ME 240 | INTRODUCTION TO MECHANICAL DESIGN AND MANUFACTURING

Final Design Report

Single-Pivot, Side-Pull Bicycle Rear-Brake Caliper

DESIGN / ANALYSIS / MANUFACTURING / TESTING

Group 23

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1. Team Members and Roles

Group 23 worked as a four-person team. Each person had a lead area to keep that part of the project moving, but all four members contributed to the design, analysis, testing, and report. Our original team also included Tadeo Vargas as FEA and Topology Lead. After he left the group, Dominic Belyaev took over those responsibilities while continuing to lead testing and validation.

Table 1. Group 23 member roles and responsibilities.

Member	Role	Primary Responsibilities
Simon Kaminer	Scheduling Lead	Coordinates team meetings, tracks deadlines, and oversees submission of all deliverables.
Max Spector	CAD & Manufacturing Lead	Owns the NX CAD models and engineering drawings; ensures the part is accurately modeled and manufacturable.
Diego Niiler	Documentation Lead	Ensures all design decisions are clearly documented and justified at each stage of the process.
Dominic Belyaev	Testing & Validation Lead; FEA & Topology Lead	Plans and executes physical testing on the bicycle; runs and interprets the FEA and topology studies (role assumed after T. Vargas left the group).

2. Introduction and Functional Decomposition

This report summarizes our design process for a single-pivot, side-pull caliper brake for the rear wheel of a bicycle. The final part was printed in SLS Nylon 12 and checked against the braking requirements from ISO 4210-2 and ISO 4210-4. The report moves through the project in the same order we worked: defining needs and metrics, analyzing loads, developing the geometry, selecting material assumptions, running FEA, planning for manufacturing, and testing the Week 8 and Week 10 parts.

The caliper is only one part of the bicycle brake system, so the functional breakdown below focuses on the pieces we designed: the two arms, the pivot interface, the cable connection, and the pad mounts. It does not include the brake lever, cable housing, wheel, or bicycle frame.

Functional decomposition

- Transmit tension from the brake cable to the brake arms.
- Convert cable tension into a clamping (normal) force at the brake pads.
- Generate a friction force between the brake pads and the wheel rim.
- Produce a braking torque that decelerates the rear wheel.
- Pivot the arms about the mounting (pivot) bolt.
- Return the pads to the released position when cable tension is removed.
- Locate and constrain the assembly relative to the bicycle frame.
- Permit adjustment of the pad-to-rim clearance.

3. Needs, Metrics, and Design Specifications

We ranked the caliper needs on a 1-5 scale and connected each need to measurable design metrics. The highest priorities were stopping the bike within the required distance and staying consistent with the ISO braking standards. The braking-distance values come from ISO 4210-2 and the brake-system input limits from ISO 4210-4. The other targets came from a mix of benchmarking, ergonomics, and the available design-space dimensions.

Table 2. Needs table with relative importance (1 = low, 5 = high).

#	Need	Importance
1	The brake stops the bicycle within the required braking distance.	5
2	The brake complies with ISO 4210-2 and ISO 4210-4.	5
3	The brake delivers sufficient pad clamping force from a comfortable hand input.	4
4	The brake fits within the design space without interfering with the frame, wheel, or the opposing arm.	4
5	The brake fully releases the pads (no residual rubbing) when the lever is released.	4
6	The brake is lightweight.	3
7	The brake allows the pad-to-rim clearance to be adjusted.	3
8	The brake survives its expected actuation life without fatigue failure.	3
9	The brake (in a production metal form) resists environmental conditions such as corrosion.	2

Table 3 is the Criteria for Evaluation used throughout the project. It gives the marginal and ideal targets for each metric and lists where each value came from. The ISO-based braking target is tied to the standard dry-ground requirement of stopping from 25 km/h within 15 m, with the tested speed adjusted using the velocity-squared relationship described in Section 8.

Table 3. Criteria for Evaluation: metrics with marginal and ideal targets.

Metric	Marginal	Ideal	Units	Source
Weight	< 300	< 150	g	Benchmarking
Braking distance	< 7.6	< 5.2	m	ISO 4210-2
Actuation force	< 180	< 100	N	Ergonomics / benchmarking

Metric	Marginal	Ideal	Units	Source
Brake lever-to-grip clearance	> 40	> 55	mm	Ergonomics
Overall caliper width	< 55	< 35	mm	Design space
Pad-to-rim clearance (released)	0.3 – 2.0	0.5 – 1.0	mm	Function
Adjustment range	> 2	> 5	mm	Function
Fatigue life	> 2000	> 5000	cycles	Fatigue analysis (Sec. 5)

4. Loading Analysis and Concept Development

4.1 Braking-distance and clamping-force relationship

To connect braking performance to the cable input, we started with a simple energy balance. The bicycle and rider are treated as a mass m slowing from speed v to zero over distance d under a friction force:

$$\tfrac{1}{2} \cdot m \cdot v^2 = \mu \cdot m \cdot g \cdot d$$

This gives d = v^2 / (2μg), where μ is the limiting tire-ground friction coefficient and g is gravitational acceleration. This relationship tells us the best possible deceleration set by rear-wheel traction. Once the caliper can create enough rim friction to lock or nearly lock the rear wheel, the caliper is no longer the limiting factor.

The cable force reaches the pads through the caliper arm geometry. For an arm with cable tension Fin at distance L1 from the pivot and pad normal force N at distance L2, moment balance gives N = Fin(L1/L2). Rim friction is then Ff = μpadN. A larger L1/L2 ratio gives more clamping force for the same cable force, but it also reduces pad travel. That tradeoff drove most of the geometry decisions.

Geometric design parameters

The main variables we could change were the cable-to-pivot length, the pad-to-pivot length, the pivot location, the pad location, and the arm cross-section. Those dimensions set the lever ratio, the pad travel, and the stiffness of the arm. These were the dimensions that changed most between the Week 7 and Week 9 designs.

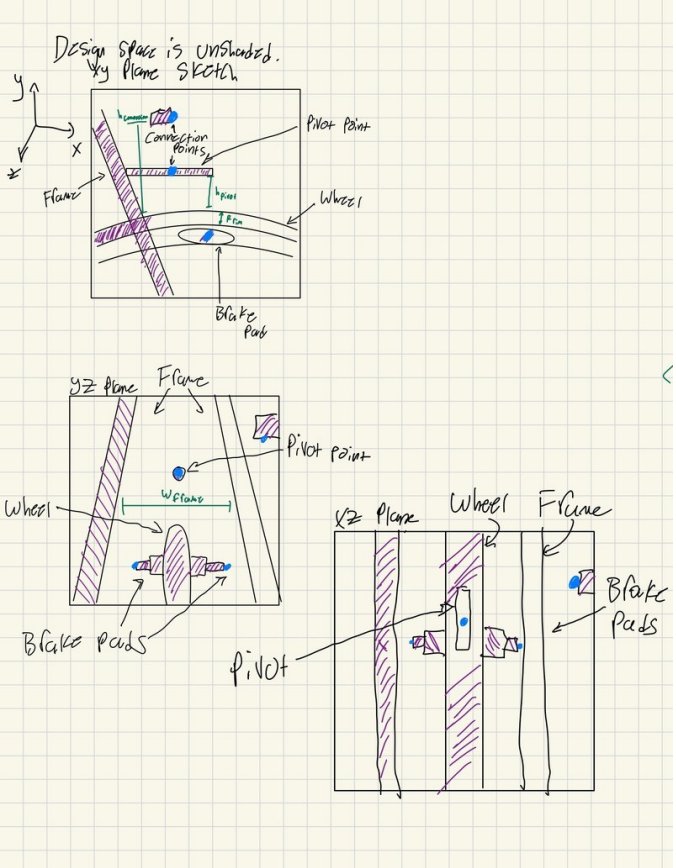


Figure 1. Maximum design space of the caliper system (front and side views), showing the wheel, frame, pad-contact region, and pivot point, with keep-out zones where material must not collide with the frame.

4.2 Free-body diagrams and point of maximum stress

We treated each caliper arm as a separate free body. The main external loads are the cable input force Fin, the pad normal force N, the pad friction force Ff, and the pivot reactions Rx, Ry, and Rz. The highest stress was expected at the arm root near the pivot because that section carries the largest bending moment from the pad load at the end of the arm.

At that cut section, the internal loads are axial force P, shear V, bending moment M, and torsion T. The normal and shear stress terms used for the hand calculation were:

$$\sigma = P / A + M \cdot c / I \qquad \tau = T \cdot r / J + V \cdot Q / (I \cdot b)$$

The bending term M*c/I is the largest contributor at the outside surface of the arm, so the root section controlled the early sizing. We used these equations to guide the first geometry, then used FEA to check the full stress state in the final models.

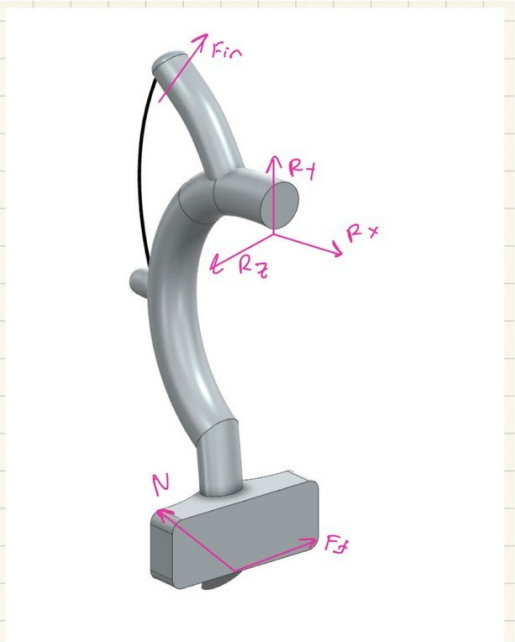


Figure 2. Free-body diagrams of the left and right caliper arms, with the cable force, pad normal and friction forces, and pivot reactions labeled, and the critical stress point identified at the arm root.

4.3 Concept iteration

Concept development started with hand sketches of the two caliper arms inside the available design space. We then translated those sketches into two NX part files. In the assembly, the pad mounting holes, cable mounting holes, and pivot holes had to line up correctly; otherwise the brake would not assemble or move. We aimed for a static safety factor of about 1.2 against the Nylon 12 strength limit. The Week 7 print met that analysis target, but it was based on the wrong bicycle design-space dimensions, so the final version had to be reworked around the correct bike geometry.

5. Material Selection and Allowable Stress

5.1 Build material and static allowable stress

The course-specified build material was SLS Nylon 12. We used an ultimate tensile strength of about 50 MPa as the static allowable stress for the FEA pass/fail check, meaning the peak von Mises stress in each arm needed to stay below 50 MPa. Nylon 12 also helped with the weight target; even after adding material for stiffness, the calipers remained well under the 150 g ideal. SLS was a good fit for this shape because the unsintered powder supports the part during printing, so the design did not need separate support structures.

5.2 Fatigue-informed allowable stress

Because the brake is actuated more than once, we also checked fatigue. The calculation followed the same process used in class: stress-concentration factors, notch sensitivity (q about 0.75-0.85), Kf = 1 + q(Kt - 1), and a Basquin/S-N estimate with Marin factors. That analysis did not support infinite life for the Nylon 12 part. For representative loading, it suggested finite life on the order of several thousand cycles, and a more conservative allowable stress of roughly 15-30 MPa for a 10^4-10^6 cycle target.

For this course prototype, we treated the final caliper as a finite-life demonstration part rather than a production brake. We therefore used a working allowable around 30-40 MPa: higher than the most conservative fatigue limit, but still below the 50 MPa static strength. Section 9 returns to this tradeoff when comparing the final FEA stresses with the fatigue estimate.

5.3 Metal alternatives for a production version

Although our printed prototype used Nylon 12, a production brake caliper would normally be aluminum. In Granta EduPack, we screened metals using three constraints: price per unit volume below $10,000/ft^3, at least acceptable salt-water corrosion resistance, and fracture toughness above 30 ksi*sqrt(in). We then compared the remaining materials using strength-to-density and stiffness-to-density indices. High-strength aluminum alloys, especially grades such as 7075-T6, offered the best balance of performance, cost, corrosion resistance, and machinability. For a production part, the critical holes would still need finish machining after forming.

6. Finite Element Analysis

We ran the FEA in NX for each arm. Stress was read as element-nodal, unaveraged von Mises stress in MPa. The models were constrained at the mounting interfaces and loaded with the cable force used in the deflection analysis (20 lbf, about 89 N). Stress spikes directly at fixtures or load points were treated as local FEA artifacts, so the reported maxima were taken from the physical load path rather than the singular constraint locations.

6.1 Stress

In the final design, both arms stayed under the 50 MPa static allowable. The stress also dropped compared with Week 7 after we added thickness near the critical root sections.

Table 4. Maximum von Mises stress and safety factor (FS = 50 MPa / σ) - initial vs. final design.

Caliper	Week 7 max stress (MPa)	Week 7 FS	Week 9 max stress (MPa)	Week 9 FS
Left	45.12	1.11	41.17	1.21
Right	48.66	1.03	40.81	1.22

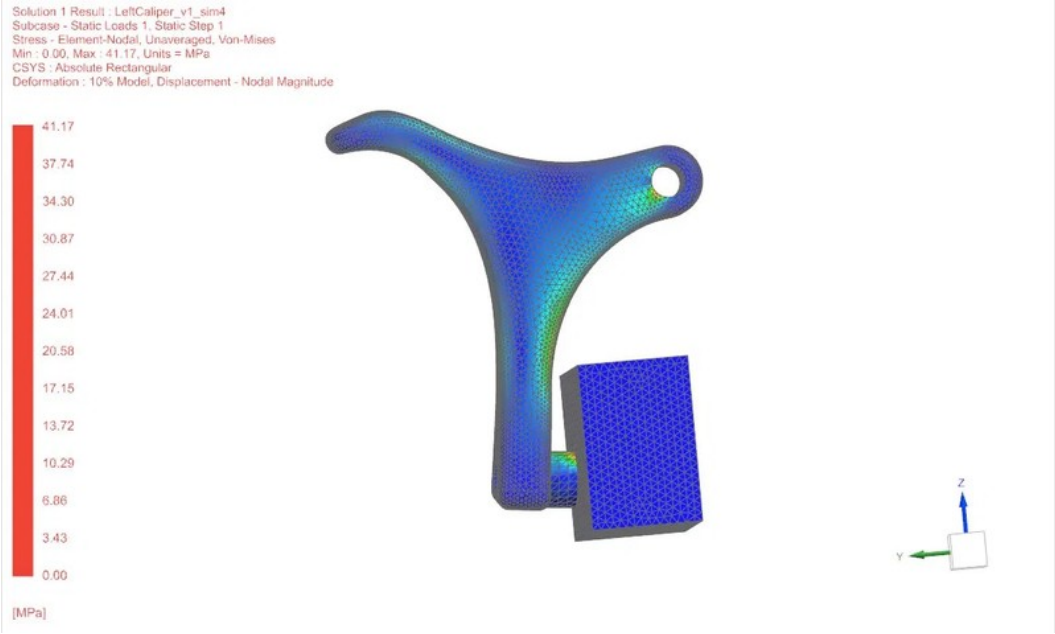


Figure 3. Left caliper von Mises stress, maximum = 41.17 MPa.

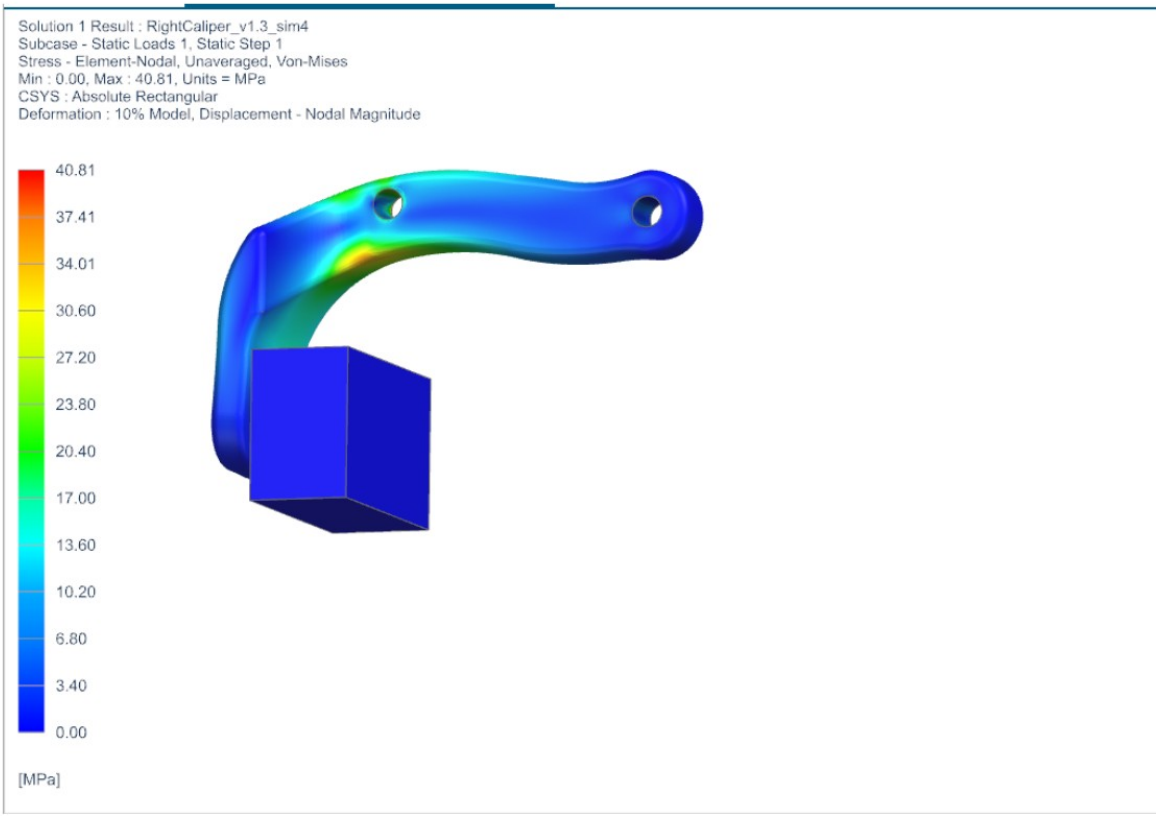


Figure 4. Right caliper von Mises stress, maximum = 40.81 MPa.

6.2 Deflection

The two arms together had to deflect less than 11 mm. If they deflected more than that, the lever would bottom out before the full hand input reached the pads. We calculated a deflection safety factor by

comparing the 11 mm limit with the simulated total deflection. The Week 9 redesign improved the deflection result for both arms.

Table 5. Maximum deflection and deflection safety factor - initial vs. final design.

Quantity	Week 7	Week 9
Left caliper deflection (mm)	3.339	2.733
Right caliper deflection (mm)	5.614	4.848
Total deflection (mm)	8.953	7.581
Safety factor (FS = 11 mm / total)	1.23	1.45

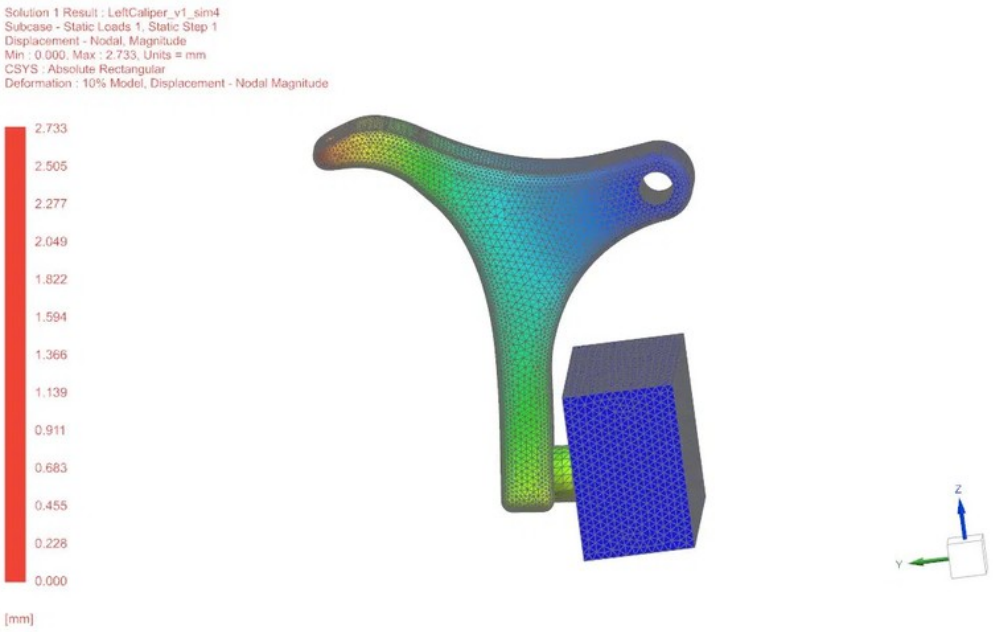


Figure 5. Left caliper displacement, maximum = 2.733 mm.

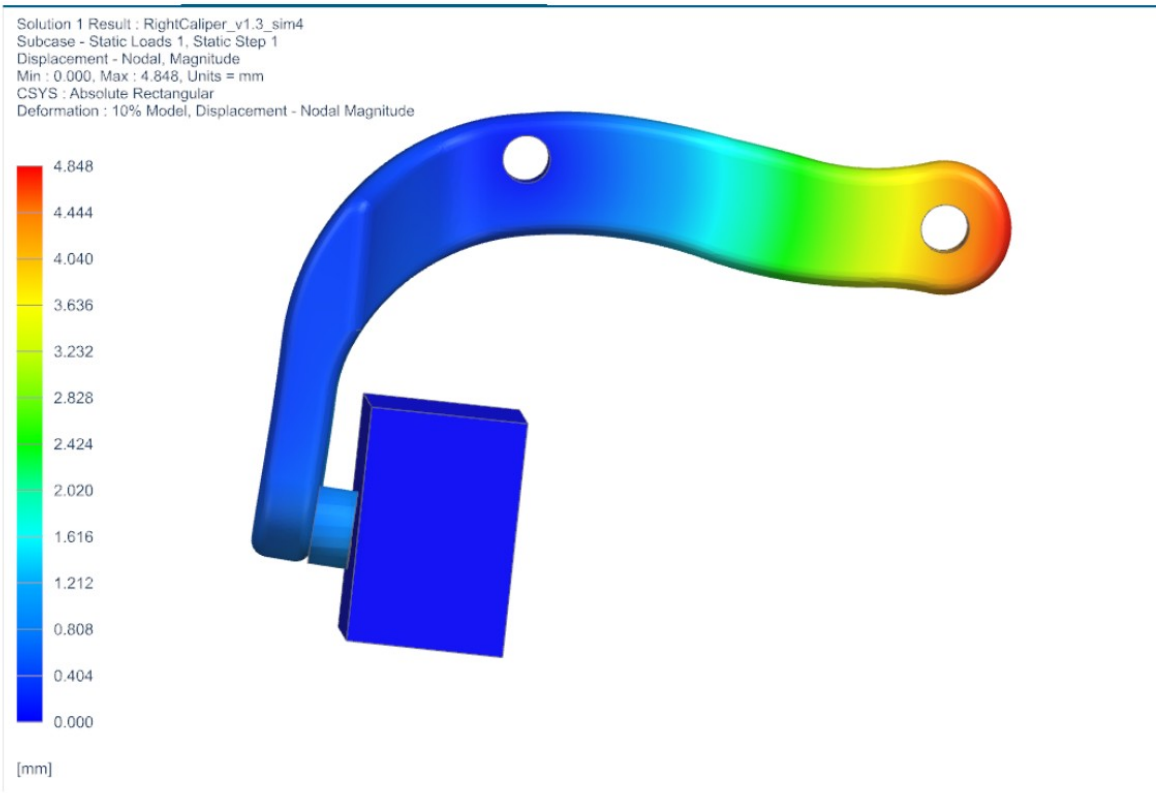


Figure 6. Right caliper displacement, maximum = 4.848 mm.

7. Design for Manufacturing and Tolerancing

7.1 Design for additive manufacturing (SLS Nylon 12)

We kept the SLS manufacturing constraints in mind while modeling the final arms. Each caliper was exported as a single solid body with no open surfaces or modeling errors. SLS does not require support structures in the same way many other additive processes do, but orientation still matters because layer direction can affect strength. We oriented the parts so the main bending loads would run mostly along the layer direction instead of across it. The group number was recessed in a non-critical area at least 1 mm deep, with text at least 3 mm tall in its smaller dimension, so it would remain readable after printing.

7.2 Tolerancing and GD&T

We added clearance to the features that connect to external hardware: the pivot bolt, the right-caliper mounting bolt, and the brake cable. The goal was to avoid post-print drilling and to let the arms rotate freely once installed. The datum scheme used the primary mounting face as datum A, with the locating holes and edges as datums B and C. The main controls were flatness on the mounting face, perpendicularity of the pivot hole to that face, and position tolerances on the pivot, cable, and pad holes. The pivot hole was the most important feature because even a small offset there would make the arms bind.

We also checked a bolt/washer/nut/plate tolerance stack-up to make sure the bolt would still protrude at least 1 mm beyond the nut in the worst case. That calculation set the basic boss dimension. For press-fit style features, we used the BB30 42 mm bottom-bracket example as a reference for medium-drive fit limits.

7.3 Alternative manufacturing scenarios

Aluminum alloy is the recommended material for a production version, but the best manufacturing process depends on volume, cost, and performance requirements.

Scenario A: Entry-level bike ($400 bike, 50,000 units/yr). Die casting is the best choice because the part can be made quickly and cheaply once tooling is paid for. At this volume, the tooling cost is spread over enough units to make sense. The pivot bore and pad-mounting holes would still need a secondary machining operation to meet the critical tolerances. Forging would improve strength, but the extra cost is difficult to justify for this price point.

Scenario B: High-performance road bike ($5,000 bike, 10,000 units/yr). Forging followed by CNC machining is the best fit. A forged 7075-T6 blank would give better fatigue resistance and avoid casting porosity, which matters for a safety-critical part. CNC finishing would then hold the tight hole tolerances, produce a premium surface finish, and allow pocketing in low-stress areas to reduce weight. Metal additive manufacturing could work, but the cost and throughput are not attractive unless the design gains a clear topology advantage.

Scenario C: Custom bike frame shop ($7,000+ bike, 200 units/yr). CNC machining from 6061-T6 billet is the most practical baseline. At 200 units per year, hard tooling for casting or forging would dominate the cost. CNC keeps the process flexible because geometry changes can be made by updating the CAM file. Metal additive manufacturing could be used for special custom geometries, but CNC is the simpler and more reliable default for this volume.

8. Testing and Validation

8.1 Test protocol and ISO adjustment

For testing, Professor Keys accelerated the bicycle to a stop line and then applied the rear brake as hard as possible. The measured values were the speed at the stop line (vtest) and the stopping distance (dtest). Because stopping distance scales approximately with velocity squared, we adjusted the result to the ISO 4210-2 reference speed using:

$$d_{max} = d_{ISO} \cdot (v_{test}^2 / v_{ISO}^2)$$

The calipers pass if dtest is less than dmax. Each group gets three attempts, and one passing run is enough because rider input and test speed vary between runs. The ISO reference values are 25 km/h stopping within 15 m on dry ground and 16 km/h stopping within 10 m on wet ground.

8.2 Week 8 - initial test (failed)

The Week 7 print failed the initial test. The problem was geometry, not weight or material strength. We had used an incorrect bike design space, so the arms and pads did not sit in the right place relative to the rim. With no cable force, the pads already rubbed the wheel; when the cable was pulled, the pads still did not press squarely on the rim. As a result, the brake could not create enough friction to stop the bike within the required distance. The mounting-bolt clearance was also too tight, so the assembly had to be forced onto the bike. Because the geometry was already wrong, we did not record the remaining metrics during that test; they would not have represented the intended design.

Table 6. Week 8 initial-test measurements. The braking-distance failure was traced to geometry, not material strength.

Metric	Marginal	Ideal	Measured	Result
Weight (g)	< 300	< 150	43.5	Pass
Braking distance (m)	< 7.6	< 5.2	> 7.62	Fail
Actuation force (N)	< 180	< 100	Not measured	—
Lever-to-grip clearance (mm)	> 40	> 55	Not measured	—
Overall caliper width (mm)	< 55	< 35	Not measured	—
Pad-to-rim clearance (mm)	0.3 – 2.0	0.5 – 1.0	Not measured	—
Adjustment range (mm)	> 2	> 5	Not measured	—
Fatigue life (cycles)	> 2000	> 5000	Not measured	—

8.3 Redesign for the final test

The final redesign was built around the accurate bicycle dimensions from Canvas. We resized both arms so the pads met the rim flat and symmetrically, increased material thickness near the root sections to improve stiffness, and reoriented the left-caliper cable hole so the cable seated vertically instead of horizontally. We also added clearance around the bolt locations so the calipers could be installed without forcing the hardware. These changes fixed the Week 8 fit problem and also lowered both peak stress and total deflection, as shown in Tables 4 and 5.

8.4 Week 10 - final test

We tested the redesigned calipers with the same protocol used in Section 8.1. Table 7 summarizes the recorded final results.

Table 7. Week 10 final-test results.

Quantity	Symbol	Value
Test velocity	v_test	Not separately recorded
Recorded braking distance	d_test	5.334
ISO-adjusted max allowable distance	d_max	7.62 (ISO marginal threshold)
Result (d_test < d_max ?)	—	PASS
Final weight	—	40
Actuation force	—	140
Pad-to-rim clearance (released)	—	0.05 (= 0.002 in)

9. Discussion and Conclusion

The final calipers passed the braking-distance test with a measured stopping distance of 5.334 m, compared with the 7.62 m ISO-adjusted threshold. They also met the weight, cable-force, and lever-clearance targets, so the geometry correction between Week 8 and Week 10 solved the main performance issue.

The redesign also improved the FEA results. Peak von Mises stress dropped from 45.12 to 41.17 MPa on the left arm and from 48.66 to 40.81 MPa on the right. Total deflection fell from 8.953 to 7.581 mm, which raised the deflection safety factor from 1.23 to 1.45.

Not every metric passed. The final caliper width was 60.6 mm, above the 55 mm marginal limit; released pad-to-rim clearance was only about 0.05 mm, below the 0.3 mm minimum; and the adjustment range was 1 mm instead of the required 2 mm. These issues came from the same constraint: after the Week 8 failure, we focused on making the brake fit and stop reliably, while compactness and adjustability became secondary.

The biggest lesson from the project was that the real bicycle geometry has to be verified before detailed design work starts. Our Week 8 failure came from using the wrong design space. The force analysis and FEA were useful, but they were applied to a geometry that did not fit the actual bike.

The most useful part of the analysis workflow was FEA. Simulating both arms before printing helped us see where the load path was working and where material was needed, especially at the arm root where the bending moment is highest. The hardest part was tolerance and interface design. Small errors in hole placement, cable routing, or arm width had a large effect on whether the brake could be installed and whether the pads contacted the rim correctly. Those issues were much easier to understand with physical hardware than from CAD alone.

The final design should be viewed as a successful course prototype, not a production-ready brake. The peak stresses around 41 MPa satisfy the 50 MPa static criterion, but they are above the most conservative fatigue allowable from Section 5. A real production version would need lower operating stress, a stronger material, or the aluminum manufacturing route discussed earlier.

The next iteration would address the three failed physical metrics. The arm width should be reduced by at least 6 mm overall, the pad mounts should move slightly outboard to create 0.3-1.0 mm of released clearance, and a slotted pad boss should be added to provide at least 2 mm of adjustment. The ideal next design would combine the correct Week 9 bike geometry with the more compact, material-efficient approach from the original topology-optimized design.