\text{pivot},t} = F_{\text{pushrod}} - F_{\text{foot}} = 9{,}000 - 1{,}500 = 7{,}500 \text{ N}$$

directed perpendicular to the arm, in the same rotational sense as the applied foot force.

For the bolts mounting the bracket to the plate we can consider the pivot, arm, pushrod, and master cylinder as one combined subsystem where the only external force onto this subsystem is the foot force itself (1500 N). Since the whole system is in equilibrium, the bolts must react a total force of 1500 N as well.

Locating the Critical Bending Section

The bending moment can be checked along the full length of the arm by breaking the pedal into parts at a position x and summing the moments.

Between the pivot and the pushrod attachment ($0 \leq x \leq 1$ in):

$$M(x) = F_{\text{foot}}(6 - x) - F_{\text{pushrod}}(1 - x)$$

Between the pushrod attachment and the pad ($1 \leq x \leq 6$ in):

$$M(x) = F_{\text{foot}}(6 - x)$$

We can evaluate this at the structural design load ($F_{\text{foot}} = 337.2$ lbf, $F_{\text{pushrod}} = 2023$ lbf):

At the pivot ($x = 0$): $M = 337.2(6) - 2023(1) \approx 0$ lbf·in

At the pushrod attachment ($x = 1$ in): $M = 337.2(5) = 1686$ lbf·in

At the pad ($x = 6$ in): $M = 337.2(0) = 0$ lbf·in

Based off of these calculations we can see that the bending moment is zero at both ends of the arm and peaks at the pushrod attachment point and thus is therefore the critical section for the bending stress.

Pivot Pin Sizing: Shear and Bearing

The pivot pin is a steel bolt (Grade 8) that is selected separately from the aluminum arm. This is because the pin is subjected to a concentrated load and it requires the higher strength that steel offers. The bracket holds the arm by clamping it between two flanges, putting the pin in double shear.

Using the pivot reaction force from the Pivot and Mounting Bolt Reactions section (7,500 N $\approx$ 1686.2 lbf), a target safety factor of 2.5, and Grade 8 steel's approximate shear strength (this would be about 60% of its tensile strength according to the von Mises yield criterion, this is 60% of 150,000 psi, giving 90,000 psi):

$$\tau_{\text{allow}} = \frac{90{,}000}{2.5} = 36{,}000 \text{ psi}$$

Solving for the required pin diameter in double shear, $\tau = \dfrac{F}{2A} = \dfrac{F}{2(\pi d^2/4)}$, where τ is the shear stress in the pin, F is the applied shear force (the pivot reaction, 1686.2 lbf), A is the pin's cross-sectional area, and d is the pin diameter:

$$d = \sqrt{\frac{2F}{\pi \tau_{\text{allow}}}} = \sqrt{\frac{2(1686.2)}{\pi(36{,}000)}} \approx 0.173 \text{ in}$$

Rounding up to a standard size, a $\frac{1}{4}$ in pin is selected.

Bearing stress at the pin hole acts on the surrounding aluminum arm, at the 0.625 in thickness we found in the Cross-Section Calculation section, where σ_{bearing} is the bearing stress, F is the same pivot reaction force, d is the pin diameter, and t is the arm thickness at the hole:

$$\sigma_{\text{bearing}} = \frac{F}{d\,t} = \frac{1686.2}{(0.25)(0.625)} = 10{,}791 \text{ psi}, \qquad FS = \frac{35{,}000}{10{,}791} \approx 3.24$$

This clears the 2.5 target safety factor with a $\frac{1}{4}$ in pin.

Pushrod Buckling Check

Steel is chosen in preference to aluminum as the material for the pushrod as a $\frac{1}{2}$"-20 thread for the Aurora CM-8 rod ends is much more reliable in steel compared to the use of a thin aluminum rod. Also, steel has a much higher safety margin compared to the pushrod's assumed length of 3 inches. Thus, the pushrod will be a solid steel rod, 0.5 in diameter, with an estimated length of 3 in based on the compact packaging established in the Architecture section. We can now use Euler buckling equation:

$$I = \frac{\pi d^4}{64} = \frac{\pi (0.5)^4}{64} = 0.00307 \text{ in}^4$$

where I is the pushrod's area moment of inertia and d is its diameter (0.5 in, assumed above).

$$F_{cr} = \frac{\pi^2 EI}{(KL)^2} = \frac{\pi^2 (29 \times 10^6)(0.00307)}{(1 \times 3)^2} \approx 97{,}600 \text{ lbf}$$

where F_{cr} is the critical buckling load, E is the pushrod material's Young's modulus (29×10^6 psi for steel), K is the effective length factor (1, for the pin-pin end conditions, meaning both ends can rotate freely but cannot move sideways), and L is the pushrod's unsupported length (3 in, assumed above).

Comparing to the structural design load pushrod force (2023 lbf, Equation 11):

$$FS_{\text{buckling}} = \frac{F_{cr}}{F_{\text{pushrod}}} = \frac{97{,}600}{2023} \approx 48.2$$

where $FS_{buckling}$ is the resulting safety factor against buckling.

This is an extremely large margin and thus buckling is not a concern for this design.

Mounting to the Provided Plate

The bracket will be mounted to the plate with 4 bolts. Assuming an even distribution across the 4 bolts:

$$F_{\text{per bolt}} = \frac{337.2}{4} = 84.3 \text{ lbf}$$

where $F_{\text{per bolt}}$ is the shear force carried by each individual bolt

Using $\frac{1}{4}$"-20 Grade 5 bolts (matching the Critical Fastener specification, Rule T.8.2) and their shear strength (approximately 60,000 psi):

$$\text{Shear capacity} = \tau_{\text{allow}} A_{\text{bolt}} = (60{,}000)(0.0491) = 2945 \text{ lbf}, \qquad FS = \frac{2945}{84.3} \approx 34.9$$

where τ_{allow} is the bolt's shear strength, $A_{\text{bolt}} = 0.0491 \text{ in}^2$ is the cross-sectional area of a $\frac{1}{4}$ in bolt (from $\pi d^2/4$), and FS is the resulting safety factor against bolt shear.

Plate bearing stress, using the given $\frac{1}{4}$ in 7075-T6 plate thickness:

$$\sigma_{\text{bearing,plate}} = \frac{F_{\text{per bolt}}}{d\,t_{\text{plate}}} = \frac{84.3}{(0.25)(0.25)} = 1349 \text{ psi}, \qquad FS = \frac{73{,}000}{1349} \approx 54.1$$

where $\sigma_{\text{bearing,plate}}$ is the bearing stress in the plate, d is the bolt diameter (0.25 in), t_{plate} is the plate thickness (0.25 in), and 73,000 psi is 7075-T6's yield strength.

Both of these checks passed with an extremely large margin.

Torsional and Off-Center Loads on the Pedal Pad

Torsion could develop if the driver's foot comes into contact with the pad in an off-centered position, but due to the fact that the width of the pad (2.75 in) is limited compared to that of the arm (2 in), a torsional analysis was not done.

Manufacturability and Assembly

Shape versus manufacturing process

The combined bracket carries three distinct features (pivot mount, master cylinder mounts, balance bar mount). Each of them is a pretty simple hole or bolt pattern in a flat piece. This makes it very suitable for waterjet cutting rather than a more complex machining process.

Assembly sequence

1. The flanged bronze bushing is to be inserted into the pivot hole of the combined bracket.
2. The pedal arm is placed on top of the bushing, and the $\frac{1}{4}$ in steel pivot pin is inserted and secured with its positive locking mechanism (Rule T.8.2).
3. The two Tilton 76-562 master cylinders are bolted to their respective mounting locations on the same bracket.
4. The Tilton 72-260 balance bar is mounted between the master cylinders, with the clamp located at the calculated quarter-length point.
5. Pushrod is mounted between the attachment point of the pedal arm and input clamp of the balance bar with Aurora CM-8 rod ends used at both ends of the pushrod.
6. The combined bracket with pedal arm, master cylinders, and balance bar is mounted onto the provided aluminum plate with the four bolts.

Reliability

Corrosion protection

The pedal arm, combined bracket, and pedal pad are all made of 6061-T6 aluminum , which is naturally corrosion resistant. The components in direct contact with the assembly that are not resistant is the steel pivot pin, the mounting bolts, the balance bar, the pushrod, and its carbon steel Aurora CM-8 rod ends so for these components without an off the shelf protection, zinc-plating should be applied to make it corrosive resistant.

Preventing fastener loosening under vibration

All the fasteners in the design use a locking system which ensures that the connection cannot be undone by means of vibration. A positive lock cannot come apart due to vibration, unlike the friction system whose locking ability decreases over time.

Bill of Materials

Part	Qty	Material / Spec	Source	Approx. Cost
Pedal arm	1	6061-T6 aluminum, 0.625 in thick, waterjet-cut	Custom made	\$30–50
Combined bracket	1	6061-T6 aluminum, waterjet-cut	Custom made	\$40–60
Master cylinder	2	9/16 in bore, aluminum	Tilton 76-562	\$145 each
Balance bar	1	7/16"-20, steel bar, includes clevises for MC-side links	Tilton 72-260	\$74
Pushrod	1	Solid steel rod, 0.5 in dia., 3 in length	Custom cut stock	\$10–15
Rod ends	2	1/2"-20, carbon steel	Aurora CM-8	\$10–15 each
Pivot bushing	1	Flanged bronze	McMaster-Carr	\$8–12
Pivot pin	1	1/4 in dia., SAE Grade 8 steel	Generic hardware	\$3–5
BOTS switch	1	McMaster 7090K208, plunger actuator, SPST-NC	McMaster-Carr	\$22.24
BSE sensor	1	Wilwood 300-11181 hydraulic pressure switch, binary	Speedway Motors	\$14.04
Fasteners, 1/4"-20 Grade 5, w/ nyloc locknut	estimate of 12	SAE Grade 5, hex head	McMaster-Carr	\$10–15 total

Table 7: Complete bill of materials for the brake pedal assembly