Harjaap Khela

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EDUCATION

Western University **Ontario, Canada**
Bachelor's of Engineering Science in Mechanical and Materials Engineering *Expected Grad: May '28*

- **Certifications:** CSWA - Certified SolidWorks Associate - Dassault Systèmes.

TECHNICAL SKILLS

Product Development: Mechanical System Architecture, Electromechanical Packaging, MCO Development, Material Selection, DFM, DFA, DFMEA, TA, CMF Specification, Design Validation Planning
Manufacturing: Injection Molding (Plastic & Metal), CNC Machining, Mass Production Drawings, GD&T, FAI/SPC Analysis, Vendor Collaboration (Overseas & Domestic), Build-to-Print Release.
CAD & Analysis: CATIA V6, SolidWorks, Surface Modeling, FEA (Ansys, SimSolid), PLM (Jira, Anovia).

EXPERIENCE

Ford, Advanced EV **California, USA**
Product Design Engineer Intern, Design Concepts Engineering Group *June '25 - Dec '25*

- Owned the **mechanical development** of a **build-to-print** consumer-electronics-style product for internal validation, creating multiple configurations to support **DOE testing** and parallel-path production architectures.
  - Developed the **cosmetic enclosure and internal mechanical architecture** from concept through tooling and production validation, balancing ID-defined A-surfaces, CMF, structural requirements, and manufacturability.
  - Worked cross-functionally with electrical engineering to define **MCOs, keep-outs and mechanical interfaces** that drove PCB layout, mounting strategy and integrated connector architecture.
  - Designed components for both **injection molding and CNC machining**, reviewing DFM with suppliers to accommodate process-specific constraints while maintaining critical interfaces across validation and production-intent designs.
- Developed and validated the **mechanical user experience** of an ecosystem of production-intent products by translating subjective characteristics into **measurable engineering requirements**.
  - Reduced axial misalignment between dynamically interfacing components through **critical tolerance analysis** accounting for **thermal expansion and material behavior**, reducing wobble and vibration under varying load conditions.
  - Designed fixtures and ergonomic test methods to quantify **touch temperature (thermal effusivity), actuation force, detent feel, and click ratios**, aligning results with ID's UX expectations.
  - Used **validation data to drive geometry, material and interface changes** that improved mechanical robustness, consistency and overall user experience.
- Designed critical **production assembly fixtures and SOP documentation** that enabled an otherwise infeasible assembly process and **unblocked a critical EVT build.**
  - Enabled successful assembly of **executive review units**, with the fixtures and associated process subsequently adopted as the production assembly standard.
  - Reduced **assembly cycle time by 50%** while also improving process tolerances and reducing yield loss.
- Owned **production definition and supplier dimensional resolution** for critical components by creating GD&T drawings, performing tolerance analyses that drove architecture and datum changes, defining **MSOP, FAI and SPC requirements**, and resolving out-of-spec supplier dimensions to enable repeatable production.

Western Formula Racing, FSAE **Ontario, Canada**
Mechanical Design Engineer, Suspension & Steering *Sep '23 - Present*

- Reworked suspension geometry and spring selection using vehicle dynamics analysis and Pacejka tire models, contributing to a **6% increase in maximum lateral grip** and reduced lap times.
- Optimized suspension components using **FEA-driven geometry refinement**, reducing component mass by **11.2%** while maintaining structural strength and reliability.
- Refined steering geometry and effort through **Ackermann, bump steer, roll steer and load analysis**, improving steering precision, driver feedback and handling consistency.

PROJECTS (SEE PORTFOLIO - VIEW ATTACHED DOCUMENT OR VISIT HARJAAPKHELA.COM)

Computer Vision Tracking Turret
Sole Mechanical Designer *Aug '26 - Present*

- Currently designing and building a computer-vision tracking turret with pan/tilt actuation, integrating camera-based target detection, precision electromechanical aiming and a CO2-powered launching system.

Harjaap Khela
Engineering Portfolio

Mechanical Product Design · Mechanical and Materials Engineering, Western University · Available Summer 2027
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CONTENTS

FORD MOTOR COMPANY · ADVANCED EV · CALIFORNIA, USA · JUNE TO DECEMBER 2025

Design Concepts Engineering

Lead DRI for the mechanical development of a consumer electronics style product built for internal validation. The work ran from early concept through production tooling, manufacturing and validation, inside industrial design defined A-surfaces and alongside electrical engineering, suppliers and manufacturing.

ROLE
Lead DRI, mechanical development

SCOPE
Enclosure, internal architecture, tooling, validation, production definition

METHODS
IM and MIM DFM, worst case tolerance analysis, CEM43 thermal dose, GD&T

DURATION
6 months

> Work performed under NDA. Engineering method, decisions and outcomes only. No product geometry, imagery, dimensions, tolerances, supplier names or production quantities are shown.

Mechanical architecture · Enclosure design within ID A-surfaces · Concept development · Production tooling · Manufacturing and DFM · Design validation · Production definition and release · Cross-functional integration

01 Scope and ownership

Internal use only, fully tooled and built in production quantities as a validation article. I was lead DRI for mechanical development from concept to production release.

Industrial Design set the A-surfaces. I designed inside them against packaging, structure, CMF, material, manufacturing, electrical integration and the validation plan.

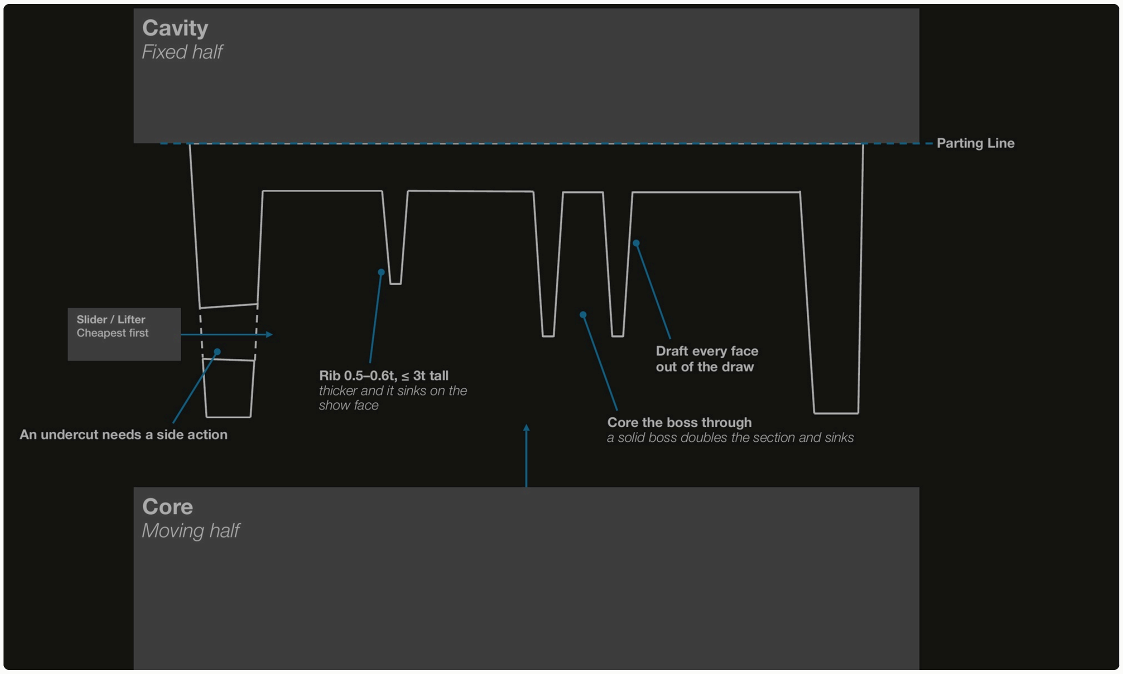
Electrical engineering and I defined the envelope together. My MCOs, interfaces and packaging envelopes drove PCB layout, component mounting and connector architecture.

02 Three build configurations

The product was built to worst case size, mass and electrical load so the receiving system was tested against a known article. Tooling would not arrive in time for the first builds, so I split the architecture by validation need and held the interfaces between configurations fixed.

CONFIG	PROCESS	PURPOSE
A	IM + MIM	Production intent, DVT and PVT
B	CNC machined	Representative hardware before tooling, EVT
C	CNC machined	Crash and impact articles

03 Injection moulding and CNC machining



Moulded part and tool: parting line, an undercut needing a side action, a rib, a cored boss and drafted faces.

IM and MIM. PC/ABS parts were fastened with Delta PT thread forming screws and MIM parts with machine screws, designed to thread engagement and minimum hole depth rules.

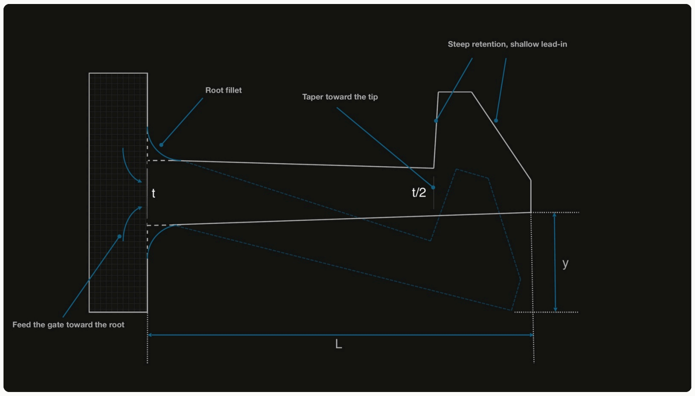
CNC machining. The parts were redesigned for machining so representative hardware existed before tooling. Moulding features were removed only where machining made them impossible or pointless.

Critical interfaces were held across both processes as closely as each allowed, so the manufacturing method did not become a variable in the test.

Gate location. Flow direction and weld line position decide whether a PC/ABS snap deflects or fractures at assembly, so I made gate location a requirement on the mould.

MIM parts are moulded oversize, debound and sintered to size, so non-uniform walls or unsupported features distort during sintering.

04 Snaps, screws and joining method selection



Snaps. Strain decides whether the snap survives assembly. Stiffness decides how much force it takes to push together.

Snap arm, worked on invented numbers

PC/ABS, E = 2300 MPa. Arm L 10 × b 5 × t 1.5 mm, undercut y = 0.80 mm, lead-in 30°, μ = 0.3.

Strain at assembly	$\varepsilon = 1.5 \cdot t \cdot y / L^2$	**1.80 %**, allowable 2 %
Second moment	$I = b \cdot t^3 / 12$	1.406 mm^4
Deflection force	$F = 3EIy / L^3$	7.8 N
Insertion force	$W = F(\mu + \tan\alpha) / (1 - \mu \cdot \tan\alpha)$	8.2 N per snap
Four snaps	4W	**33 N** to assemble

Strain scales with t/L^2 and force with t^3/L^3. Halving thickness cuts strain by two and force by eight, so the snap gets safe but feels loose. Lengthening the arm is usually the better change.

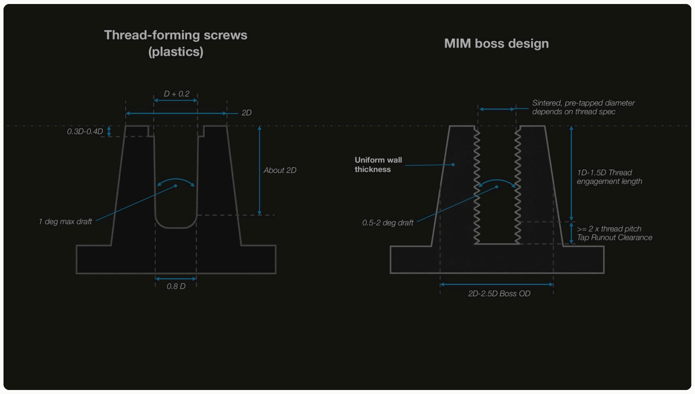
Screw engagement. For the same screw and the same clamp load, a thread forming screw in PC/ABS needs roughly three times the engagement of a machine screw in MIM. The joint is limited by the polymer under the head, not by the screw.

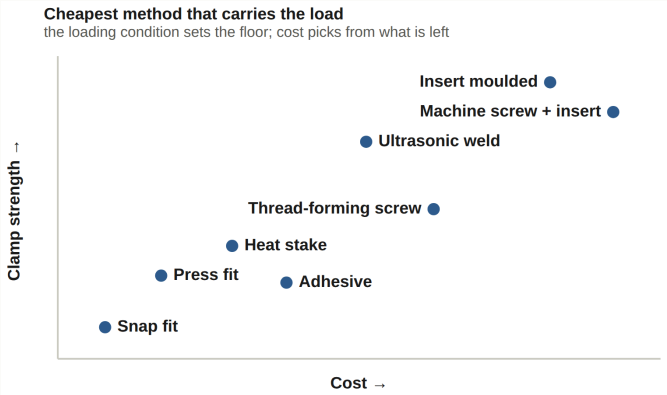
M2 screw into a moulded cover and a sintered boss

M2, stress area 2.07 mm^2, screw 700 MPa. MIM shear 312 MPa, PC/ABS shear 35 MPa. Head Ø4.0 on a Ø2.4 hole, PC/ABS 15 MPa bearing. Nut factor K = 0.20 dry, d = 2 mm.

Screw capacity	$A_t \cdot \sigma = 2.07 \times 700$	**1449 N**
Engagement, sintered	$L_e = 1449 / (312 \times 0.5\pi D)$	0.74 D, spec **1.5 D**
Engagement, moulded	$L_e = 150 / (35 \times 0.5\pi D) \times 3$	**2.05 D**
Head bearing limit	$\pi/4\ (4.0^2 - 2.4^2) \times 15$ MPa	**121 N**
Preload from torque	$F = T / (K \cdot d)$ at the handbook 0.25 N·m	**625 N**, five times the head limit
Torque to suit the head	$T = K \cdot F \cdot d = 0.20 \times 100 \times 0.002$	**0.040 N·m** for 100 N

The screw is not the weak link. The polymer under the head is. The answer is a flange head, a compression limiter, or a torque spec written from the bearing calculation rather than the fastener table.





Joining method. Screws were used because they went into a MIM housing and the same joint located and clamped a PCB. The general rule is the cheapest method that carries the load.

Insert moulding gives the best pull out and torque out and the worst serviceability. The inserts I worked on were not threaded. They were retained by undercuts and locking geometry, with walls thick enough that shrink loads them in compression and a tight shutoff to keep flash off clean faces.

Two shot moulding moulds the second resin onto the first, designing the bond and interlock into the geometry. One fewer assembly step and tolerance loop, in exchange for a more complex tool and a compatible material pair.

05 Touch temperature safety

"Feels too hot" is not a design requirement, so I measured what the interface does to a finger. I soaked the interfaces and measured heat transfer with a **thermisthesiometer**, a skin analogue probe, and converted it with the **CEM43 thermal dose equation** into equivalent minutes at 43 °C.

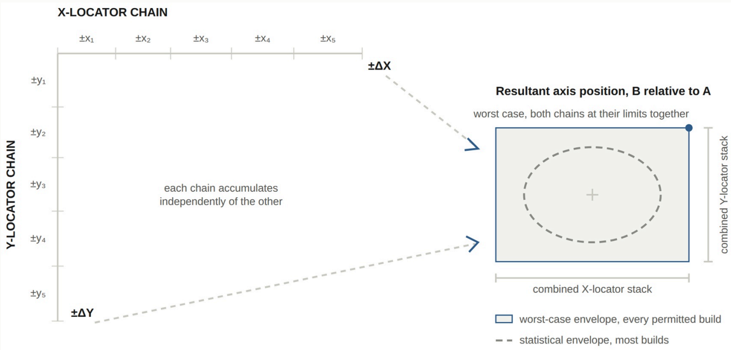
Thermal dose

Sapareto and Dewey. T is the measured contact temperature history, t the time at each step.

Thermal dose	$CEM43 = \Sigma\, t \cdot R^{(43 - T)}$	R = 0.25 below 43 °C, 0.5 at or above

Material choice. Two materials beat SUS304 thermally, but SUS304 passed and was what ID and CMF wanted.

06 Worst case tolerance analysis



Worst case is the full rectangle, corners included. A statistical method treats those corners as unlikely.

Two dynamic components had to hold a tight axial alignment, one set by an **X-locator** and one by a **Y-locator**, each with its own long chain. Misalignment was visible to the user, so I ran it worst case rather than RSS. The results changed architecture, dimensions, interfaces and datum strategy, and ID accepted the revised design.

07 Click feel measurement

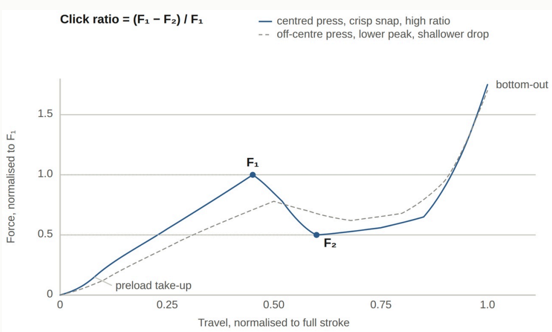
Controls built on the same switch can feel very different. On a separate set of production intent products I built the methods that turned feel into requirements. A tactile switch rises to a peak force $\mathbf{F_1}$, snaps through, and drops to $\mathbf{F_2}$ before bottoming out. The drop relative to F_1 is the click ratio.

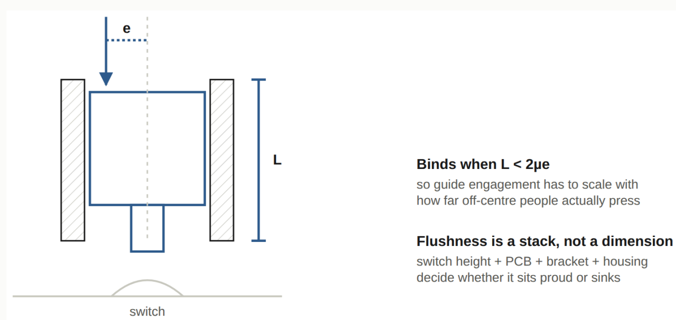
Preload. The tolerance stack between press surface and switch sets how much force is already on the dome. Too much and F_1 arrives early with a shallow snap, too little and there is lost motion.

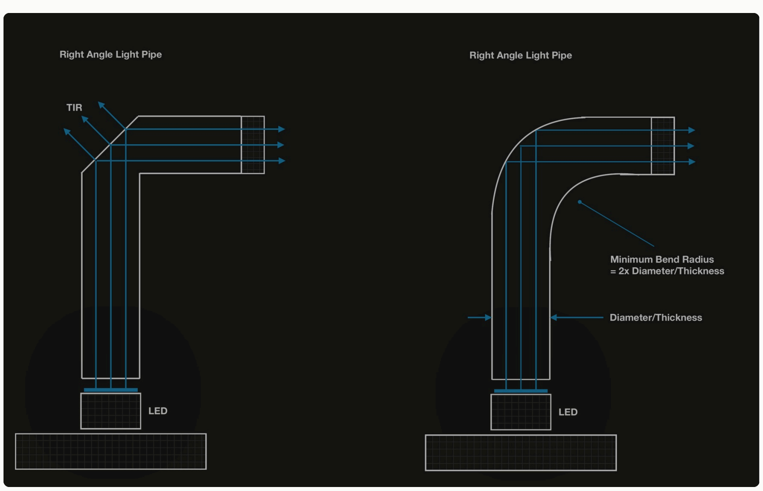
Press location. Users press off centre, so guide scheme, surface stiffness, switch position and allowable tilt matter more than the switch. A short guide binds before it translates.

Detent feel is measured as torque against angle: peak torque, assist depth, seat pitch and consistency between seats and directions.

Test fixtures. Force on an Instron and torque on a Meca500 six axis arm, with fixtures I designed so the part is rigidly located and the load axis stays normal. Click ratio, F_1, torque, travel, detent pitch and the **spread across press locations** became specifications.

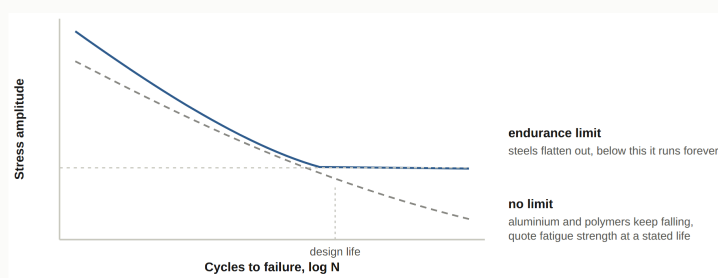






Light pipes. Exit face brightness is set by the LED coupling gap, which makes it a tolerance problem.

Fatigue. The plastic around a control sees every one of its rated switch cycles.



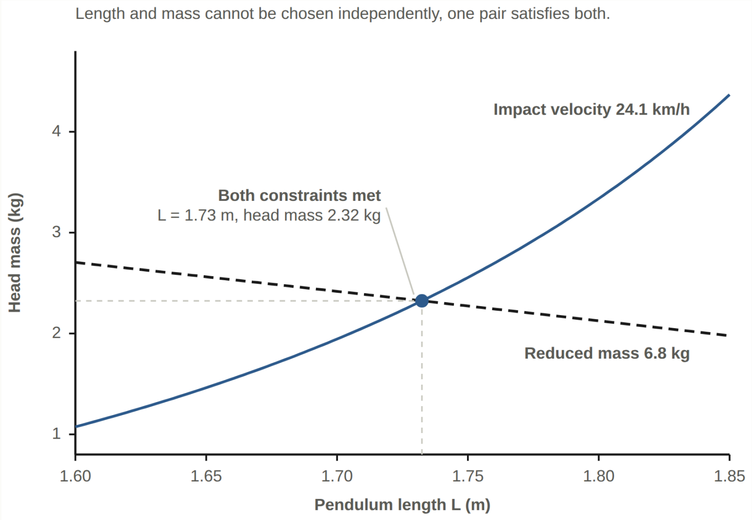
08 Assembly fixtures

On a separate product the supplier could not hold the precision assembly needed, tried several fixtures, and declared a **no-build condition**. The component was already released, so I solved it as a process problem using properties the parts already had, on site at an overseas vendor.

Outcome. Unblocked a critical EVT build, allowed blind assembly, and became the production process and SOP.

09 Head impact pendulum

ECE R21 Annex IV requires interior parts to survive a defined head impact without sharp edges. Official testing is slow, so I designed an in-house pendulum to screen candidates, confirmed against the official test by validation engineering. The regulation fixes impact speed and reduced mass, not arm length or headform mass, and those two are coupled.



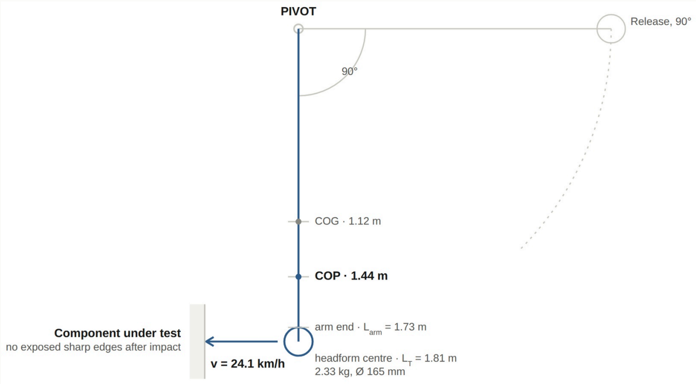
Each constraint solved for the head mass it needs. One length and mass satisfies both.

Two constraints on one pendulum

Arm treated as a uniform rod, headform as a point mass at L_T. Released from horizontal through 90°.

Energy at release	$m \cdot g \cdot \ell_{cm} = \frac{1}{2} \cdot I \cdot \omega^2$	$\omega = \sqrt{2g / a}$
Centre of percussion	$a = I / (m \cdot \ell_{cm})$	sets where the impact lands
Impact speed	$v = L_T \cdot \omega$, so $a = 2g \cdot L_T^2 / v^2$	**$a = 0.438 \cdot L_T^2$** at 24.1 km/h
Reduced mass	$m_r = m \cdot \ell_{cm} / a$	must equal **6.8 kg**
Solved together	quadratic in m_h	arm **1.73 m**, headform **2.33 kg**

The two requirements move in opposite directions with arm length, so exactly one pair satisfies both.



To scale along the arm, measured from the pivot.

PENDULUM	VALUE	SOURCE
Impact speed	24.1 km/h	ECE R21 Annex IV
Reduced mass	6.8 kg	ECE R21 Annex IV
Release	Horizontal, 90°	Regulation
Arm length	1.73 m	Derived
Headform mass	2.33 kg	Derived
Headform centre	1.81 m from pivot	Derived
Centre of gravity	1.12 m	Derived
Centre of percussion	1.44 m	Derived

10 Production drawings and release

I also owned production definition for critical parts: GD&T drawings, tolerance analysis, datum strategy, FAI and SPC requirements, MSOP and build to print release, and resolved supplier FAI deviations case by case.

WESTERN FORMULA RACING · FSAE · ONTARIO, CANADA · SEPTEMBER 2023 TO PRESENT

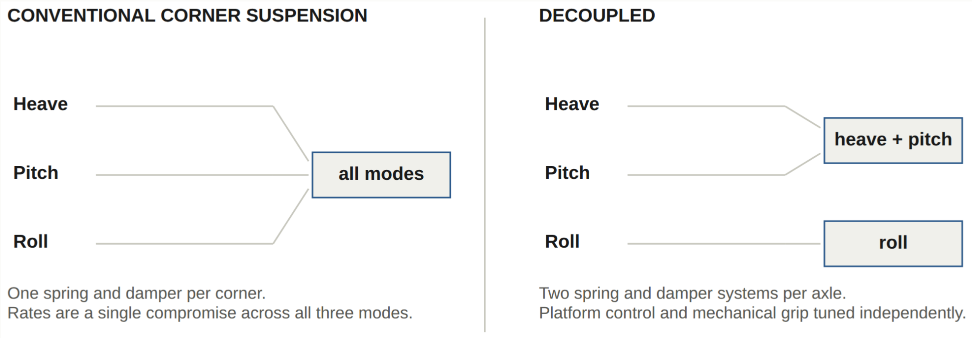
Suspension and Steering

Mechanical Design Engineer on a decoupled suspension car. The work covered suspension geometry and spring selection, FEA driven mass reduction on the front damper cradle, and steering geometry refinement.

OUTCOMES
+6% maximum lateral grip · −11.2% cradle mass · free play 5° to 2°

METHODS
Pacejka tire models · vehicle dynamics · FEA · Ackermann, bump and roll steer

01 Decoupled suspension architecture



Which vehicle modes actuate which element.

A conventional corner suspension actuates one spring in every vehicle mode, so its rate is a single compromise across heave, pitch and roll. Decoupling separates those decisions. There are two spring and damper systems per axle, driven so that heave and pitch act on one element and roll acts on the other. Each mode can then be tuned without disturbing the others.

MODE	CONVENTIONAL	DECOUPLED
Heave	Corner spring	Heave and pitch element
Pitch	Corner spring	Heave and pitch element
Roll	Corner spring and ARB	Roll element

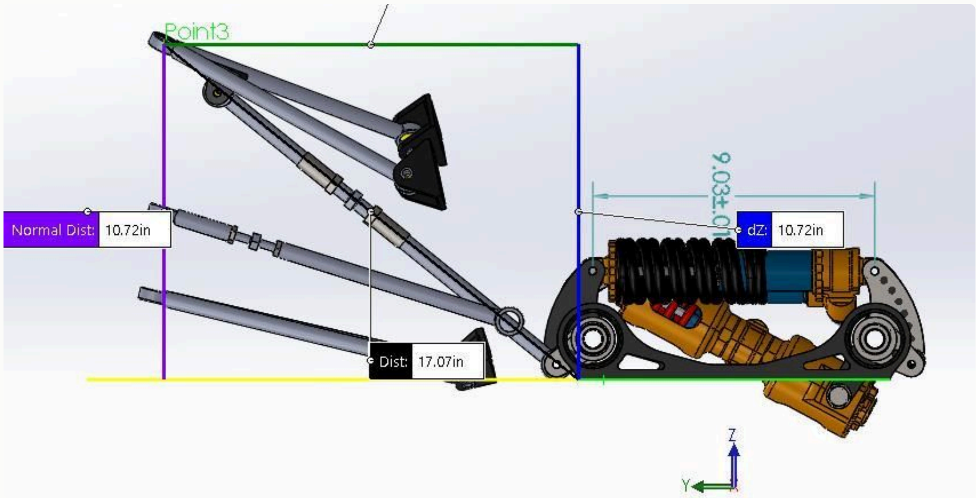
The practical consequence is that a roll stiffness change no longer drags heave stiffness with it, so balance can be moved between axles without re-deciding ride. That is the reason the architecture is worth its extra part count.

02 Geometry and spring selection

I reworked the suspension geometry and spring selection using vehicle dynamics analysis built on Pacejka tire models and measured mass properties. The tire model is what makes the problem tractable. It gives lateral force against slip angle and vertical load, so a geometry change can be evaluated as a change in grip rather than as an opinion.

Spring selection and deflection calculations were run against braking and cornering cases to keep the platform inside its useful range under load transfer. Geometry and motion ratios were refined iteratively. Motion ratio drives effective wheel rate, so the two cannot be chosen separately.

The result was a 6% increase in maximum lateral grip and reduced lap times.



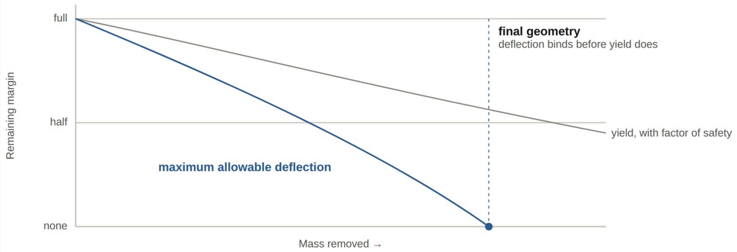
03 Front damper cradle

Unsprung mass and chassis mounted mass both cost lap time. The damper cradle carries real load, so taking mass out of it means proving that the remaining material is enough.

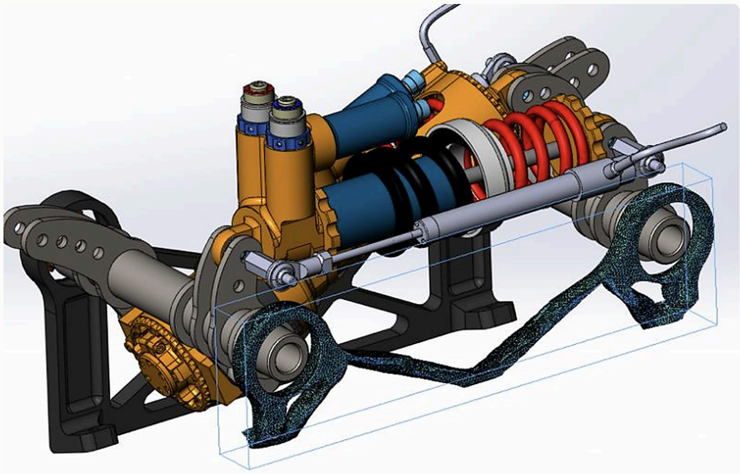
I optimised the geometry using iterative FEA rather than a single sizing pass. Load the baseline, find where material is doing nothing, remove it, and re-run against the same constraints until the geometry stops improving.

Input loads came from on-vehicle data acquisition on previous cars and covered the maximum cases across the suspension operating envelope. Two constraints had to survive every iteration. The first was yield with the required factor of safety. The second was a deflection limit, because the cradle locates the damper and compliance there is lost wheel control.

The final geometry came in 11.2% lighter than the baseline while satisfying both.



Schematic, not measured data. Deflection reaches its limit first, so deflection sets how much material can go.



04 Steering geometry and effort

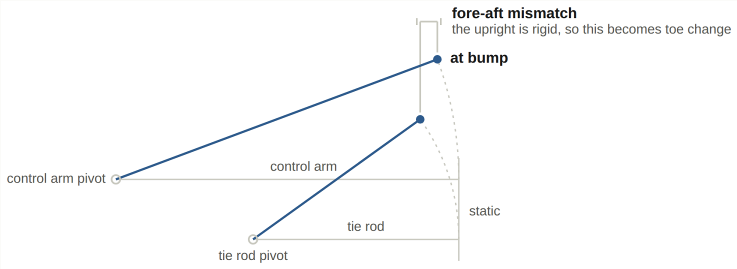
The steering work covered Ackermann geometry, bump steer, roll steer, and effort and load analysis. Bump steer and roll steer are the two that quietly ruin a car. If the tie rod arc does not match the control arm arc then the wheel steers itself through suspension travel and the driver is correcting inputs they never made.

Steering effort was calculated from driver input against lateral and longitudinal acceleration, so that the rack ratio landed somewhere a driver could still place the car accurately at the end of an endurance run. I also designed the column and rack mounting for compliance, because free play anywhere in the path reads at the wheel as though it were in the geometry.

Ackermann. The inner wheel travels a tighter radius than the outer, so parallel steer fights the tires at low speed and scrubs. How much Ackermann to run is not a free choice. At high slip angle the load transferred outer tire wants less steer rather than more, so the geometry is a compromise placed against where the car actually spends its lap rather than against the textbook construction.

Roll steer is the same defect expressed in roll instead of heave. The two sides travel in opposite directions, so the fore and aft mismatch at the tie rod ends becomes a steer input proportional to roll angle. It shows up as a car that changes line mid corner without the driver moving their hands.

Most of the outcome is qualitative, covering precision, feedback and consistency. One part is not. **Free play at the wheel came down from 5° to 2°.** That is the difference between a driver's first two degrees of input doing nothing and doing something.



The control arm and tie rod swing on different centres and lengths, so through travel their outer ends want different fore and aft positions. The difference comes out as steer.

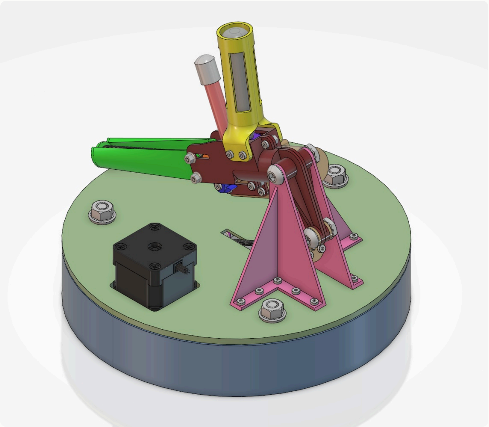
05 Outcomes

AREA	RESULT
Grip	+6% maximum lateral grip, reduced lap times
Cradle	−11.2% mass at the same yield factor of safety and deflection limit
Steering	Free play at the wheel 5° to 2°
Method	Pacejka based evaluation replaced qualitative geometry argument

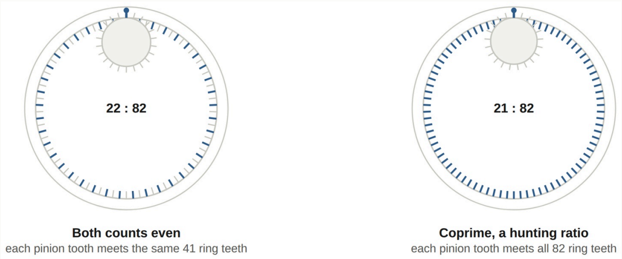
PERSONAL PROJECT · SOLE MECHANICAL DESIGNER · AUGUST 2026 TO PRESENT

Computer Vision Tracking Turret

A two axis turret that tracks a target under vision guidance and engages it with a CO_2 powered launcher. Continuous rotation in yaw, bounded travel in pitch, and a gear train that has to run without backlash. Mechanical design only.



01 Yaw and pitch axes



Every ring tooth a single marked pinion tooth will ever touch. With both counts even it meets the same 41 teeth forever. At 21 teeth it works its way around all 82.

Two degrees of freedom, each with its own actuator. X is yaw and rotates continuously. Y is pitch, bounded at 59° from 34.5° of elevation to 24.5° of depression. Continuous rotation rules out a sector gear or a linkage and constrains how power and signal cross the joint.

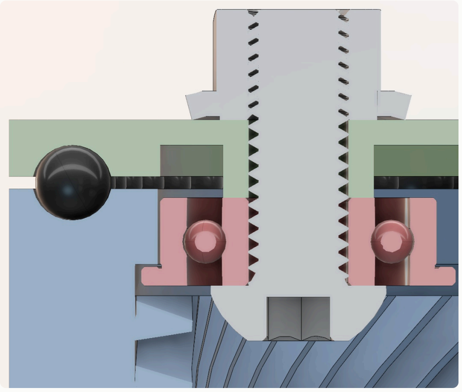
Tooth counts. The first layout paired a 22 tooth pinion with an 82 tooth ring. With both counts even, a pinion tooth only ever meets the same half of the ring, so any form error repeats at a fixed period. A 21 tooth pinion fixes it, at a 4.8% change in ratio (upward, which the torque budget wanted) and half a module of motor mount offset.

02 Locating the rotating plate

The top plate carries the whole head. It has to turn about one axis and stay put in the other five, and each of those constraints is a separate decision.

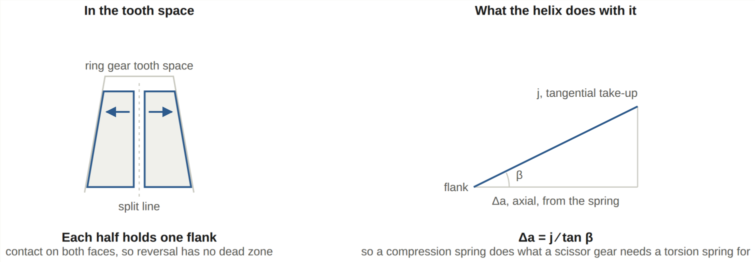
The ball groove runs 0.1 mm of clearance and the bearings run 0.2 mm, so the race locates the plate and the bearings only retain it axially and catch overloads. Specifying it as a relationship rather than two numbers is what survives printing, since both features drift with the same process bias.

The pinion rides on the moving plate and the ring is fixed, so plate float is centre distance error. At a 20° pressure angle, 0.1 mm of float is 0.073 mm of backlash.



03 Backlash in the yaw drive

Backlash puts dead motion in the control loop and the turret hunts instead of settling. An internal ring gear is compact but every reversal spends the tooth clearance. The options are a split gear, a cycloidal reducer or a harmonic reducer. **Nothing is decided.**



What the two halves do in the tooth space, and how the helix angle turns the axial push from the spring into tangential take up.

The split gear is specified furthest. The herringbone pinion splits at its apex into two opposite hand helices. Pushing them apart loads each against opposite flanks, and the helix angle turns the axial push into tangential take up.

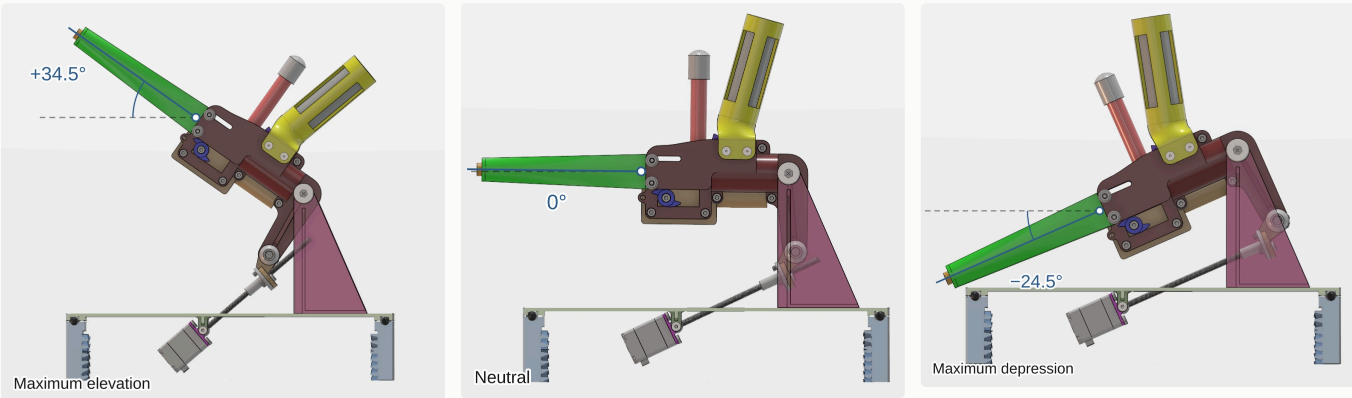
Preload only holds up to the working load, which is fine at about 2.4 N at the pinion. What decides it is the measured backlash of the current base, whether the ring bore can be freed for a cycloidal drive, and whether preload drag fits the motor budget.

04 Lead screw pitch drive

Pitch carries an unbalanced mass about a horizontal axis, so the load is gravity and never changes sign. A stepper drives a single start, fine lead screw that **self-locks**, so the axis holds position unpowered.

The screw pivots on a trunnion at the base and the nut drives a linkage that swings the head.

The payload preloads its own drive. The moment never reverses, so the screw never crosses its backlash and pitch needs no anti-backlash parts. Keeping the CG below the bore axis holds worst case moment at 87% of peak, against 57% if the axis went to vertical.

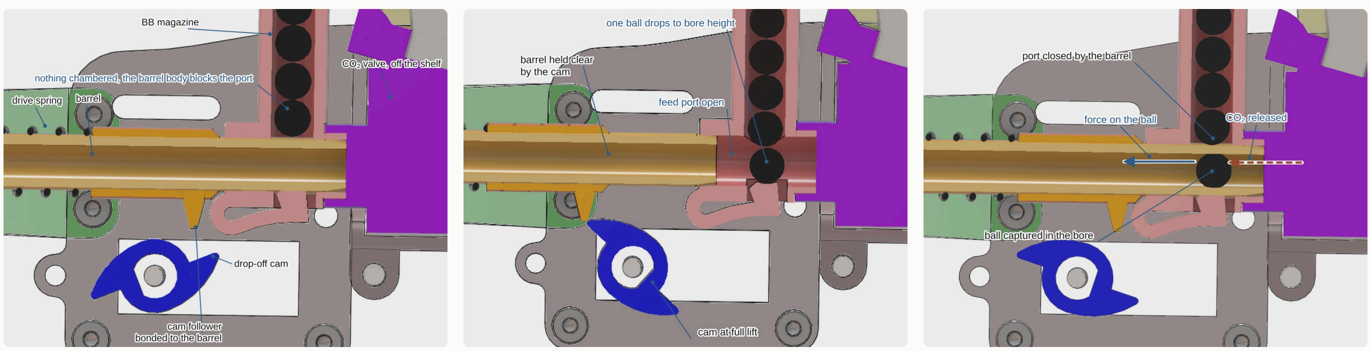


The three limits of travel, measured on the barrel centreline with neutral taken as zero. +34.5°, 0° and −24.5°. The ends are set by the linkage geometry reaching its limit.

05 Firing sequence

The launcher cocks, feeds and fires on one stroke of the barrel. Barrel position opens the feed port and trips the valve, so there is no feed actuator or trigger. A drop-off cam loads the spring over most of a revolution and releases it in milliseconds, so rate of fire is set by cam speed.

The CO_2 valve is off the shelf. The cam follower is currently bonded to the barrel, and the next revision replaces the bond with a groove and retaining ring.



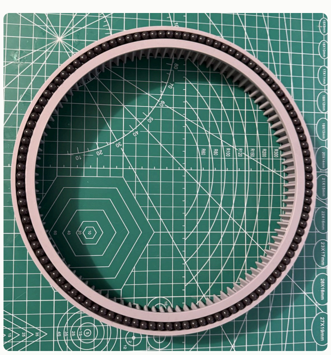
At rest the barrel is home against the valve and its body blocks the magazine port. At full cam lift the port is open and one ball has dropped to bore height. At discharge the port is closed, the ball is captured in the bore, and CO_2 drives it down the barrel.

06 Prototype findings and revisions

Friction. The first yaw bearing scraped. I changed it to airsoft BBs in a ground race. Adjacent balls still rub, so a cage is next.

Herringbone teeth. Smoother torque at low slew rate than a spur, and opposed helices cancel the axial thrust that would lift the plate off its race.

Print orientation over part count. Splitting the one piece firing mechanism let each part be oriented for finish and hole accuracy. More parts, easier assembly.



Status. Electronics not yet mounted (ESP32 in the ring gear bore, ESP32-CAM at the muzzle). No tracking or accuracy figures yet because none have been measured.