CF-01 | HIGH-SPEED CAM–FOLLOWER REDESIGN

Investigation of predicted contact loss and robustness-constrained design screening

STAGE 10 • Consultancy-grade technical case study • 11 November 2026

Evidence classification: synthetic inputs (S) | calculated predictions (C) | numerical verification (V) | proposed analysis/test (P)

EXECUTIVE SUMMARY — SIX-POINT OVERVIEW

01 THE CHALLENGE

Increase a spring-return cam–follower mechanism from 200 to 260 rpm without losing contact or creating unacceptable durability risk.

02 KEY OBSERVATION

At 260 rpm, the original prescribed motion requires a minimum cam force of −25.52 N; unilateral contact cannot supply tension.

03 ROOT CAUSE / MECHANISM

A reduced-order model predicts separation and re-impact. This is a plausible model mechanism, not a validated industrial root cause.

04 ENGINEERING SOLUTION

Candidate F combines 2.0→1.4 kg effective mass, 65→85 N preload and a fifth-order polynomial cam law.

05 RESULTS & EVIDENCE

Candidate F: +36.17 N nominal and +16.26 N worst-corner force margin; nominal Hertz pressure 214.37 MPa. No FEA or tests.

06 KEY LEARNING

Contact feasibility must be checked with unilateral dynamics; final reliability also depends on structural stiffness, friction, bearings and validation.

EXECUTIVE DECISION | Candidate F is the preferred analytical development concept; Candidate G is the fallback. Production release: NO-GO until assembly, dynamics, contact durability and physical validation are substantiated.

1 | Problem, scope and assumptions

A synthetic packaging mechanism uses a radial disc cam to drive an in-line translating roller follower against a return spring. The original 200 rpm operating point is assumed acceptable; the design objective is 260 rpm. The study investigates speed-dependent contact loss and redesign options without representing an actual client failure.

Parameter	Original	Candidate F	Fallback G
Effective moving mass	2.00 kg	1.40 kg	1.40 kg
Spring preload	65 N	85 N	90 N
Spring stiffness	1,000 N/m	1,000 N/m	1,000 N/m
Viscous damping	8 N·s/m	8 N·s/m	8 N·s/m
Cam law	Cycloidal	3–4–5 polynomial	3–4–5 polynomial
Lift / rise / return	30 mm / 90° / 90°	Same	Same
Base / roller radius	50 / 12 mm	Same	Same

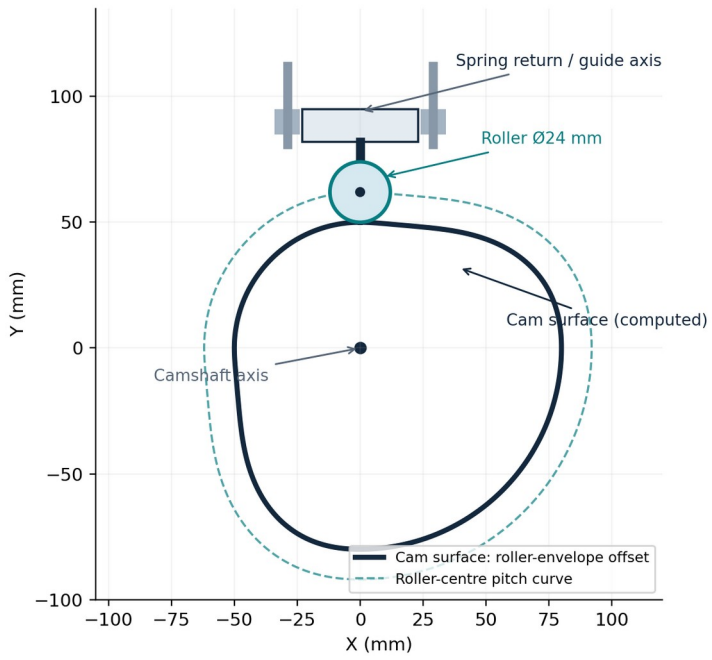


Figure 1. Mathematically generated Candidate F cam contour and pitch curve, with the roller shown at low dwell. Carriage and guide details are conceptual, not dimensionally released.

2 | Governing kinematics and unilateral dynamics

For a 90° rise, let $u = \theta/\beta$, $\beta = \pi/2$, and $h = 0.03$ m. The original cycloidal law is $f(u) = u - \sin(2\pi u)/(2\pi)$. The redesigned polynomial law is $f(u) = 10u^3 - 15u^4 + 6u^5$. Rise displacement is $s = hf$; return is $s = h(1-f)$. High and low dwells occupy the remaining 90° sectors.

At constant shaft speed $\omega = 2\pi n/60$: velocity $\dot{s} = \omega\, ds/d\theta$, acceleration $\ddot{s} = \omega^2\, d^2s/d\theta^2$. The generalized cam force needed for prescribed motion is $Q_{req} = m\ddot{s} + c\dot{s} + ks + F_0$. Frictionless unilateral contact requires $Q \geq 0$, gap $g = x - s \geq 0$ and $Q g = 0$.

If Q_{req} becomes negative, the cam cannot pull the follower. During free flight: $m\ddot{x} + c\dot{x} + kx + F_0 = 0$. Recontact is detected when g returns to zero with closing relative velocity. The reported reference impact energies assume a one-dimensional, perfectly inelastic relative-velocity law ($e = 0$), not a compliant 3D impact simulation.

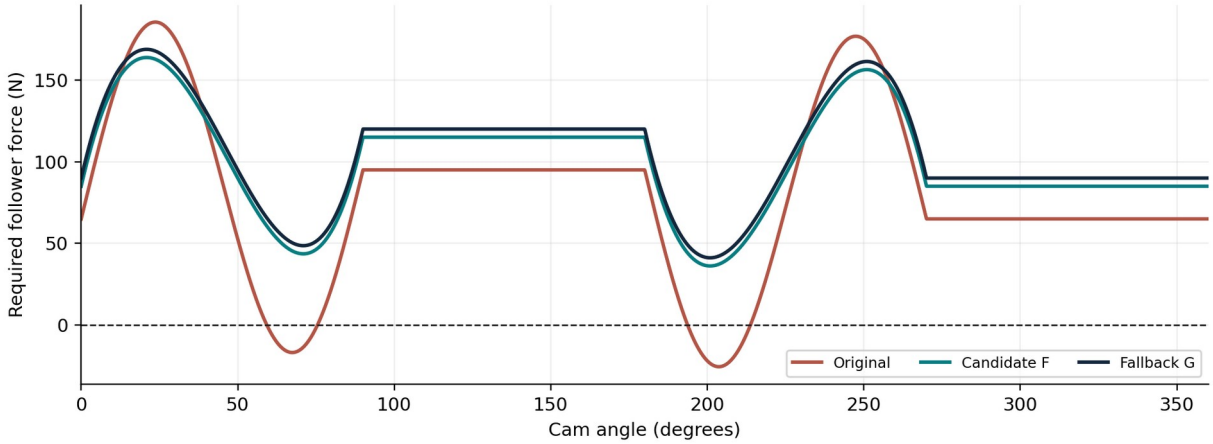


Figure 2. Independently recalculated constrained-contact force histories at 260 rpm. Negative values are physically inadmissible for maintained unilateral contact.

3 | Contact-loss chronology and numerical checks

The original design has Qmin ≈ +21.54 N at 200 rpm and −25.52 N at 260 rpm. The onset of negative constrained force is approximately 229.4 rpm for the synthetic baseline.

Event	Separation angle	Recontact angle	Closing speed	Relative energy
1	59.297°	92.015°	0.2052 m/s	0.04212 J
2	193.704°	235.077°	0.3651 m/s	0.13332 J

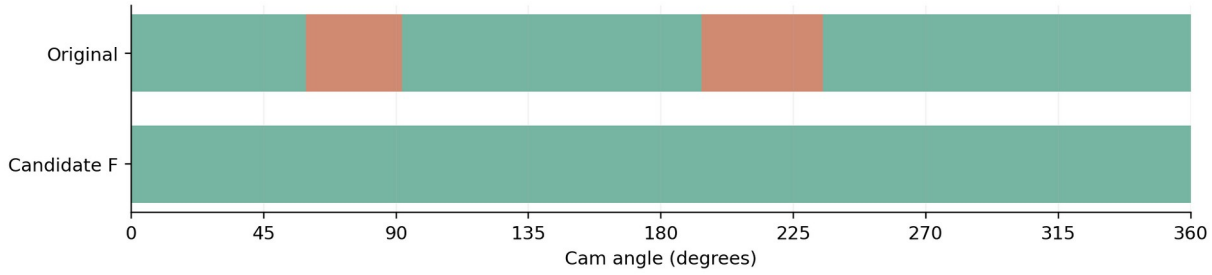


Figure 3. Model-predicted contact intervals at 260 rpm. Green: prescribed contact; terracotta: free flight. Candidate F is shown under nominal rigid-model conditions.

The reference free-flight equation admits a closed-form underdamped oscillator solution. Independent root-finding using that solution reproduces the event chronology. Free-flight energy satisfies $d/dt(½m\dot{x}^2 + ½kx^2 + F0x) = -c\dot{x}^2$. Earlier residual checks were small, but a full documented solver convergence, restitution and rebound sensitivity study remains open.

4 | Cam profile, contact force and Hertz screening

For an in-line translating roller follower, pitch radius r = Rb + Rroller + s. The pitch curve is p(θ) = [r sinθ, r cosθ]; the actual cam contour is its inward normal offset by roller radius. Pressure angle satisfies $\tan|\alpha| = |r'|/r$. Signed pitch curvature is $\kappa = (r^2 + 2r'^2 - rr'')/(r^2 + r'^2)^{(3/2)}$. Local offset regularity requires $1 - R_{roller}\kappa \neq 0$.

The calculated minimum local offset factors are 0.755 for the original law and 0.765 for the polynomial. These sampled checks do not prove global non-self-intersection, cutter access or assembly clearance.

Under maintained frictionless contact, normal force N = Qreq/cosα. With E = 210 GPa and ν = 0.30 for both bodies, E* = 115.38 GPa. For ideal elastic line contact, peak pressure $p0 = \sqrt{[N E^*/(\pi L R^*)]}$, with nominal width L = 12 mm and equivalent local radius R*. The result excludes edge loading, crowning, lubrication, misalignment and impacts.

Metric at 260 rpm	Original	Candidate F	Fallback G
Minimum generalized force	−25.52 N	+36.17 N	+41.17 N
Maximum generalized force	185.52 N	163.83 N	168.83 N
Peak normal force	195.66 N	171.34 N	176.61 N
Peak pressure angle	26.74°	25.32°	25.32°
Peak nominal Hertz pressure	226.64 MPa	214.37 MPa	217.78 MPa

Important: peak normal force and peak Hertz pressure occur at different cam angles. For the original design, constrained-motion pressure is a screening quantity; actual re-impact force cannot be inferred from it.

5 | Redesign development and robustness

The redesign combines reduced inertia, increased preload and a polynomial motion law. Increasing preload alone improves retention but raises continuous contact load; reducing mass alone lowers inertia but may compromise stiffness; the motion-law change alone does not remove predicted separation at 260 rpm.

Variant	Minimum force at 260 rpm	Peak nominal pressure
Original cycloidal, 2.0 kg, 65 N	−25.52 N	226.64 MPa
Preload only: 2.0 kg, 95 N	+4.48 N	244.96 MPa
Mass only: 1.5 kg, 65 N	+2.67 N	209.86 MPa
Polynomial only: 2.0 kg, 65 N	−14.89 N	219.08 MPa
Candidate F: 1.4 kg, 85 N	+36.17 N	214.37 MPa
Fallback G: 1.4 kg, 90 N	+41.17 N	217.78 MPa

The synthetic uncertainty envelope varies mass ±5%, preload ±10%, stiffness ±10%, speed 255–265 rpm and damping 0–12 N·s/m. A 32-corner evaluation gives the results below. These bounds are not measured tolerance distributions or an exhaustive physical reliability assessment.

Design	Worst-corner minimum force	Worst-corner peak pressure
Original	−47.33 N	238.65 MPa
Candidate F	+16.26 N	226.15 MPa
Fallback G	+20.76 N	229.73 MPa

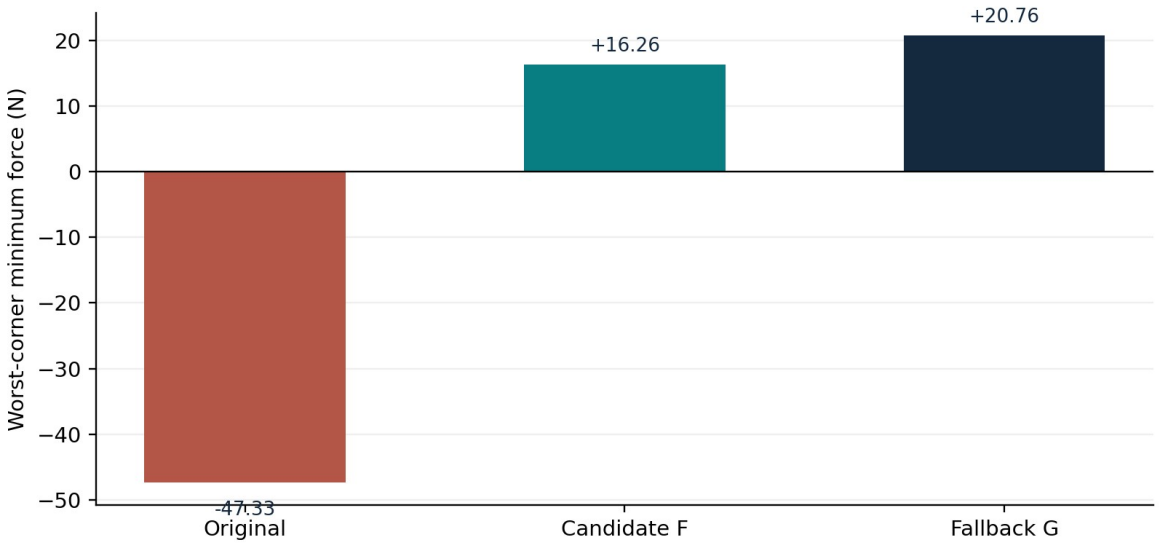


Figure 4. Worst-corner force-retention screening. Positive values indicate prescribed-motion feasibility within the evaluated model.

F is preferred for its lower nominal contact pressure; G provides 4.5 N more worst-corner force margin for moderately higher pressure. Neither is globally optimal, and the envelope excludes realistic combined process loads, guide friction, runout and structural feedback.

6 | Structural assembly, bearing and tribology

A reference assembly consists of two linear guides, a translating carriage, a roller and bearing, a pin, return spring, and process-tool attachment. The Stage 7 provisional 1.4 kg component allocation is not CAD verified. Effective inertia must be derived consistently from $T = \frac{1}{2}\,\dot{q}^{T}M(q)\dot{q}$; roller rotation and any reflected inertias cannot be treated as an arbitrary mass allowance.

Earlier illustrative cantilever and two-beam calculations gave differing stiffness and frequency estimates. Neither represents the complete assembly. Required FE outputs include relative tool-to-contact displacement, pin and bearing-seat stresses, guide reactions, modal participation, joint compliance and dynamic amplification. No such FE results exist.

A compliant cam–roller model must resolve finite-width contact, local curvature, pin/bearing compliance, guide support and suitable normal/tangential contact laws. The bearing study needs an actual selected bearing, speed and load spectrum, internal clearance, fits and lubrication. No wear rate, rolling-contact fatigue life, bearing rating life or endurance result can be inferred from the present synthetic data.

7 | Competing mechanisms and verification evidence

Hypothesis	Status	Discriminating evidence
Separation and re-impact	Model-supported, not validated	Encoder + displacement + load/acceleration
Excessive rolling-contact stress	Open	Damage morphology and verified load spectrum
Structural resonance	Open	Modal and operating vibration tests
Guide friction / misalignment	Open	Guide loads, alignment, hysteresis
Bearing damage	Open	Bearing specification and inspection
Lubrication / wear	Open	Roughness, lubricant, temperature
Cam profile / runout error	Open	Profile metrology and runout measurements

8 | Requirements-to-evidence matrix

Requirement / claim	Available evidence	Disposition
260 rpm kinematic motion	Defined analytical motion law	Analytical pass
Original contact loss at 260 rpm	Reproduced force and event roots	Model-supported
F nominal contact retention	+36.17 N calculation	Model pass
F previous envelope margin	+16.26 N corner study	Screening pass
Cam pressure angle <30°	25.32° calculated	Screening pass
Global cam manufacturability	No global CAD review	Open
Effective moving mass ≤1.4 kg	Provisional mass budget	Open
Assembly stiffness and modal response	No representative FEA/test	Open
Combined disturbance robustness	No coupled process/friction model	Open
Contact and bearing durability	No qualified material/bearing data	Open
Validated physical root cause	No field or test observations	Not claimed
Service-life improvement	No fatigue/endurance evidence	Not claimed

9 | Numerical verification and experimental qualification plan

Numerical: (1) compare closed-form free flight with ODE results; (2) refine event-detection time steps and tolerances; (3) examine restitution and repeated impacts; (4) verify energy/work accounting; (5) check complete cam offset for self-intersections; (6) benchmark contact models against appropriate elastic solutions; (7) conduct structural and contact mesh-convergence studies. These are planned gates, not completed FEA.

Experimental: establish physical mass and spring properties; inspect the cam contour and runout; record encoder-synchronized carriage displacement, acceleration and loads during controlled speed sweeps; identify structural modes; inspect bearing temperature, lubrication and surface condition; then undertake an approved endurance programme. Define functional tolerances and safety shutdown thresholds before testing.

10 | Skeptical technical audit and disposition

Finding	Severity	Disposition
Model-predicted separation is not a validated industrial root cause	Critical	Explicit limitation
1.4 kg effective mass and stiffness are unverified	Critical	CAD + kinetic energy + FEA required
No fatigue, wear or bearing-life evidence	Critical	No life claim permitted
Combined process load and guide friction omitted	High	Coupled sensitivity required
One-dimensional impact law not local damage model	High	Compliant normal contact required
Global cam manufacturability not closed	High	CAD/tolerance check required
Event solver convergence and rebound not complete	High	Numerical V&V required
Selected F not globally optimal	Medium	Describe as candidate only

FINAL ENGINEERING DECISION. Reject the original configuration for the stated 260 rpm rigid-model continuous-contact requirement. Advance Candidate F as the preferred analytical development concept, with Candidate G retained as a preload-margin fallback. Production release is NO-GO pending physical assembly substantiation, coupled dynamics, contact and bearing durability, and experimental validation.

PORTFOLIO CLAIM BOUNDARY. This is a synthetic, reproducible physics-led design investigation. It does not describe a completed client engagement, validated repair, measured industrial root cause, fatigue-life improvement or commercial result.

Reproducibility note

Accompanying file: CF01_reproducible_calculations.py. Requires NumPy and SciPy. Reproduces constrained force, pressure angle, Hertz screening, uncertainty corners and analytical free-flight events. All results remain conditional on the documented synthetic assumptions.