

Machine-Tool Construction Practice Applied to a DIY Filament 3D Printer: Design, Failure Analysis, and Rebuild of “Boaz”

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Abstract—Do-it-yourself filament 3D printers inherit their construction methods from the RepRap project, which optimised for self-replication from cheap parts rather than for accuracy. We test whether five practices from machine-tool building transfer to this class of machine: exact constraint, interference fits on locating features, kinematic couplings, stiffness before squareness, and metrology performed during assembly. The test article is “Boaz,” a CoreXY printer that was built, run in production, diagnosed, and then rebuilt. Both builds share the same design, kinematics, and most components and differ in how well the precision practices were executed, so the first build serves as a control. Eight performance criteria were fixed before construction. The first build failed at least four: locating pins fitted with clearance instead of interference, rails aligned to printed jigs good to 0.2 mm, and a bed whose mesh envelope reached 1.6 mm, forcing an operator to jog Z offset on every first layer. The rebuild met seven of the eight, with runout at every pressed idler below 0.001 in, a bed mesh envelope of 0.2 mm over a 100 × 100 grid, and roughly 100 consecutive production plates with no first-layer intervention; the Y-axis resonance sweep capped acceleration at 9500 mm s⁻² against a 20 000 mm s⁻² target. Over a ten-day, 1000-part PETG contract it held 0.2 mm functional tolerance on threaded parts at about 50 parts per day, against 60 for a \$300 commercial machine alongside it. The throughput comparison was lost; the accuracy claim was not. We report eleven construction rules with the observation that earned each.

Index Terms—Additive manufacturing, fused filament fabrication, CoreXY, precision engineering, kinematic coupling, interference fit, metrology, input shaping, Klipper

I. INTRODUCTION

Almost every construction convention in hobby 3D printing descends from RepRap: printed plastic structural parts, 20 × 20 mm extrusion frames, idlers mounted on M5 screws, rails aligned by eye. Adrian Bowyer founded that project in 2005, four years before the controlling Stratasys patent on the process expired [1], [2], and its goal was a machine that could manufacture a substantial fraction of its own parts from the cheapest available components. Those conventions answer that question well. They were never an answer to a question about dimensional accuracy or speed.

Ask the accuracy question and a different set of answers falls out, almost all of them worked out by machine-tool builders decades before anyone extruded plastic through a nozzle: exact-constraint design, kinematic couplings, interfer-

ence fits on locating features, and metrology treated as a construction step rather than an inspection step [3], [4].

A. Claim

Five construction practices make machine tools accurate: exact constraint, interference fits on locating features, kinematic couplings, stiffness before squareness, and metrology performed *during* assembly rather than after it. All five transfer to filament 3D printers, and a DIY machine built with them will outperform a DIY machine of comparable cost built with conventional hobby practice.

The claim does not say a DIY machine beats a commercial one on cost, time to first print, or throughput per dollar. Section VII reports it losing a production race to a factory-built machine costing a fraction as much. The claim is about method, and the comparison class is other machines a reader might build.

B. Test article and control

The machine is called Boaz (Fig. 1). Design began in October 2023 and the first print came off it in 2024. It is a CoreXY machine on a two-tier 20 × 40 mm extrusion frame, with a belted three-actuator Z axis carrying a kinematically coupled cast-aluminium tooling plate, an SLM-printed aluminium toolhead fed by a remote CPAP blower, and Klipper firmware [5] on a BIGTREETECH Octopus Pro.

No choice in that list is novel on its own. What gives the project evidentiary value is that the machine was built twice. The first build followed the right principles with the wrong technique: dowel pins in clearance-fit holes, rails aligned with jigs accurate to 0.2 mm, a heatbed chosen on price. It produced grainy walls, parts that would not assemble, and first layers that required an operator. The second build differs in execution, not in design principles, kinematics, or most components. The first build is therefore the control, and the comparison isolates the variable under test.

Eight criteria were fixed before construction, each with a number and a defined measurement procedure, because criteria written afterwards are indistinguishable from excuses. Table I states them with the outcome for the rebuilt machine. Two are worth defending. Print-success percentage is nearly meaningless without a matched part mix, so the reliability



Fig. 1. Boaz in its final configuration: two-tier 20 × 40 mm gusset-joined frame, CoreXY motion on a carbon-fibre X beam, a three-actuator belted Z axis carrying a kinematically coupled ATP-5 tooling plate, and a folding polycarbonate enclosure.

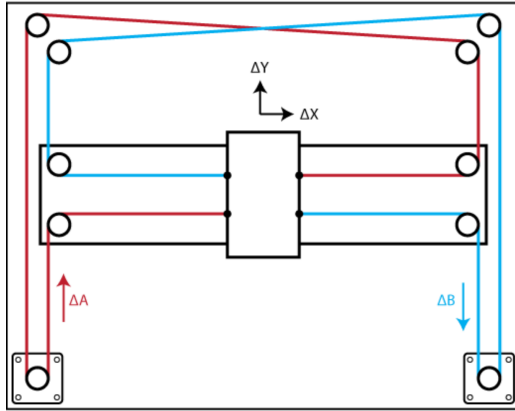


Fig. 2. CoreXY. Two crossed belts driven by two stationary motors. Neither motor is assigned to an axis; X and Y motion are produced by their sum and difference.

criterion is instead whether a human must stand over the first layer, which is a countable event. And total indicated runout at the pressed idler pins was the one construction metric that predicted surface finish, for the reason Section II-B gives.

II. DESIGN BACKGROUND

Table II summarises the design choice made for each subsystem and the constraint that forces it. The rest of this section expands the four that the results in Section VII turn on.

A. Motion architecture

CoreXY [11] keeps both motors on the frame without giving up independent XY motion (Fig. 2):

$$\Delta X = \frac{1}{2} (\Delta A + \Delta B), \quad \Delta Y = \frac{1}{2} (\Delta A - \Delta B) \quad (1)$$



Fig. 3. An idler mounted on an M5 screw. The screw's shank is undersized relative to nominal, so the bearing bore is a clearance fit on it and the idler's axis can sit anywhere within that clearance.

where ΔA and ΔB are the two motors' belt displacements. Removing a 300 g motor from a 700 g gantry raises achievable acceleration by roughly 75 % before any other change.

Three costs land on precision. The crossed routing needs at least eight idlers, each a rotating element whose axis position and perpendicularity enter the belt path geometry. A difference between the two belt tensions applies a net moment and rotates the gantry about the vertical axis, *racking*, which prints as skew, so the belts must be tensioned together, iterating and checking gantry squareness, and verified with an artifact that measures diagonals. Equal frequencies on unequal spans do not mean equal tensions. And (1) smears every mechanical fault across both axes, so a single sticky idler produces artifacts along diagonals. The mass argument justifies *stationary motors*, not CoreXY specifically; Section IX returns to that.

B. Precision assemblies

A metric screw's manufactured shank runs undersize, commonly by about 0.2 mm on an M5, so that the thread can be rolled and the screw passes through a clearance hole. A bearing with a 5.00 mm bore on such a screw therefore has up to 0.2 mm of diametral clearance (Fig. 3), and it does not stay centred: belt tension pulls it against one side, and which side depends on the direction of the net belt force, which changes as the toolhead moves.

The usual defence is that a fixed offset disappears into the coordinate system once the machine homes. It fails twice. The error is not fixed: it depends on toolhead position and direction of travel, which is a backlash signature appearing as a step at every reversal. And the errors accumulate rather than cancel. Treating the idlers as random and independent, the most generous assumption available, a 0.2 mm per-idler bound over eight idlers gives an expected belt-path error of order $\sqrt{8} \times 0.1 \text{ mm} \approx 0.28 \text{ mm}$, which (1) halves to about 0.14 mm at the nozzle. Layer stacking artifacts become visible at a few tens of microns, so the mechanism is several times larger than the defect it produces.

The correct locating element is a hardened ground dowel pin pressed into a reamed hole. The hole is reamed *under*

TABLE I

SUCCESS CRITERIA FIXED BEFORE CONSTRUCTION, WITH THE MEASUREMENT PROCEDURE AND THE OUTCOME FOR THE REBUILT MACHINE. TARGETS WERE SET TO MATCH OR BEAT COMPETENT ENTHUSIAST-GRADE DIY MACHINES. THE FIRST BUILD FAILED THE FOUR MARKED †.

Criterion	Measurement	Target	Result (rebuilt machine)
Geometric accuracy†	Califlower artifact; X, Y and both diagonals	Skew $< 0.1^\circ$; axis error $< 0.2\%$	Met after firmware correction; pre-correction angle and axis scale not recorded
Functional tolerance†	Printed threaded lid engages and turns freely	≤ 0.2 mm	Met; the first build could not print the part
Bed flatness as the firmware sees it†	Klipper bed mesh over the full print area	Envelope ≤ 0.2 mm	Met: 0–0.2 mm over 100×100 at 70°C
First-layer autonomy†	Consecutive production plates started without jogging Z offset	≥ 10 plates	Met: ≈ 100 plates
Locating-feature runout	Dial indicator on each pressed idler, several angular positions and heights	< 0.001 in TIR	Met on every assembly retained
Dynamic behaviour	Klipper resonance sweep, accelerometer on toolhead	Single dominant peak; recommended accel. $\geq 20\,000\text{ mm s}^{-2}$	Not met: X one peak, $20\,900\text{ mm s}^{-2}$ (MZV); Y two peaks, 9500 mm s^{-2} (EI)
Speed/quality trade	#SpeedBoatRace 3DBenchy, model unmodified, race slicer settings	≤ 20 min at acceptable surface quality	Met: 16 min, first attempt
Sustained production	Parts per day over a multi-day contract	≥ 50 per day	Met at ≈ 50 ; comparison machine did 60

TABLE II

SUBSYSTEM DESIGN CHOICES AND THE GOVERNING CONSTRAINT. ENTRIES MARKED † ARE EXPANDED IN THE TEXT.

Subsystem	Governing constraint	Choice
Frame section	$A \approx 10\text{ N}$ toolhead reaction reversing tens of times per second prints as banding that scales with speed; second moment of area goes as the cube of depth.	20×40 mm extrusion; 40×40 mm would have been correct on the uprights, which the Z actuators load in bending.
Frame joints	Squareness needs an adjustable joint. Blind joints inherit the saw cut's angle, and three bolt axes intersect at a box corner.	Gusset plates: the fastener clearance is the adjustment range.
XY architecture†	$a = F/m$ with torque fixed; a gantry-mounted NEMA 17 is 40–55 % of the moving mass.	CoreXY, both motors on the frame.
Idler location†	An M5 shank runs ≈ 0.2 mm undersize, giving direction-dependent play that accumulates over 8–12 idlers.	Ground dowel pins in reamed holes, 0.05 mm interference.
Locating assemblies	Printed plastic is anisotropic, creeps under bolt load near T_g , and expands $3\text{--}4\times$ faster than aluminium.	Laser-cut plates on machined metal standoffs [6], [7].
Belt width†	$k = E_{\text{eff}}A/L$, and CoreXY spans run several times the axis travel.	9 mm GT2, not 6 mm.
Linear guides	Moment load from a nozzle strike sizes the rail; preload trades stiffness against friction at speed.	MGN12H [8], light preload, thinner grease than the datasheet's NLGI 1 [9].
Gantry beam	Load is torsional. A closed section gives $J \approx 4A_m^2 t/s$ with enclosed area squared; an open one gives $J \approx \frac{1}{3} \sum bt^3$. T-slots leave extrusion behaving nearly as an open section.	Closed tube. Steel first, carbon fibre after the rebuild, with an internal standoff at every rail fastener.
Toolhead	Flow bounds print time; the toolhead is the only mass moving in X; a centre-of-mass offset d turns drive force F into a moment Fd .	SLM aluminium body, geared extruder, long-melt-zone hotend, centroid balanced with a brass plate.
Part cooling	A concentrated jet suits fine features; large layers stay hot over a wide region.	Remote CPAP blower on the frame; distributed-outlet duct after the rebuild.
Z drive	Ball screws are priced in accuracy grades, react misalignment into the frame, and cost ≈ 70 mm of travel. Belts cannot self-lock.	Belted Z with a 1:5 belt reduction, sized against the bed's gravity torque; margin not rechecked since the plate change.
Bed location†	A rigid body has six degrees of freedom; bolting a plate at more points injects stress it absorbs by bending.	Three-point Maxwell kinematic coupling, one ball per driven actuator.
Bed plate†	A thin plate cannot be flattened by mounting it to something stiffer; it bulges between fixings.	Cast, stress-relieved ATP-5 tooling plate [10].

nominal, so material is displaced as the pin is pressed home and contact pressure holds it by friction. Boaz's rebuild uses a 4.95 mm reamer against a measured 5.000 mm pin, a nominal 0.05 mm interference. That is a heavy press by ISO 286 standards: the preferred H7/p6 fit allows at most 0.02 mm at this diameter [12], and one percent of the diameter yields the aluminium around the hole rather than merely straining it.

Interference is not a free parameter. Too little and the pin is loose. Too much, and the press force is high enough that the pin goes in crooked, converting a positional error into an angular one. The angular error is worse: the idler's runout then varies along its face, so the belt sees a different path length at its top edge than at its bottom, and the idler steers the belt off the pulley. The way to stay inside the window is to control both sides, reamed hole and measured pin, rather than to press harder. Anaerobic retaining compound bonds a pin where it sits, so it supplements a correct fit rather than substituting for one. The measurable outcome is one number: total indicated runout at each idler below 0.001 in.

C. Locating the bed

Z does not merely position the bed; it orients it, and an orientation error of a tenth of a degree is a first-layer failure across the plate. The conventional design bolts the bed rigidly to two or three carriages, which is over-constraint: bed, carriages, rails and frame form a closed kinematic loop, and any inconsistency in it is absorbed by elastic deformation of the bed. The bed is bent by its own mounting in a pattern that cannot be adjusted out, and the firmware compensates for a topology the machine created itself.

The alternative is *exact constraint*: a rigid body has six degrees of freedom, so locate it with exactly six independent contact points and no more [4]. A Maxwell kinematic coupling does this with three balls in three vee grooves radiating from the centre. Opened up, with each ball-and-groove joint on its own driven Z actuator, it gives three things at once. The bed rests on three points and is stressed by nothing but its own weight. Tilt becomes an adjustable degree of freedom, so Klipper's `Z_TILT_ADJUST` [13] can measure it with the probe and correct it mechanically before every print, rather than smearing a physical error into a mesh. And as the plate heats, each ball slides along its vee, accommodating thermal growth without generating stress.

A coupling that locates a plane perfectly still cannot help when the thing it locates is not planar. A PCB heatbed is thin, warps with thermal cycling, and is not flat when it arrives, and a plate that thin cannot be flattened by mounting it to something stiffer: when plate and rack disagree, the plate wins locally and bulges between the fixings. Flatness has to be bought in the plate.

III. DESIGN METHOD

A CoreXY motion system is geometrically a plane figure: belt segments, idler centres, pulley centres, rail lines. Everything that determines whether the machine works is visible in that plane and invisible in an assembly of solids: whether

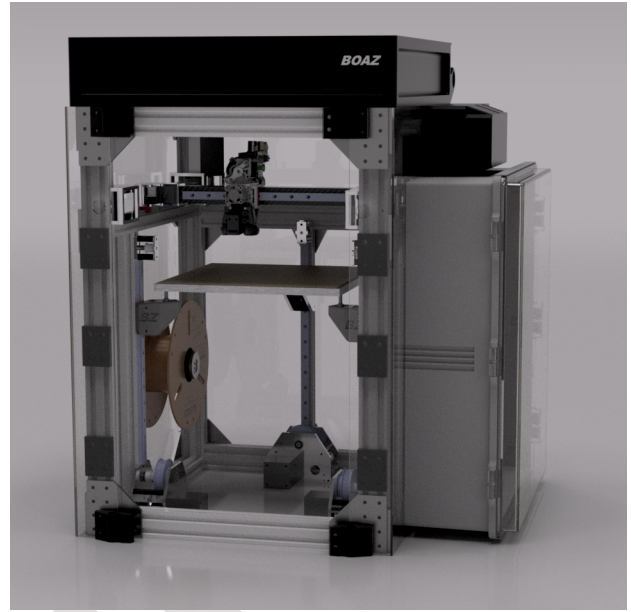


Fig. 4. The CAD assembly at the close of the design phase. Every plate visible here is an extrusion from the 2D master sketch, so belt runs, idler centres and rail lines are inputs to the solid model rather than quantities measured out of it.

belt runs are parallel to the axes they drive, whether the two loops are symmetric, whether the tensioner has travel. The first artifact was therefore a fully constrained 2D sketch holding every centre, every belt run and the envelope of every plate, dimensioned against the frame's inside dimensions. Plates are extruded from it, so when something moves it moves in the sketch and everything downstream follows. In a solid-first workflow the belt path is an emergent property of a dozen independently placed components and checking it means measuring the assembly; here it is an input. Fig. 4 shows the assembly the sketch produces.

Two features follow from Section II-B and are cut into the plates at this stage. Dowel-pin holes were drawn at their finished interference size, on the reasoning that this keeps the fabricator's tolerance out of the fit. It does not, because a cut hole carries its own tolerance whatever size is drawn; the rebuild (Section VI-C) draws them undersize and reams to dimension. Every standoff position is dimensioned from the same datum as the dowel holes, so a stack's parallelism depends on the cutting process rather than on assembly care.

The toolhead took three iterations. The first put a compact hotend under an extruder mounted above it, with dead vertical space that placed the largest single mass as far as possible from the plane of motion. The second used a lighter geared extruder rotated so its motor mass sat inboard, and a large-melt-zone hotend whose own mass pulls the centroid down. The third added the duct and a brass counterweight, after which CAD put the centroid forward of the carriage and within the plane of motion. That verification is only as good as the densities in the model: CAD packages assign a default material to every imported body, and a hotend imported as generic plastic

weighs a third of what the real part weighs.

Z design began from the bed size and three joint locations in an isosceles arrangement. The bed chosen on cost was a Prusa MK52 PCB heatbed [14], whose mounting pattern does not correspond to three kinematic points, so an intermediate rack was needed to carry the balls. That adapter is worth noticing at design time: it exists only because a component chosen on price does not present the interface the design needs, and an adapter between a cheap part and a precision requirement inherits the cheap part's compliance. What the drawings did not contain is any means of tensioning either Z belt.

IV. FIRST BUILD

This section records a machine built to the right design with the wrong technique.

A. Frame

The build began committed to blind joints, and all three predicted failures arrived in sequence. Checked against a machinist square, some joints came out square and others did not; the variable is not assembly technique but the squareness of the factory saw cut, because a zero-compliance joint reports whatever angle it was cut with. Tapping the deep blind holes the joint needs loaded the tap with chips it could not clear, and it seized. And at the three-way corners the bolt axes intersect, so the extrusions tying the two side sub-assemblies together required gussets regardless. The sub-assemblies were rebuilt with gusset plates, used more generously than a strength calculation would justify, because frame flex under gantry acceleration is expensive to diagnose and nearly impossible to retrofit.

Squaring then ran on granite: assemble finger-tight, reference a machinist square into each internal corner until the gap closes along the full blade, tighten in a cross pattern re-checking after each pass because tightening moves the joint, and re-check after the box is closed. Two details separate this from going through the motions. The granite supplies the reference plane, and squaring on a surface that is not flat squares to the wrong reference. And the square must be of known machinist grade, because the error being chased is about a tenth of a degree over 300 mm and a hardware-store square's own error is larger than that.

B. Dowel pins, wrong in three ways

The hole was reamed oversize. Motor-mount holes were reamed to 5 mm, nominal and matching the pin, on the understanding that a press fit meant a hole the same size as the pin. It does not. Execution made it worse: the reamer ran in a drill press with roughly 0.05 mm of spindle runout, and a reamer that is not running true cuts a hole larger than itself.

The pin was undersize. The pins were sourced cheaply and measured about 0.04 mm under nominal, so the assembly had clearance where it needed interference. Retaining compound was tried and failed: it bonds a pin where it sits, and in a clearance hole the pin sits off-centre. The fix used was a strip of plastic bag about 50 μ m thick wrapped around the pin. It

worked in the sense that the machine could be assembled. A polymer shim in a locating joint creeps under load in a heated chamber, is not uniform around the circumference so it adds its own eccentricity, and is unrepeatable.

Other joints went the other way. Several were pressed at roughly 0.2 mm of interference, four times the specified value, on the theory that if some interference is good then more is better. That much on a 5 mm pin needs enormous force even with an arbor press, and a pin needing that much force does not go in straight. The pin ends up tight but not perpendicular, so the idler runs with runout that varies across its face and steers the belt. Not one of the first build's press fits landed between the two extremes.

The tensioner compounded it by design rather than by execution. Its idler pin was given a slot to slide in, deliberately oversize by 0.1 mm so that it could not seize. The clearance that prevents seizing is exactly the clearance that lets the idler shift under belt load, which reintroduces a screw-mounted idler at the one position in the loop that sets belt tension.

C. Rails, Z, and the frame that moved

With the gantry on the Y rails, two rails and one rigid beam, the frame visibly bent inward and outward as the gantry traversed Y. Reseating the rails repeatedly reduced the effect without eliminating it, and the reason is instrumentation rather than patience: the alignment jigs were printed parts holding about 0.2 mm, and the error being chased is smaller than that. You cannot align to a tolerance finer than your reference. The X beam was a 1 in steel tube fitted without internal standoffs, so the rail's fasteners clamp the top wall toward the cavity and bend the surface the rail is meant to reference.

The three Z actuators were built in PLA with aluminium replacements planned. The intent was to get the machine moving; the effect was to put a compliant element in the one axis whose entire job is to hold a position. Belt tension in a Z actuator is a standing load, present whenever the machine is assembled, and the printed body creeps under it, so a bed whose height depends on how long the machine has been standing cannot be calibrated once. The replacements did not arrive until after all the production work in Section V, so that entire body of data was collected on a component known at design time to be wrong.

Neither Z belt had a tensioner, because the CAD did not draw one. The main belt is clamped by printed teeth with no adjustment at all, so the only way to tension it is to stretch it by hand and force it into the teeth under load; the resulting tension differs across the three actuators and cannot be measured or reproduced. A tensioner is a few plates, a screw, and an afternoon in CAD. Omitting it does not save that afternoon; it spends the afternoon repeatedly at assembly. The MK52 bed was mounted at five of nine available holes, assuming the PCB was stiff enough to bridge the unsupported regions. It is not.

D. Electronics and firmware

Table III is the first build's power inventory at 24 V. The bounding sum exceeds the supply that was fitted, deliberately:

TABLE III

FIRST-BUILD POWER INVENTORY AT 24 V. MOTOR FIGURES TAKE PROGRAMMED PEAK CURRENT TIMES BUS VOLTAGE, WHICH IS A BOUND RATHER THAN AN OPERATING POINT.

Load	Power	Basis
Heated bed (MK52)	220 W	manufacturer rating
XY motors (2×)	72 W	1.5 A peak each
Z motors (3×)	108 W	1.5 A peak each
Hotend heater	100 W	sized for the large melt zone
Extruder motor	48 W	2 A peak
CPAP blower	50 W	measured at full speed
Controller, host, fans	25 W	allowance
Bounding sum	623 W	

it assumes every load at maximum simultaneously, which does not occur, since heaters draw their duty-cycle average once at setpoint and motors reach peak current only during acceleration. The realistic worst case is both heaters in initial heat-up with the motors idle, about 345 W. The machine uses a Meanwell LRS-600-24 [15], on three rules: run a single bus voltage so that one failure mode and one fusing scheme cover the machine; size against realistic concurrent load rather than the bounding sum; and buy from a manufacturer that publishes derating curves, because a nominal 600 W from an unbranded supply is not 600 W at 40 °C inside an enclosed machine. The rebuild moved two lines: the tooling plate's 350 W Keenovo mat replaced the 220 W MK52, and the XY pair left the bus for a 240 W Meanwell NDR-240-48 [16] (Sec. VI), taking the 24 V bound to 621 W (drivers run at 1 A) and the heat-up case to 475 W, 79 % of the LRS-600's rating.

The controller choice is a firmware choice. Klipper runs motion planning on a Linux host that streams precomputed step timings to a microcontroller which does nothing but execute them [5], where RepRapFirmware runs it on the board [17]. Boaz's kinematics are ordinary, so RepRapFirmware's configuration depth buys nothing; both shape and mesh, and Klipper's gain is the resonance workflow, the eddy probe, and a one-file configuration. The board is a BIGTREE TECH Octopus Pro 1.1 [18], and its socketed drivers earned their place: the machine changed driver type mid-life, which on a board with soldered drivers means a new board. The first build used TMC2209s [19], whose limitation is voltage: torque at speed is set by how fast current can be forced into the winding against its back-EMF, so a 2 A-class driver on a 24 V bus caps torque at print speed, and with it XY acceleration there. Electronics sat on an ABS sheet bonded across the bottom of the frame, which is a shelf rather than an enclosure.

Firmware bring-up went in dependency order, XY, then toolhead, then Z and probing, each subsystem proven before the next was enabled, because a machine with five unvalidated subsystems produces symptoms that cannot be attributed. `Z_TILT_ADJUST` [13] sits before meshing in the start sequence: tilt is a physical correction that removes a plane, mesh is a software correction for what remains, and meshing first means measuring a tilt about to be removed.

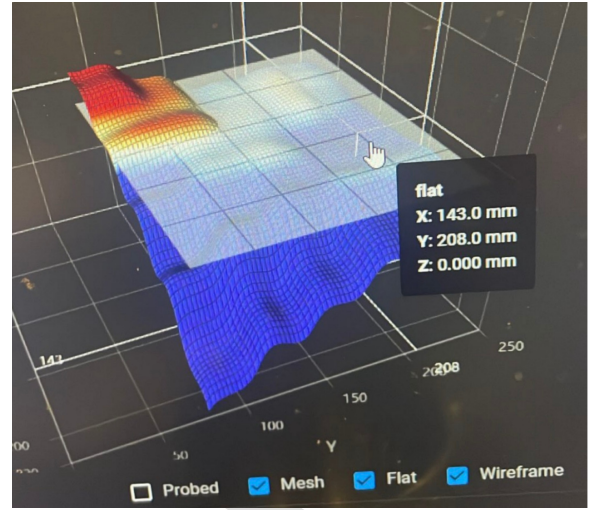


Fig. 5. Bed mesh of the MK52 as installed on the first build. Total envelope from +0.4 mm to −1.2 mm, with the largest deviations in the unsupported regions between the five mounting points.

For the mesh, Klipper's bicubic interpolation accepts dense grids where Lagrange is capped at a small one [20], and exposes a tension parameter. The first build reasoned that a badly deviating bed needs maximum compensation and set tension to its maximum. That is not what the parameter does: it controls the curvature of the interpolating surface between probed points, so a high value makes the surface swing hard between samples.

What existed at the end of the first build was a working machine whose firmware was compensating for mechanical defects rather than residual error.

V. FIRST-BUILD RESULTS AND DIAGNOSIS

The evaluation was a two-month charity printing challenge, chosen because it measures throughput, repeatability and human attention per part rather than the quality of one showpiece. Boaz printed roughly 200 toys against about 150 for a Prusa i3 alongside it. That comparison flatters Boaz: the new machine got maintenance and tuning between batches that the older one did not, and its output was held down by something unrelated to its mechanics: with the electronics on an open shelf under an open frame it was never run overnight, and a machine that cannot be left alone at night cannot use two-thirds of the available hours.

The dominant problem was the first layer, and its cause was the bed (Fig. 5). The mesh envelope ran 1.6 mm peak to peak against a first layer height of 0.2 mm, with the worst sag in the unsupported regions between the five fixings. Every first layer had to be jogged manually, with an operator applying live Z offset as the toolhead moved between regions. On a machine meant for unattended production that is disqualifying.

Two software remedies failed, instructively. A denser mesh resolves local deviation that a sparse one averages away, but the MicroProbe touches every point physically and repeats each one, so a dense mesh was not viable per-print on

production batches. Raising bicubic tension to its maximum was expected to increase compensation and did not, because tension controls the curvature of the interpolated surface between probed points and no interpolation strategy recovers a shape the probe never sampled. The general result is worth stating on its own: software compensation is interpolation between measurements, so where the mechanical error has spatial frequency higher than the sampling density no setting recovers it, and more aggressive interpolation only worsens the error between samples. Firmware compensation is for residual error, not for a defective part.

The XY system produced two failures with one cause. The belts had been tensioned sequentially, which racks a CoreXY gantry, so the X axis was skewed and every part made in that period carries it; under-tensioned belts also skipped teeth, and a skipped tooth is an unrecoverable coordinate shift. Tensioning both together, iterating and checking gantry squareness, resolved both. Asked afterwards to print PETG, two more assumptions failed: the duct was designed for concentrated cooling of fine features and over-cooled PETG into delamination, and the frame had been designed to accept panels and had gone a year without them, so contraction in open air produced warp and corner lift.

A. Diagnosis

Surface finish traces to runout in the belt path. One idler was measurably skewed on its pin after a few hundred hours, so the joint had not held location. Holes were reamed at or above nominal on a drill press with runout, pins were undersize by roughly 0.04 mm, and the resulting joints were clearance fits, some shimmed with plastic film, some pressed at 0.2 mm and therefore angular. Each affected idler contributes a position-dependent error to effective belt path length, and with eight or more those errors accumulate. This became the rebuild's top priority, because it is the defect least amenable to compensation: a warped bed can be meshed around and a slow probe replaced, but a position error that depends on direction of travel cannot be calibrated out, because there is no fixed value to calibrate.

Speed was capped by three factors: a 1 in steel tube roughly three times the mass of an aluminium one of the same section and the dominant moving mass in Y; a deliberately thick duct that pulled the centre of mass down, added mass where mass is most expensive, and still failed at cooling a wide area; and TMC2209s on a 24 V bus.

Reliability decomposes into two faults that were being blamed on each other. The plate was not flat and was not stiff enough to be flattened by its mounting, which needs a different plate. The probe was too slow to sample densely enough to compensate for anything but a smooth, low-order error, which would remain a problem even on a good plate.

Every item on that list is a construction or component defect. Not one is a defect in the design principles of Section II. That is the strongest available evidence that the principles were right and the execution was not.

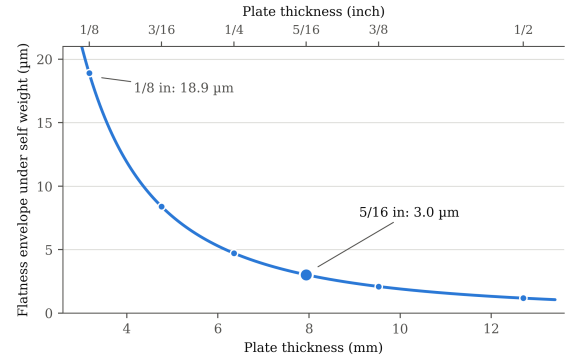


Fig. 6. Flatness envelope of a square aluminium plate on three point supports under its own weight, computed for this bed: a 300 mm plate with supports on a 120 mm circle. Markers are finite-element results; the curve is the $1/h^2$ law plate theory gives for self weight [21].

VI. REBUILD

The rebuild applies Section II deliberately, with measurement at each step. It changed execution, not design principles.

A. Bed and probe

The first response to the bad mesh was to support the PCB bed better rather than replace it: use all nine holes, shim each standoff until the plate reads flat, and let the rack define the topology. The approach needs the rack to be much stiffer than the plate, and it was not: when a shim lifted a low region the plate pushed back and the rack's joints yielded first. Even where a shim took effect the correction was local, because lifting the plate at a mounting point changes its curvature and new deviation appears in the spans. Bolting a plate down at nine points is nine vertical constraints on a body whose out-of-plane position needs three, and the six redundant constraints inject internal stress whose distribution depends on assembly order and torque.

The replacement is ATP-5 cast aluminium tooling plate [10], thickness chosen against gravity rather than heating. Deflection on three supports goes as qa^4/D with $D \propto h^3$, and self weight makes $q \propto h$, so sag falls as $1/h^2$ [21], [22]. Fig. 6 puts numbers on that by finite element analysis, the same method used to set the thickness of metrology surface plates [23]: at 1/8 inch the plate would hold 18.9 µm of self-weight error, a tenth of the flatness budget spent before heating, mounting, or probe error is counted, while the 5/16 inch (7.94 mm) plate chosen holds 3.0 µm. The plate is now the machine's flatness reference, so its stiffness is not a place to economise, and the mass penalty is small in use because in CoreXY the bed moves only in Z, though it eats into the Z reduction's unpowered margin. Building it took three holes for the kinematic balls, a bonded 350 W Keenovo silicone mat below and a magnetic sheet above. There is no rack, because there is nothing to adapt.

With the new bed installed the limiting factor moved to the probe. A touch probe measures one point per physical contact, each costing a deceleration, a deploy, a touch and

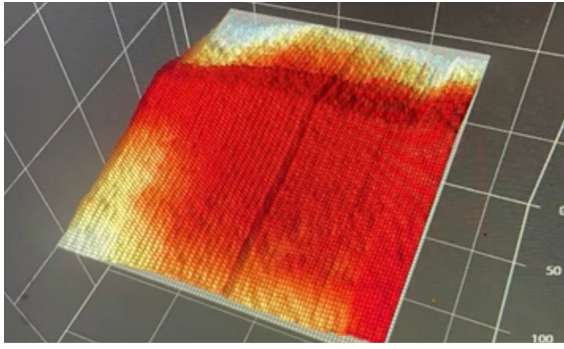


Fig. 7. A 100×100 mesh of the ATP-5 bed at 70°C , taken with the eddy probe in a few minutes. Total envelope 0–0.2 mm, eight times better than Fig. 5 (also at 70°C), at ten thousand sample points against seventy.

a retract, and that cost does not fall with better firmware. An eddy-current sensor instead drives an inductive coil at radio frequency above a conductive surface: the field induces circulating currents in the metal, those produce an opposing field, and the coil's effective inductance varies with distance. Three consequences follow. It needs a conductive target, and on Boaz reads the PEI spring-steel sheet, not the plate (over the MK52's Cryogrip sheet it never worked, cause unknown). It never touches, so mesh density becomes nearly free. And it is temperature sensitive, so calibration must include a thermal model; skipping that gives a probe accurate at one bed temperature and wrong at every other, wrong by an amount that looks like a real change in bed height [24].

A dense mesh is only meaningful if the machine knows where the probe was when each sample was taken. If the gantry is skewed or racked, a nominally horizontal probing pass is physically diagonal and the mesh is a distorted map of a real surface. The probe therefore requires the motion work in Section VI-C; on the first build's XY system it would have produced far more data of no better quality.

The probe produces 100×100 meshes in a few minutes (Fig. 7) with an envelope of 0–0.2 mm, against 1.6 mm on the first build. Most of that improvement belongs to the plate; the probe is what makes it usable, because the first layer became hands-off. Dense data also reversed an earlier belief: bicubic tension at maximum now produced visible over-compensation, chasing small local features and overshooting between them, and had to be lowered. When samples are dense the interpolating surface should barely swing, because the data already contains the shape. Sparse data invites aggressive interpolation and punishes it, and when a compensation parameter has to be pushed to its limit to be useful, the problem is almost always upstream of the compensation.

B. Cooling and gantry mass

The original duct failed at both ends of the material range: PETG was over-cooled and delaminated, while large PLA layers were under-cooled, because a narrow jet cools a small patch repeatedly while the rest of the layer cools by natural convection. The replacement is an annular plenum

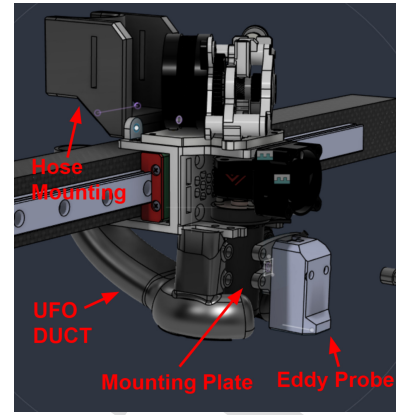


Fig. 8. The rebuilt cooling assembly: the “UFO” duct with its annular plenum and distributed outlets, the hose-anchoring fixture, and the laser-cut aluminium doubler plate.

with distributed outlets (Fig. 8), converting blower output into coverage rather than velocity and removing the dependence on direction of travel. Inlet geometry matters as much as disk diameter, because all the flow enters at one point and an abrupt entry starves the outlets furthest around the ring, which looks exactly like a levelling problem. The aerodynamic rule did not change between duct generations. What changed is the objective it was applied to: the physics is invariant, and the specification was wrong. A printed fixture now anchors the CPAP hose out of the plane of motion, and the extruder doubler plate, iterated from PLA to ABS, was finally cut in aluminium. When a part is being iterated in successively better plastics, the answer is usually not the next plastic.

The steel X beam was the largest single moving mass and a drop-in change. Steel and aluminium are nearly identical in specific stiffness, which is why swapping one for the other buys less than intuition suggests; carbon fibre is in a different class, with very high axial modulus at a density below aluminium's and high internal damping. The catch is the 1 mm wall, thin enough that a local clamping load crushes it, so an internal standoff is mandatory at every rail fastener, each exactly the internal height of the tube. Short leaves the wall deflecting before load transfers; long pushes the walls apart and puts the rail on a convex surface. The brass counterweight stayed, because a balanced toolhead at slightly higher mass excites less rotation than a lighter one that rocks; no sweep was run without it, so that argument is the whole case.

C. Motion system

The first motion system could have been repaired. It was redesigned instead, because the defects were distributed, with nearly every pressed joint wrong in one of two directions, and several were designed in rather than built in, most obviously the tensioner's clearance slot. Rotating each motor 45° in plan and moving both belt loops to the rear recovered roughly 50 mm of rail length per axis at no cost in frame size, and the XY motors were replaced with longer-body units driven by 60 V-rated TMC5160 modules on a dedicated 48 V supply, a

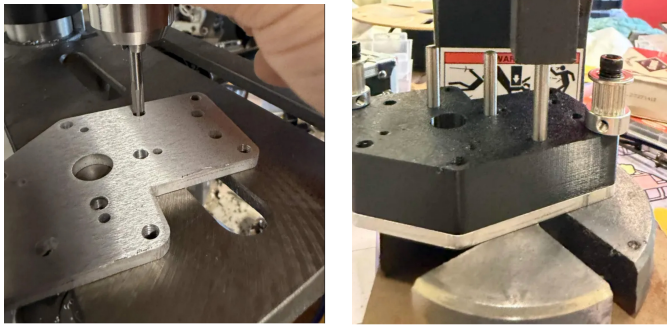


Fig. 9. Left: reaming to 4.95 mm. The reamer is held in the drill press for perpendicularity and turned by hand, so the spindle's runout cannot enlarge the hole. Right: pins pressed with an arbor press through plastic jigs that hold them perpendicular to the plate.



Fig. 10. Runout verification. The assembly is squared between granite blocks and a dial indicator reads the idler's running surface at several angular positions and at several heights. Disagreement between heights indicates the pin is not perpendicular.

240 W Meanwell NDR-240-48 [25], [16], because a stepper's winding is an inductance and the rate at which current can be forced into it against its back-EMF is set by available voltage.

The tensioner is the clearest fix in the rebuild. The first design let the dowel pin slide in an oversized slot, reintroducing clearance at the one idler whose position sets belt tension. The new design presses the pin rigidly into an SLM aluminium block and constrains the *block* to slide in a pair of trammable channels, so the compliant degree of freedom moved from the locating joint to a guide surface. The X-beam idler mounts gained spacers between top and bottom plates, so plate parallelism is set by spacer lengths rather than by press-fit luck.

Dowel pins were then done in five steps, each of which the first build skipped or got wrong.

- 1) **Draw the hole undersize**, 0.2 mm under the 5 mm pin, so the finished size is set by reaming rather than by the cutting process.
- 2) **Measure the pins** on arrival with a micrometer. This two-minute operation caused half the first build's fit problem when it was skipped: a fit is a relationship between two parts, and only one of them can be machined.
- 3) **Ream by hand** to 4.95 mm against a measured 5.00 mm pin. The reamer sat in the drill press for perpendicularity but was turned by hand (Fig. 9, left), because a powered spindle with 0.05 mm of runout cuts a hole larger than the



Fig. 11. Rail parallelism by dial indicator. The gauge rides the carriage and reads against a granite block clamped to the frame. The reference is the granite, never the extrusion.

reamer.

- 4) **Press through a jig** that holds the pin perpendicular to the plate (Fig. 9, right). Without one, a pin entering under a few kilonewtons finds its own angle, and an angular error is the one that cannot be corrected later.
- 5) **Measure every assembly**, squared between granite blocks and read with a dial indicator at several angular positions (Fig. 10). Acceptance was total indicated runout below 0.001 in; assemblies that failed were remade.

Readings were taken at more than one height, which is what distinguishes a real measurement from a ritual. Equal runout at top and bottom means the pin is offset but parallel, which is tolerable; different runout at different heights means the pin is tilted, which is not, because it steers the belt. One finding here is not obvious from any drawing: pressing a bearing hard against a plastic spacer loads it axially, and an axially preloaded bearing is stiff to turn, which shows up as mysteriously high motor current. A thin shim between spacer and bearing race takes the press load off the bearing.

The rails used jigs only for initial centring, then measurement (Fig. 11): a dial indicator on each carriage reading against a granite block clamped to the machine, traversed along full travel, so any deviation as the carriage moves is non-parallelism in microns with a sign. The granite is the reference because the extrusion is not straight to the tolerance the rails need; aligning a rail to the frame it is bolted to measures nothing, because the error being hunted is in the reference. With the rails trammed and a thinner lubricant in place of the Mobilux EP 1 that had made the axes stiff, the gantry traversed full Y travel with no visible frame deflection and no binding, for the first time in the project's life.

The machine was also closed in at this point. FFF printing emits ultrafine particles and volatile organic compounds, which is the reason that decides whether a machine can run overnight in a room where someone sleeps, and a warm chamber additionally reduces the thermal gradient that makes polymer contraction differential. The upper enclosure is hinged

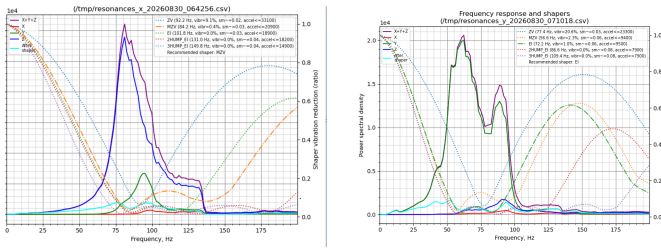


Fig. 12. Resonance sweeps of the rebuilt machine, 30 August 2026. Left, X: one dominant peak, 20 900 mm s⁻² with MZV (33 100 mm s⁻² with ZV). Right, Y, half an hour later: two peaks near 62 Hz and 90 Hz; EI at 72.2 Hz, 9500 mm s⁻².

so it folds back, because a fixed top makes belts, idlers and gantry reachable only by removing panels. The lower tier's open ABS shelf became a sealed transparent box carrying supply, controller and terminals on DIN rail, so debris and conductive carbon dust stay out and a fault in the electronics is a fault inside a box rather than under a machine [26].

D. Calibration

The motion rebuild left a residual: a small standing rotation of the gantry from a remaining imbalance in belt tension, below what a machinist square resolves on a frame this size and above what a 0.2 mm functional tolerance can absorb. The instrument is a printed artifact. The Califlower tool [27] is measured on X, Y and both diagonals, and the diagonals are the point: a pure scaling error changes X and Y and leaves the diagonals in proportion, while a skew changes the diagonals differentially. From those four numbers the tool computes the skew angle and emits the firmware correction. Correcting skew is not fixing skew: the transform hides a mechanical state, so a growing correction should be read as a symptom rather than as a number to update.

Slicer work produced one result that generalises. Each extruded line contracts along its own length as it cools, so the first layer's stress field has the geometry of its infill pattern. On a circular part a rectilinear first layer lays every line as a chord and pulls the perimeter inward at the edges, which is the classic circular-part lift; changing the pattern so paths follow the outline redistributes that stress. The remaining adjustments were ordinary, and most presets in a modern slicer [28] were adequate, itself a result, because a machine needing heroic slicer settings is compensating for something mechanical.

Input shaping convolves commanded motion with a short impulse sequence designed so that the vibration excited by each impulse cancels the vibration left by the previous one [29], and Klipper identifies the modes with an accelerometer sweep [30], [31]. Fig. 12 is the strongest evidence here in both directions, and it reads as a construction audit rather than a tuning input. The X response (left) is one dominant peak with a clean roll-off: one well-defined compliant mode, which is what a well-constrained structure looks like. The peak near 81 Hz is on the Z channel, which is the machine's X: the sensor board is mounted on its side, rotated 90°. Y carries a smaller peak near 95 Hz. The failure signature is the opposite,

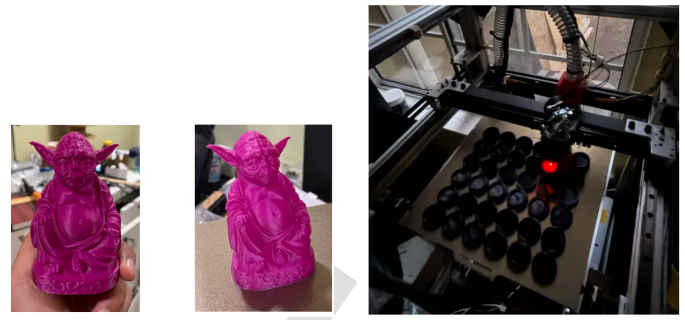


Fig. 13. Left: a tall, highly detailed statue printed on the rebuilt machine. Fine features stress XY accuracy, and height stresses Z consistency over hundreds of layers. Right: full plates of PETG medicine boxes from the production contract. The threaded lids require better than 0.2 mm on the mating features, which the first build could not hold.

multiple peaks or a broad noisy response, meaning several modes at similar energies from loose joints, binding rails, or a flexing frame, and the Y response (right) shows it: two peaks near 62 Hz and 90 Hz, the second two-thirds the height of the first, from compliances not yet identified. A shaper cancels one identified mode well and cannot cancel a forest of them, and the attempt costs print quality directly, because heavier shaping rounds corners and smooths detail. Mechanical work made shaping cheap on X; Y is unfinished.

X's 20 900 mm s⁻² under MZV clears the criterion for that axis; Y's 9500 mm s⁻² under EI does not, and since Klipper takes the lower axis the machine's usable acceleration is under half the criterion (Table I). One further caveat: the accelerometer sat at the top of the toolhead, not the nozzle, so it misses the compliance below it and under-reports any nozzle-side mode.

VII. EVALUATION

The first system test was a large statue (Fig. 13, left), chosen because it stresses subsystems in combination: fine surface detail requires XY positional accuracy and consistent extrusion, and height requires that layer stacking stay consistent over hundreds of layers. It printed cleanly, which was the first evidence that the rebuild had addressed the runout diagnosed in Section V rather than moved it.

A. Production contract

The real evaluation was a contract for roughly 1000 pet-medicine boxes in PETG for a medical organisation in Iraq, specified to survive sustained heat and sun (Fig. 13, right). Two machines ran the job over ten days: Boaz and an Elegoo Centauri Carbon, a factory-built CoreXY machine costing a small fraction of Boaz's parts bill.

Boaz lost by roughly 20% (Table IV), and the loss is reported rather than explained away. The claim in Section I is about method relative to other machines a reader might build, not about beating a mass-produced product on throughput per dollar. Anyone whose objective is boxes per day should buy the Elegoo.

TABLE IV
PRODUCTION COMPARISON OVER THE TEN-DAY CONTRACT.

	Boaz	Elegoo Centauri Carbon
Boxes per day	≈ 50	≈ 60
Enclosure	Yes (passive)	Yes (passive)
First-layer attention	None	None
Consumer cost	Not sold	Approx. \$300

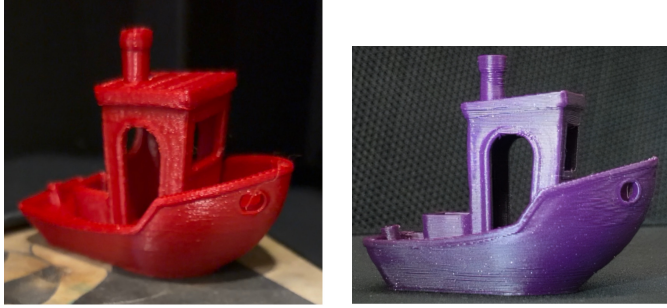


Fig. 14. #SpeedBoatRace 3DBenchy: Boaz (left) and a Hev-Ort (right), both at 16 min. At equal print time the smokestack is cleaner and the bow and stern carry fewer artifacts.

What the contract established is that the rebuilt machine holds functional tolerance. The boxes have threaded lids, and threads that engage and turn freely demand better than 0.2 mm on the mating features. The same part had been attempted on the old motion system and failed. Credit belongs jointly to the dowel work and the skew correction.

Fifty boxes a day for ten days is on the order of a hundred plates, started one after another as the previous batch was removed, with no Z-offset jogging on any of them. Against the first build, where an operator stood over every first layer, this is the largest single change in the machine's character. It comes from the flat plate and the dense meshing together, and neither alone is sufficient. The criterion asked for ten consecutive plates; the contract supplied ten times that. The job ran PETG in a closed chamber, and carbon-fibre PETG also ran without the layer-adhesion problems the open-frame machine had shown.

B. Dynamic behaviour and speed

The resonance sweeps (Fig. 12) split: X met the criterion, $20\,900\text{ mm s}^{-2}$ under MZV against $20\,000\text{ mm s}^{-2}$; Y did not, two peaks and 9500 mm s^{-2} under EI, and Klipper takes the lower axis. One gap in the evidence belongs here: the machine was never run without shaping once shaping was available, so no matched before/after pair of prints exists.

The #SpeedBoatRace convention prints a 3DBenchy [32] as fast as possible with the model unmodified and the race's fixed slicer settings, as here, and it exists because speed and quality trade against each other: any machine can be fast if the output is allowed to be poor. Boaz printed one in 16 min on its first attempt against a 20 min target. Fig. 14 compares it with a Hev-Ort, a well-regarded enthusiast DIY machine [33], at the same 16 min: the smokestack is cleaner and the bow and stern

carry fewer artifacts, both cooling-limited features, so credit goes to the duct and blower. Two caveats. This was a first attempt with no per-print tuning, so the figure is a floor rather than a limit. And a single paired comparison is an anecdote, so the honest claim is that Boaz is competitive with enthusiast-built machines at equal print time, not that it is superior by a measured margin.

Eight criteria were fixed in advance and the rebuilt machine met seven, with the production criterion met at its floor and lost outright as a comparison, and the dynamic criterion missed on the Y axis. The first build failed at least four.

VIII. DISCUSSION AND LIMITATIONS

The evidence is one machine built twice. Both builds used the same design principles, the same kinematics, and largely the same components, and they differ in whether the precision practices were executed correctly, which makes the comparison internally controlled in a way most build logs are not.

It is still $n = 1$. Nothing here establishes an effect size, and several changes between the builds were made together rather than one at a time, so individual contributions cannot be separated: the bed and the probe were replaced in the same period, and the motion rebuild and the gantry mass reduction happened close enough together that the resonance sweep measures their combined effect. Three further gaps are worth naming. There is no matched before/after resonance pair on this machine. The Benchy comparison is a single paired print against one other machine. And the production comparison ran two machines with different operators' attention over ten days, which is a field trial rather than a controlled experiment.

One pattern recurred often enough to state as a result. Every heuristic patch in this project resolved a symptom and destroyed the information needed to diagnose anything measured afterwards. The plastic shim in a dowel joint made the machine assemblable and left an eccentric, creeping joint whose contribution could not be separated from the others. Bicubic tension at maximum made a sparse mesh look like compensation and hid the fact that the plate was the problem. The effect compounds, because every subsequent measurement is taken on a machine with a known defect in it. The related pattern is that provisional parts are not provisional: the PLA Z actuators stayed through the entire first production period, and because the provisional version was compliant where the final version is stiff, the data collected in between cannot separate the two.

Two non-technical constraints shaped this project more than any design decision. The build was funded at about \$100 a month, which leaves time to think between purchases and stretches the build long enough that testing gets compressed at the end; several choices that read as engineering judgements were budget decisions, including the MK52 bed and the cheap dowel pins. There was also sustained pressure to finish before the machine was ready, and the defects that introduced did not announce themselves until a hundred hours of printing later.

The techniques in this paper are not new. They are standard practice for CNC mills and lathes and have been for decades, and what was not obvious is that they apply at this scale, to a machine that pushes plastic. The machine does not need to be a machine tool; it needs to be built like one. Getting at that knowledge was harder than applying it: beyond written sources, individual people mattered, and answers were sometimes deliberately vague or treated as not worth the time. Access is gated by social capital rather than by ability to use the information, and writing the numbers down is a partial remedy.

IX. CONCLUSION

A. Eleven rules, with the evidence that earned each

A rule without evidence attached is an opinion. Table V states each rule with the observation in this project that produced it. One further rule belongs with them and is not on the list: fix causes rather than symptoms, for the reason given in Section VIII.

B. An open question

The motion-architecture argument in Section II, read carefully, establishes that the motors must not ride on the moving structure. CoreXY is one way to achieve that and not the only one. Cross-gantry architectures arrange the X and Y beams as a cross with the toolhead at the intersection, keeping the motors on the frame while offering two things CoreXY does not: shorter belt spans, and therefore higher drive stiffness, since a belt's spring rate $k = E_{\text{eff}}A/L$ falls with free length; and belt loops that do not cross, so a fault on one axis stays there. Motor count does not separate them, since four-motor CoreXY exists [7].

This project provides no evidence on which is better, because it built one machine. Settling it needs either a comparison of drive stiffness and moving mass at matched build volume, or two machines built to the same standard with different kinematics. That is the most interesting work this project leaves undone, and it is orthogonal to the claim tested here: both architectures need a stiff frame, pressed dowels, measured rail parallelism, and metrology during assembly. Choosing the kinematics wrong costs some acceleration. Building either one badly costs everything.

C. Claim, settled

The first build produced grainy walls, parts that would not assemble, and first layers requiring an operator. The second met seven of the eight criteria fixed in advance: runout under 0.001 in, a bed mesh envelope of 0.2 mm, ten days of production plates without intervention, a single-peak X response permitting $20\,900\text{ mm s}^{-2}$, and a 16 min Benchy on the first attempt. The eighth, acceleration, fell to the Y sweep's 9500 mm s^{-2} .

That is one experiment with one machine and should be read as such. What it establishes is not that Boaz is a good printer; it is a competent one that lost a production race to a \$300 machine. What it establishes is that the difference between its

two builds was the execution of precision practice, and that the difference was large, measurable, and visible in every part the machine made.

Those practices are old, they are documented, and they are cheaper than the components most builders agonise over. A dial indicator and a granite block cost less than a set of rails. The hobby's persistent quality gap is not a materials problem and not a kinematics problem. It is a measurement problem, and it was solved a century ago in the next workshop over.

ACKNOWLEDGMENTS

Members of an online machine-building community, several of them former CNC technicians and toolmakers, reviewed design decisions during the project and supplied the measurement practice this paper reports. One example stands for many: the requirement that a belt tensioning screw act through the centre of its block, so that belt load cannot tilt the block and misalign the idler it carries. That advice appears in no build guide the author could find.

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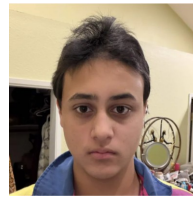
TABLE V
THE ELEVEN CONSTRUCTION RULES AND THE OBSERVATION THAT EARNED EACH.

Rule	Evidence
1 Stiff frame in large-section extrusion, gusset-joined, with more joints than strength requires.	Blind joints failed three ways: angle inherited from the saw cut, hand taps seizing in deep blind aluminium, and three bolt axes intersecting at a box corner. Gusset clearance holes are the adjustment range that makes tramming possible.
2 Enclose the machine.	The frame ran a year without the panels it was designed for, and PETG warped in open air. Enclosed, PETG ran Boaz's half of a thousand-part contract without adhesion failures; so did carbon-fibre PETG.
3 Enclose the electronics separately.	An open shelf collected debris and discouraged overnight running, which cost more throughput than any mechanical limit did. Machining carbon fibre later released conductive dust into the same space.
4 Design the toolhead for flow, cooling, and centre of mass.	A concentrated duct over-cooled PETG and under-cooled large PLA layers; the distributed replacement fixed both without losing fine detail. Balance was bought with a brass plate, mass added to a moving assembly, on the centre-of-mass argument; no sweep was run without it.
5 Build load-bearing and locating assemblies from sheet metal and metal spacers.	Printed Z actuators bent under standing belt tension and put runout into the axis whose only job is position. A printed extruder doubler warped through two generations of polymer before being cut in aluminium.
6 Locate idlers on dowel pins in reamed holes, ≈ 0.05 mm interference on a 5 mm pin.	The first build reamed to nominal on a drill press with runout, bought pins 0.04 mm undersize, shimmed with plastic film, and pressed others at 0.2 mm: clearance fits and angular fits and nothing between. The rebuild held below 0.001 in TIR.
7 Square the frame on granite with a machinist square, during assembly.	A frame a few tenths of a degree out of square produces systematic sub-percent error in every part, and firmware cannot correct skew that varies with position.
8 Align rails and verify pressed joints against external granite, never against the frame.	Y rails set with printed jigs good to 0.2 mm left the frame visibly bending as the gantry traversed. Trammed by indicator against clamped granite, the same rails moved through full travel with no visible deflection.
9 Remove mass from what moves; add it to what does not.	A steel X beam and a thick duct forced acceleration down to keep quality acceptable; carbon fibre and a lighter duct raised it on X; Y is still capped. Mass that buys balance is a different trade from mass that rides along.
10 Adopt sensing that removes the human.	A touch probe forced a trade between mesh density and cycle time. An eddy probe reading the steel print sheet produced 100×100 meshes in minutes and made the first layer hands-off for a ten-day production run.
11 Calibrate with printed artifacts, and re-calibrate after every mechanical change.	Residual racking too small for a machinist square still prevented threaded parts from assembling. The Califlower artifact measured it and produced the firmware correction that fixed it.

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