

Ovo

A folding hydrokinetic charger, designed end to end as parametric code-CAD

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Rendered from the CAD assembly: blades deployed on their stops, roll fins fitted. Rotor diameter 194 mm, egg 75 mm in diameter.

Abstract. Ovo is a portable hydrokinetic charger for rivers and towing: a 75 by 161 mm egg that unfolds into a 194 mm three-blade rotor. The rotor sits downstream of the hull and its blades fold forward into shallow pockets in the tail, so that by design the flow opens them and the rotor thrust holds them on their stops (a water test must show that the flow overcomes the stowed detent); stowed, the product is a closed egg. The blades come from a constraint-aware blade element momentum (BEM) design with Prandtl losses, the Buhl high-induction correction and NeuralFoil polars: every printability and folding constraint is priced in power coefficient, and together they cost 2.4 % against the unconstrained optimum. The model predicts a peak $C_P = 0.367^{\text{est}}$ and $18.3 \text{ W}^{\text{est}}$ of shaft power at 1.5 m/s. A generator trade study selects the CubeMars GL40 for a v2 build and shows that winding losses move the electrical optimum far above the rotor's C_P peak, to a load line that must also respect a cavitation limit. The whole machine (72 parts) is built by one Python pipeline in build123d from a single parameter file and is verified automatically: interference in 7 assembly states, a rotor spin sweep, fold and closing sweeps and 25 key clearances on fine meshes of the CAD solids, and watertight print meshes. The v1 is a display build with real bearings, folding blades and magnetic detents but no electronics. It has not been printed or tested in water yet; every performance figure in this report is a model estimate and is marked ^{est}.

75 × 161 mm

stowed egg, blades flush in pockets

194 mm

rotor diameter, 3 folding blades

18.3 W^{est}

shaft power, 1.5 m/s, C_P 0.367

7.66 W^{est}

DC through a diode bridge, GL40 generator, 1.5 m/s

72

parts in the assembly

25 / 25

key clearances pass, measured on the CAD

235 g

printed filament, display build

25

STL print files, all watertight, shells support-free

Contents

1 Introduction	2
1.1 Problem	2
1.2 Requirements and scope	3
2 Concept and key decisions	3
3 Hull	4
4 Blade design	5
4.1 Method	5
4.2 Design tip speed ratio	6
4.3 What the constraints cost	6
4.4 Performance	7
4.5 Generator coupling and the load line	7
4.6 Cavitation cap	9
4.7 Loads, structure and the hinge geometry	10
4.8 Verification and sensitivity	10
5 Fold kinematics, pockets and detents	10
5.1 Hinge and the stowed blade	10
5.2 Pockets	11
5.3 Detents and hub loads	12
6 Layout and drivetrain	12
6.1 Axial stack	12
6.2 Bearings, shaft and thrust path	12
6.3 Seals, joint and bayonet	14
6.4 Generator trade study and the GL40	14
7 Roll stability and fins	16
8 CAD architecture and verification	16
8.1 Code-CAD from a single source of truth	16
8.2 Parts and states	17
8.3 Verification	18
9 Printability and print plan	19
10 Bill of materials and cost	20
11 Prior art	21
12 Limitations and v2 roadmap	21
12.1 Limitations	21
12.2 Roadmap	22
13 Conclusion	22
References	22

How to read the numbers. Ovo exists as CAD, design studies and print files. Nothing has been printed, bench-tested or put in water. Values marked ^{est} are outputs of models (BEM with NeuralFoil polars, a generator loss model, a layout mass and buoyancy model) and carry their uncertainty, discussed where they appear. Dimensions and clearances are measured on the CAD solids by the build's own checks. The design and analysis numbers in this report are pulled by `scripts/report.py` from the study and check JSON files and the parameter file of the repository, so the report cannot drift from the model.

1 Introduction

1.1 Problem

Moving water is a dense energy source. The kinetic power through a rotor disc of area A is

$$P_{\text{avail}} = \frac{1}{2} \rho A U^3, \quad (1)$$

so a 194 mm rotor in a 1.5 m/s river current (water at $\rho = 1000 \text{ kg/m}^3$) sees 49.9 W. A good small rotor extracts a third of that, which is enough to charge a phone or a power bank from a canoe camp, a moored boat or a yacht under tow. The difficulty is size: a rotor large enough to matter does not fit in a bag, and the commercial products

Table 1: Requirements and their realisation. Out of scope for v1: electronics, a water test and CFD.

ID	Requirement	How the v1 design meets it
R1	Stows as a closed shape that fits a bag; nothing protrudes	75 × 161 mm egg; blades in tail pockets, at most 0.065 mm above the hull; fins slide off, flush strips close their rails
R2	Rotor as large as practical when deployed	194 mm swept diameter, 2.6 × the egg diameter
R3	Real mechanics in the display build	rotor on a 608 and a 688 ball bearing; blades fold on steel pins; magnetic detents hold both positions
R4	Ready for a v2 with a real generator and sealing, without redesign	GL40 generator bay and mount, lip seal pocket, O-ring equator joint, M12 cable gland seat on a v2 front shell
R5	Printable on a Bambu Lab printer (256 mm cube), one material per plate	all 25 print files fit the bed; shells print without supports; blades on a printed cradle
R6	One parametric source of truth	every dimension in <code>ovo/params.py</code> ; studies write JSON, the CAD reads both
R7	Every geometric claim checked automatically	interference, clearances, spin, fold and closing sweeps, mesh audits (Section 8)

reviewed in Section 11 either use small rotors or rigid housings. Ovo's goal is the largest practical rotor when running and a closed, compact shape when carried.

1.2 Requirements and scope

Table 1 lists the requirements set with the project owner at the start (`docs/design-brief.md`) and where each one is met. The design flow speed is 1.5 m/s, a decent river current; towing covers 2 to 2.5 m/s. The working range studied is 1.0 to 2.5 m/s.

The scope is deliberately limited. The v1 is a **display build**: it shows real mechanics, but the generator is a printed replica with the real envelope and hole patterns, and there is no electronics. All performance numbers are therefore predictions. Section 12 lists what they rest on and the measurements that would confirm them.

2 Concept and key decisions

The concept follows from R1 and R2: a large rotor that folds into a small closed body. Table 2 summarises the decisions, the alternatives that were considered and what each decision costs.

No duct. A duct can raise the flow through a given rotor, but it is a rigid ring at least as large as the rotor. It cannot fold, and it would fix the stowed diameter near 194 mm instead of 75 mm. An open axial rotor folds naturally about its hub.

Downstream rotor, blades folding forward (Figure 1). Each blade hinges about a tangential pin on the hub. Folded, it points upstream along the tail, so the flow is meant to drag its tip aft and open it; running, the thrust holds it on the stop. Deployment is passive by design, which a water test still has to confirm: the magnet detent that holds a stowed blade may resist the flow (Section 12). The load path is benign: the dominant blade load, the

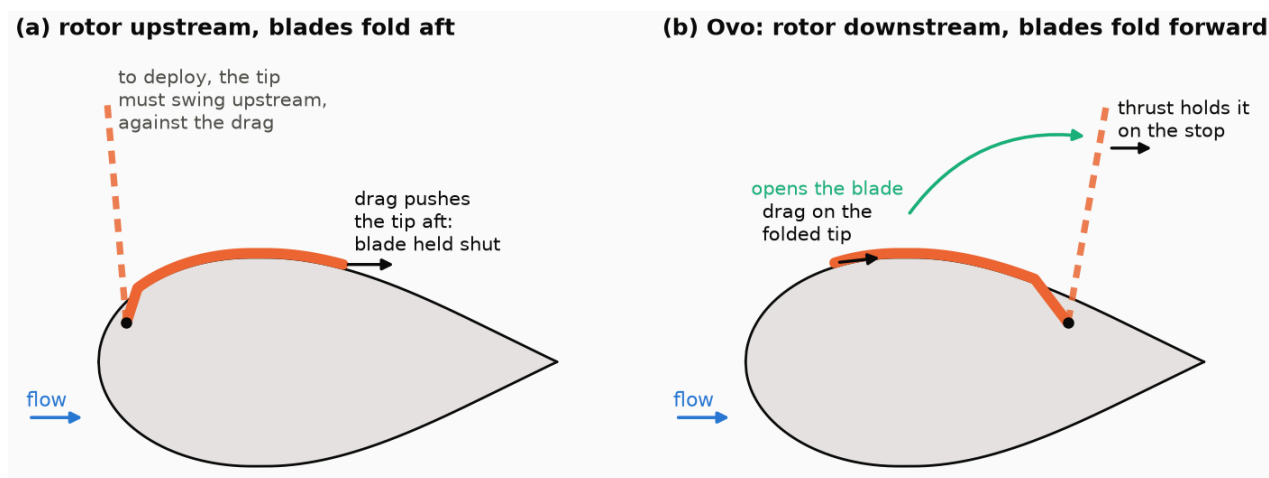


Figure 1: Why the rotor sits downstream and its blades fold forward (schematic). (a) With the rotor at the nose, folded blades lie aft along the hull and the drag on their tips holds them shut; deploying would need the tip to move upstream. (b) With the rotor at the tail, folded blades lie forward over the tail: drag on the tip is meant to swing them open, and in operation the rotor thrust presses each blade onto its deployed stop.

Table 2: Key design decisions.

Decision	Alternatives	Why	Price paid
open axial rotor, 3 blades	ducted rotor; cross-flow rotor	folds about its hub; no rigid ring	no flow augmentation; tip losses
rotor downstream, blades fold forward	rotor at the nose, blades fold aft	flow should open folded blades (design intent); thrust holds them on the stop	overhung rotor; fold only at the stowing index
pre-cone from the hull curve	straight blade, deeper pockets	stowed blade flush with the hull	curved pitch axis with an in-plane lead
Myring egg split at the equator	cylinder with end caps	closed smooth form; one profile for shells, hub and spinner	short body (L/D 2.20), drag not optimised
magnetic detents	springs, ball detents	no wear parts; hold deployed and stowed	holding force only modelled
slide-in recessed fins	fixed fins	stowed egg stays 75 mm	rail play: couple ratio spread (Section 7)
constraint-aware BEM	textbook optimum blade	printable, foldable, cost of each constraint known	2.4 % of C_P at the design point ^{est}

flapwise moment about the hinge, pushes the blade onto its stop. The price is mechanical: the rotor sits behind the bearings (an overhung load, Section 6.2), and the blades can only fold home at one rotor angle in every 120° (the stowing index), where each blade lies over a pocket.

Pre-cone taken from the hull. A 90° fold about a tangential axis maps “aft” in the deployed blade to “outward” in the stowed one. A blade whose pitch axis leans aft by the local hull slope therefore hugs the convex tail when folded. Deployed, the same lean is a downstream cone of 12.6° on average from hinge to tip (25.0° locally at the root, -16.0° at the tip, where the curved blade turns back upstream). Coning costs nothing measurable in power (Section 4), and downstream coning is expected to help passive yaw alignment; that benefit has not been analysed here.

An egg. A Myring body of revolution (Section 3) gives a smooth, closed form whose tail curve doubles as the bed for the folded blades. The egg splits at its equator into two printed shells joined by a bayonet with an O-ring groove. Roll fins slide into rails recessed in the front shell and come off for packing, so the stowed egg keeps its 75 mm diameter.

3 Hull

The hull is a Myring body of revolution [1]: an elliptic-type nose, a short parallel band and a cubic tail that closes to a point with a prescribed half-angle. With the nose length a , band b , tail length c , diameter D , nose exponent n and tail half-angle θ :

$$r(x) = \frac{D}{2} \left[1 - \left(\frac{x-a}{a} \right)^2 \right]^{1/n}, \quad 0 \leq x < a, \quad (2)$$

$$r(x) = \frac{D}{2} - \left(\frac{3D}{2c^2} - \frac{\tan \theta}{c} \right) \xi^2 + \left(\frac{D}{c^3} - \frac{\tan \theta}{c^2} \right) \xi^3, \quad \xi = x - a - b, \quad (3)$$

with $r = D/2$ on the band. Ovo uses $a = 55$, $b = 5$, $c = 105$ mm, $D = 75$ mm, $n = 2$ (an elliptic nose: the wide end of an egg) and $\theta = 25^\circ$, which keeps the tail full enough to house the rear bearing and the shaft seal: the radius is still 26.7 mm at the static tail end ($x = 108$ mm). The profile length is 165 mm (L/D 2.20). The built egg ends at $x = 160.99$ mm, where the spinner cap rounds the Myring point to R3 (L/D 2.15).

The body encloses 401 cm³ (0.40 L), wets 287 cm² and has a frontal area of 44.2 cm². Its length Reynolds number at 1.5 m/s is 2.5×10^5 , in the transitional range. The shell wall is 2.4 mm (six perimeters of a 0.4 mm nozzle). The split plane is the equator, $x = 57.5$ mm, in the middle of the parallel band, so the O-ring seals on a cylindrical surface. The hull’s drag was not computed: the tow-load estimate uses $C_D = 0.10$ on the frontal area (1.38 N^{est} at 2.5 m/s), a stated assumption. Myring’s own study found the drag of such bodies fairly insensitive to the fullness of nose and tail within a moderate range [1], which is what allows the shape to be chosen for packaging here; whether Ovo’s short, full egg lies inside that range at its much lower Reynolds number was not checked. The nose carries a printed tow eye sized for 231 N, three times the working load of 76.9 N^{est} at 2.5 m/s (rotor thrust at $C_T = 0.8$ plus hull and fin drag).

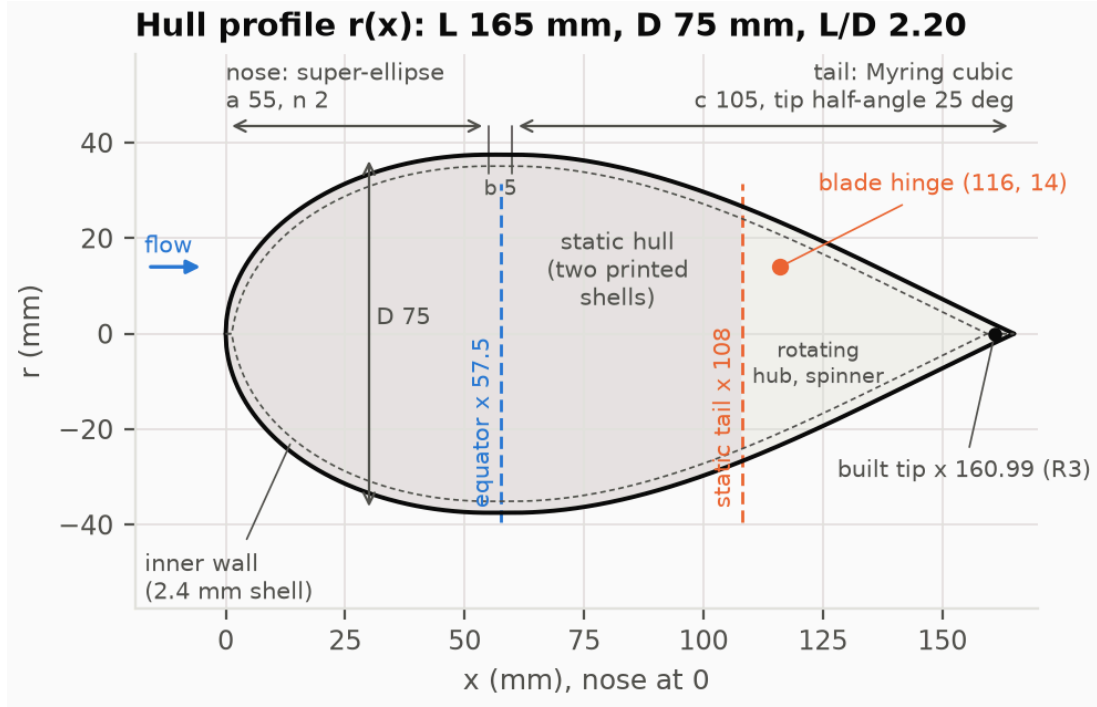


Figure 2: Hull profile from ovo/profile.py. Forward of $x = 108$ mm the hull is the two printed shells; aft of it the same curve is continued by the rotating hub and spinner, so the stowed product reads as one egg.

4 Blade design

4.1 Method

The rotor is designed with blade element momentum theory [2], [3], extended for a coned rotor. The pitch axis leans aft by the cone angle β ; axial momentum goes through the projected annulus $2\pi r dr$ and the spanwise component of the inflow is neglected. With axial and tangential induction a , a' , tip speed ratio $\lambda = \Omega R/U$, solidity $\sigma = Bc/(2\pi r)$ and twist θ_t :

$$\tan \varphi = \frac{U(1-a) \cos \beta}{\Omega r(1+a')}, \quad \alpha = \varphi - \theta_t, \quad (4)$$

$$\frac{a}{1-a} = \frac{\sigma C_n \cos^2 \beta}{4F \sin^2 \varphi}, \quad \frac{a'}{1+a'} = \frac{\sigma C_t}{4F \sin \varphi \cos \varphi}, \quad (5)$$

$$C_n = C_l \cos \varphi + C_d \sin \varphi, \quad C_t = C_l \sin \varphi - C_d \cos \varphi, \quad (6)$$

$$F = \frac{2}{\pi} \arccos(e^{-f_{\text{tip}}}) \cdot \frac{2}{\pi} \arccos(e^{-f_{\text{hub}}}), \quad f_{\text{tip}} = \frac{B}{2} \frac{R-r}{r \sin \varphi}, \quad f_{\text{hub}} = \frac{B}{2} \frac{r-R_h}{R_h \sin \varphi}. \quad (7)$$

F is the product of the Prandtl tip and hub loss factors; the hub-loss radius R_h is the aerodynamic root, $r = 30$ mm. Above $a = 0.4$ the momentum relation is replaced by Buhl's empirical branch [4] for the local thrust coefficient, used in the closed form of Ning [5]:

$$C_T = \frac{8}{9} + \left(4F - \frac{40}{9}\right)a + \left(\frac{50}{9} - 4F\right)a^2, \quad a > 0.4. \quad (8)$$

Thrust, torque and the power and thrust coefficients follow by integration over 24 annular elements:

$$C_P = \frac{P}{\frac{1}{2}\rho AU^3}, \quad C_T = \frac{T}{\frac{1}{2}\rho AU^2}, \quad P = \Omega Q. \quad (9)$$

The equations are solved by a relaxed fixed-point iteration on (a, a') ; elements that do not converge fall back to Ning's bracketed residual in φ [5]. Lift and drag come from NeuralFoil [6], a neural network trained on XFOIL [7] analyses, evaluated on the exact section the CAD builds (NACA 44xx with the blunt printed trailing edge) at each element's Reynolds number, with $n_{\text{crit}} = 5$ for disturbed river inflow and a printed surface. At 1.5 m/s the

Table 3: Design TSR study at 1.5 m/s (BEM estimates^{est}). The copper-loss index is $(Q/Q_{\text{TSR } 4})^2$ at equal generator: lower torque at higher TSR means lower winding loss. The chosen design is in bold.

Design TSR	C_P optimum	C_P Schmitz	C_P printable	printable peak (TSR)	chord root / tip (mm)	rpm at 1.5 m/s	torque (N m)	copper-loss index
3.0	0.3641	0.3585	0.3590	0.3653 (3.63)	28.0 / 21.2	443	0.386	1.71
3.5	0.3722	0.3676	0.3660	0.3670 (3.76)	28.0 / 14.2	517	0.337	1.30
4.0	0.3757	0.3718	0.3666	0.3670 (3.82)	25.1 / 14.0	591	0.296	1.00
4.5	0.3754	0.3720	0.3616	0.3663 (3.93)	20.8 / 14.0	665	0.259	0.77
5.0	0.3718	0.3688	0.3513	0.3641 (4.09)	17.2 / 14.0	738	0.227	0.59
5.5	0.3654	0.3627	0.3355	0.3593 (4.31)	14.3 / 14.0	812	0.197	0.44
6.0	0.3565	0.3540	0.3131	0.3585 (4.33)	14.0 / 14.0	886	0.168	0.32

elements run at Re 55,357 to 82,910, a transitional regime where XFOIL-type predictions are least certain; the sensitivity study below prices that uncertainty.

Two reference blades bracket the design. The **BEM optimum** lets every annulus maximise its own tangential force (annuli are independent in BEM), including drag and both loss factors. The **Schmitz** blade uses the classical closed form [8]

$$\varphi_{\text{opt}} = \frac{2}{3} \arctan\left(\frac{1}{\lambda_r}\right), \quad c = \frac{16\pi r}{BC_l} \sin^2\left(\frac{\varphi_{\text{opt}}}{2}\right), \quad (10)$$

at the lift coefficient of best lift-to-drag ratio. The **printable blade** is then a straight-tapered, linearly twisted blade whose four parameters (root and tip chord and twist) are optimised with Nelder-Mead for C_P at the design point, inside the printability and folding bounds.

4.2 Design tip speed ratio

For each candidate design TSR from 3 to 6 the study computes the BEM optimum, the Schmitz blade and the printable blade (Table 3). Unconstrained, the optimum is flat between TSR 4 and 4.5. The printability bounds change the picture: the optimum chord at 0.9 R shrinks with TSR, so the 14 mm tip-chord floor makes high-TSR blades increasingly over-solid outboard, and below TSR 4 the root runs into the 28 mm chord cap (three stowed blades must fit side by side around the tail). The printable blade designed for TSR 4 has the highest C_P at its own design point and is selected. Its C_P peak sits at TSR 3.82 at 1.5 m/s.

4.3 What the constraints cost

The printable blade differs from the BEM optimum in four ways: a blunt trailing edge of 0.6 mm (1.5 nozzle lines, 2.4 % of the chord at the root and 4.3 % at the tip), straight chord and twist laws, a chord window of 14 to 28 mm and a root-twist cap of 25° from the folding geometry, plus the 12° aft cone. Table 4 adds them one at a time, re-optimising the linear laws under all constraints so far, so that each step's cost belongs to its constraint and not to the fitting method.

The blunt trailing edge is the largest single cost (1.6 %): its base drag is the price of printing the blade. The linear laws cost 0.8 %, because a straight line cannot follow the optimum where the loss factors unload it at the root and at the tip (Figure 3). The chord window and the twist cap cost almost nothing at TSR 4, and the cone is neutral within the model. In total the printable, foldable blade gives up 2.4 % of C_P (0.3757 to 0.3666). For scale, merely drawing straight lines through the optimum at 0.7 R and 0.9 R, without optimising them, would have lost 1.6 % on top of the blunt edge.

The final blade: 3 blades, tip radius 97 mm, aerodynamic root at 30 mm, chord 25.1 to 14.0 mm and twist 17.2° to 5.2° (both linear in radius), NACA 44xx sections from 18 % to 12 % thickness with the 0.6 mm trailing edge, stacked on a pitch axis at 0.30 of the chord.

Table 4: Constraint-cost chain at the design point (1.5 m/s, TSR 4; BEM estimates^{est}).

Step (cumulative)	C_P at TSR 4	step	total	chord root / tip (mm)	twist root / tip (°)
BEM optimum (sharp TE, no cone)	0.3757			24.7 / 11.6	8.6 / 2.1
+ blunt TE 0.6 mm	0.3695	-1.6 %	-1.6 %	24.7 / 10.9	9.1 / 2.6
+ linear chord and twist laws	0.3665	-0.8 %	-2.5 %	26.7 / 11.4	18.1 / 4.6
+ chord within [14, 28] mm	0.3661	-0.1 %	-2.5 %	25.0 / 14.0	17.7 / 5.4
+ root twist at most 25°	0.3661	+0.0 %	-2.5 %	25.0 / 14.0	17.7 / 5.4
+ 12° aft cone (final blade)	0.3666	+0.1 %	-2.4 %	25.1 / 14.0	17.2 / 5.2

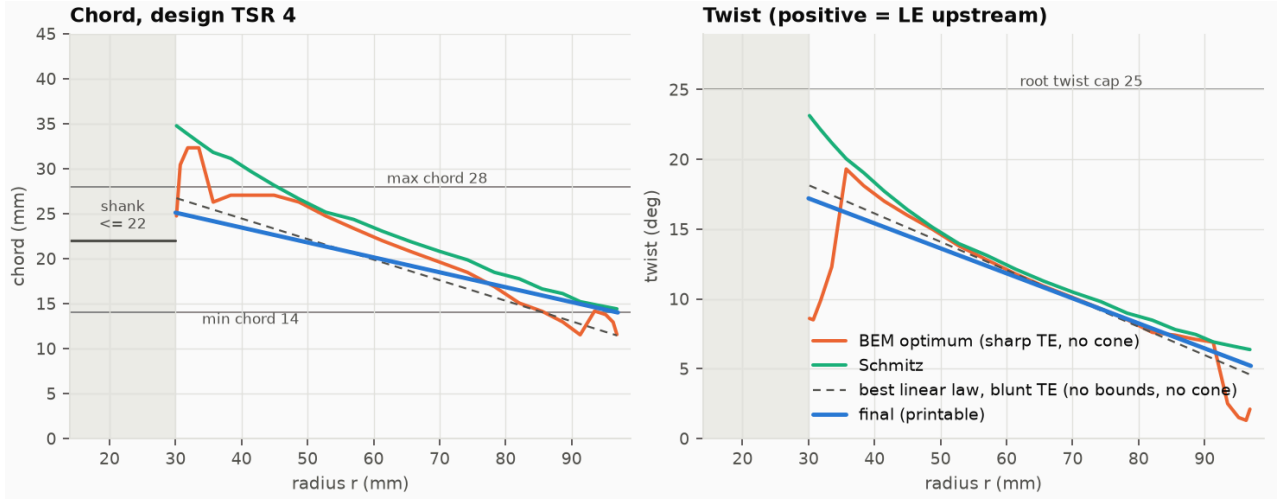


Figure 3: Chord and twist along the span: BEM optimum, Schmitz, the best unbounded linear law and the final printable blade. The shaded band is the shank between the hinge and the aerodynamic root.

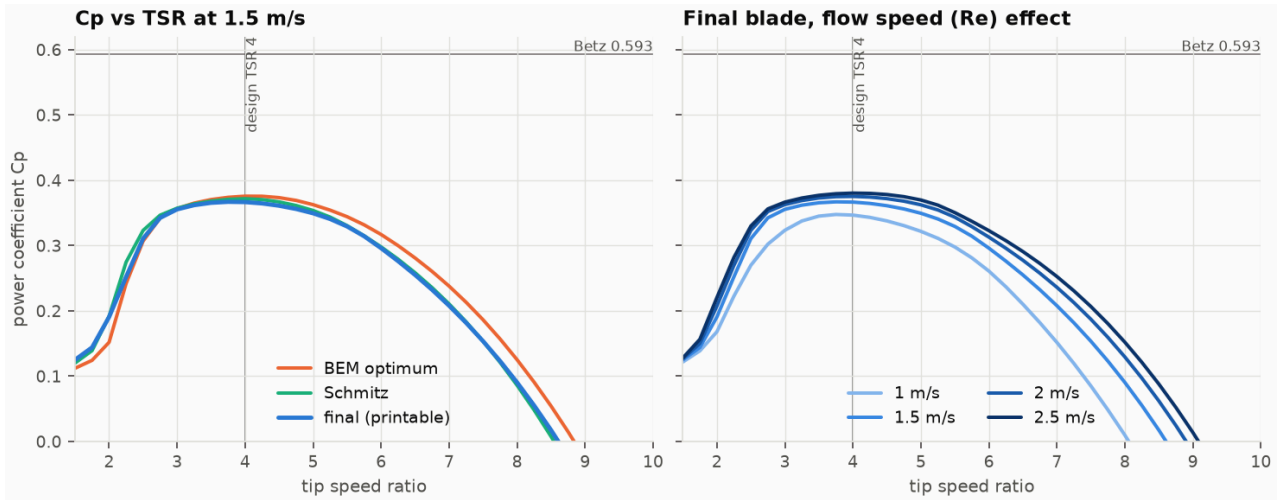


Figure 4: Power coefficient against tip speed ratio (BEM^{est}). Left: the final blade against the two references at 1.5 m/s. Right: the final blade from 1.0 to 2.5 m/s; C_P rises with Reynolds number.

4.4 Performance

At 1.5 m/s the model gives 18.3 W^{est} at 564 rpm and 0.310 N m, against 49.9 W in the swept area (Equation 1). The peak C_P climbs from 0.348 at 1.0 m/s to 0.380 at 2.5 m/s as the Reynolds number rises. Unloaded, the rotor runs away to 2236 rpm (TSR 9.09) at 2.5 m/s; over all TSRs up to runaway the largest thrust at 2.5 m/s is 70.3 N and the largest flap moment 1.379 N m per blade, the design loads for the hub, bearings and tow eye.

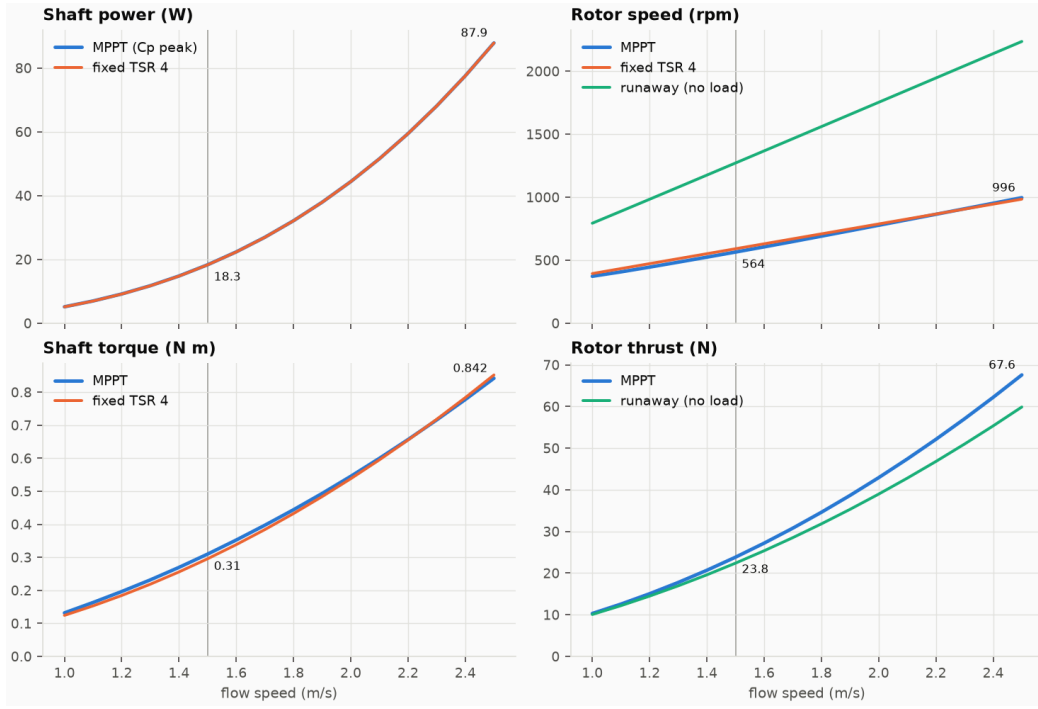
4.5 Generator coupling and the load line

A generator does not run the rotor at its C_P peak. With the hobby convention for the motor constant K_V (rpm per volt of line-to-line peak back-EMF), the torque constant and the losses are

$$K_{t0} = \frac{60}{2\pi K_V}, \quad P_{cu} = R_{ll} \left(\frac{Q - Q_d}{K_{t0}} \right)^2, \quad P_{net} = (Q - Q_d)\Omega - P_{cu}, \quad (11)$$

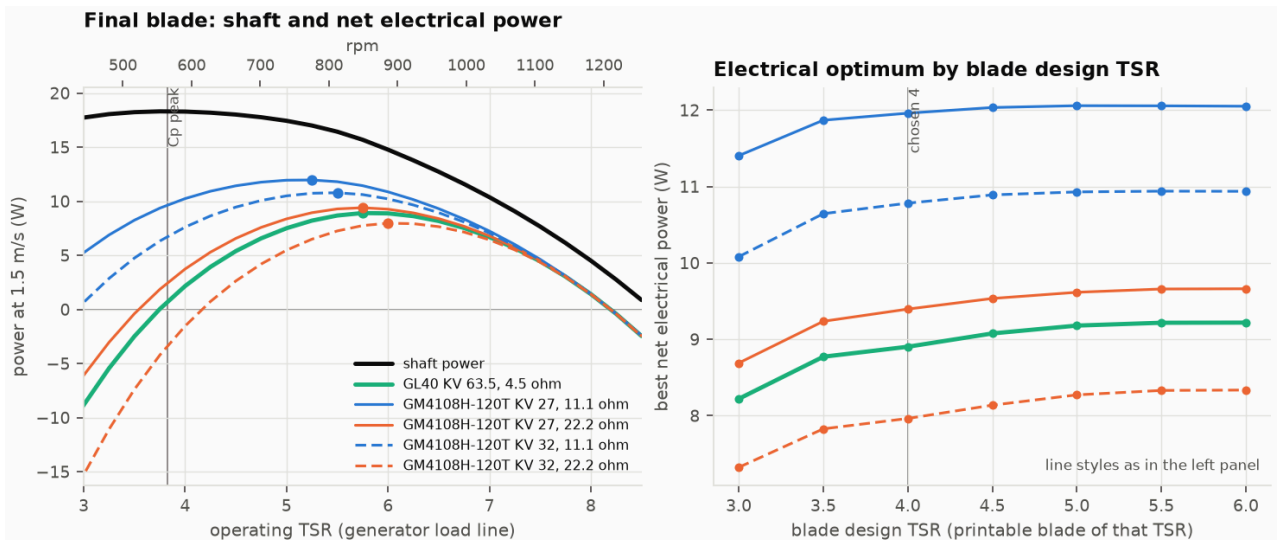
Table 5: Final blade at its C_P peak (maximum power point), BEM estimates^{est}. The flap moment is about the hinge pin and pushes the blade onto its deployed stop.

Flow (m/s)	TSR	rpm	C_P	shaft power (W)	torque (N m)	thrust (N)	flap moment per blade (N m)
1.0	3.78	372	0.348	5.1	0.132	10.3	0.198
1.5	3.82	564	0.367	18.3	0.310	23.8	0.456
2.0	3.95	777	0.375	44.4	0.545	42.9	0.821
2.5	4.05	996	0.380	87.9	0.842	67.6	1.295

Figure 5: Shaft power, rotor speed, torque and thrust against flow speed (BEM^{est}).

where R_{ll} is the line-to-line winding resistance and Q_d a constant iron and friction drag torque. Copper loss grows with the square of the torque, so a generator with appreciable resistance prefers a higher TSR, where the same power comes at lower torque. For the recommended CubeMars GL40 (Section 6.4) the estimate at 1.5 m/s is stark: at the C_P peak the copper loss is 16.8 W and the net output 0.0 W^{est}. The best net point is at TSR 5.75, 849 rpm and 0.176 N m, giving 8.9 W^{est} (15.7 W shaft, 4.6 W copper, 2.2 W drag). The operating point is therefore set by the generator, not by the blade.

This raised a design question: should the blade itself be designed for TSR 5 to 6? The right panel of Figure 6 answers it. A blade designed for the electrical optimum gains at most a few percent of net power, less than the polar uncertainty, while it gives up root chord (a weaker lug transition) and C_P peak. TSR 4 stays the right blade, and it remains the best choice if a lower-resistance generator moves the electrical optimum back toward the C_P peak.

Figure 6: Left: shaft and net electrical power of the final blade at 1.5 m/s against the operating TSR for the generator readings of the trade study (estimates^{est}). Right: best net power of the printable blade designed for each TSR.

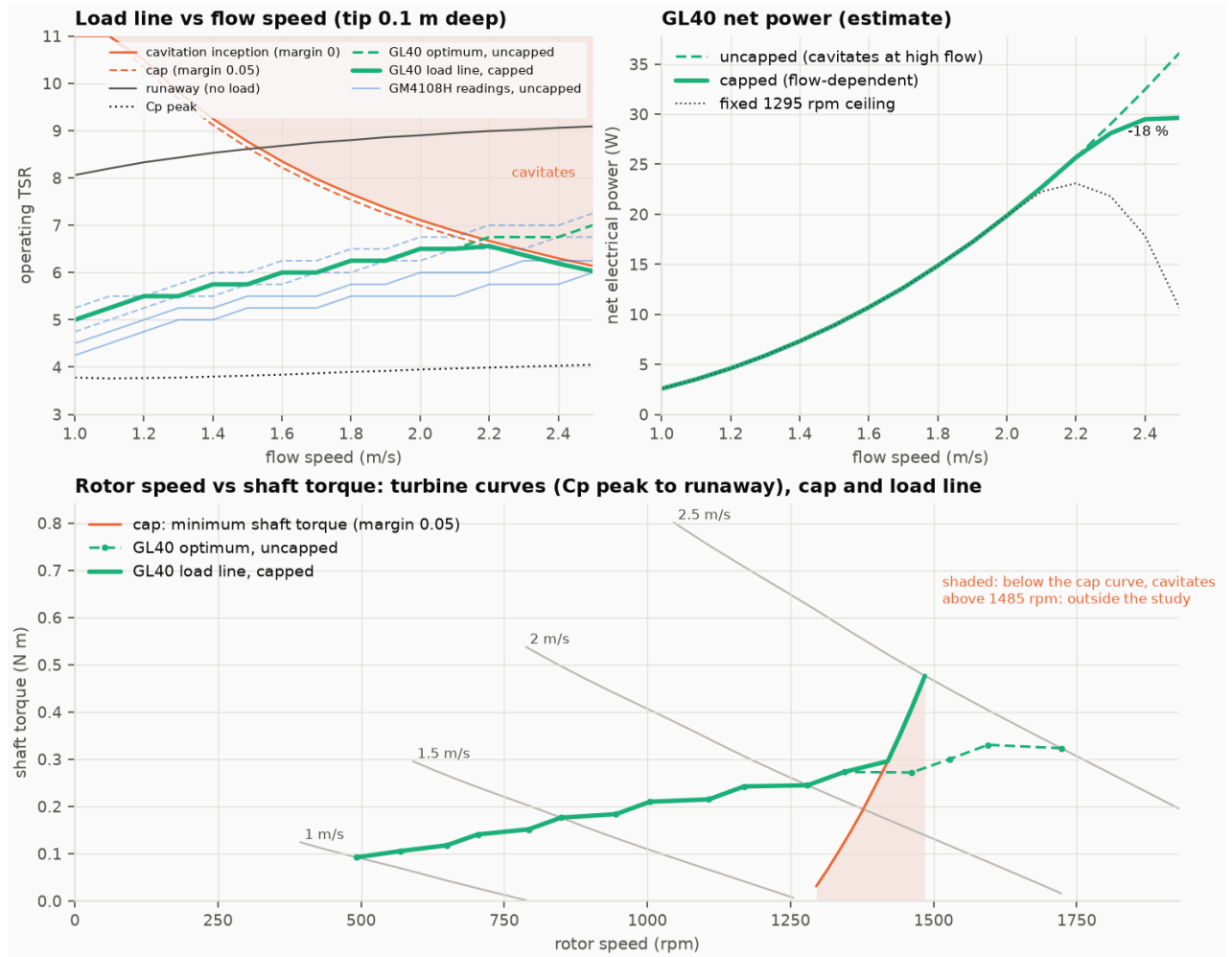


Figure 7: GL40 load line against flow speed with the cavitation cap (estimates^{est}). Top left: operating TSR, cavitation limits and runaway. Top right: net electrical power, uncapped, capped and with a single rpm ceiling. Bottom: the cap in the rotor speed and shaft torque plane, where a controller can apply it.

4.6 Cavitation cap

Near the free surface a fast blade tip can cavitate. The inception criterion compares the cavitation number with the minimum pressure coefficient of the section [9]:

$$\sigma_v = \frac{p_{\text{atm}} + \rho gh - p_v}{\frac{1}{2}\rho W^2}, \quad m = \sigma_v + C_{p,\min} \geq 0.05, \quad (12)$$

with the tip at its highest point, 0.1 m deep, W the local relative speed and $C_{p,\min}$ from NeuralFoil. The margin $m = 0.05$ covers the difference between NeuralFoil and an inviscid panel solution of the tip sections (0.02 to 0.04). At the C_P peak there is no cavitation at any flow speed (margin 1.13 at 2.5 m/s). But the cavitation-free TSR falls quickly with flow, from 8.77 at 1.5 m/s to 6.14 at 2.5 m/s, while the generator's electrical optimum rises with flow. They cross at 2.2 m/s (Figure 7, Table 6).

Table 6: GL40 load line on the final blade, tip 0.1 m deep (estimates^{est}).

Flow (m/s)	C_P -peak TSR	optimum TSR (rpm)	net (W)	cav. margin	cav.-free to TSR	capped TSR (rpm)	capped net (W)	cap cost
1.0	3.78	5.00 (492)	2.6	+7.39	none	5.00 (492)	2.6	0
1.5	3.82	5.75 (849)	8.9	+1.95	8.77	5.75 (849)	8.9	0
2.0	3.95	6.50 (1280)	19.9	+0.29	7.11	6.50 (1280)	19.9	0
2.1	3.97	6.50 (1344)	22.7	+0.18	6.88	6.50 (1344)	22.7	0
2.2	3.99	6.75 (1462)	25.7	-0.04	6.67	6.56 (1420)	25.7	0
2.3	4.01	6.75 (1528)	29.0	-0.12	6.48	6.37 (1441)	28.1	-3.1 %
2.4	4.03	6.75 (1595)	32.5	-0.19	6.30	6.19 (1463)	29.5	-9.2 %
2.5	4.05	7.00 (1723)	36.1	-0.34	6.14	6.03 (1485)	29.6	-18.0 %

The recommended control law tracks the electrical optimum but never exceeds the cap. It changes nothing up to the crossing and costs 18.0 % of net power at 2.5 m/s (29.6 W instead of 36.1 W^{est}, at TSR 6.03 and 1485 rpm). The controller does not know the flow speed, but the cap can be written without it, as a minimum shaft torque at a given speed: above 1295 rpm the shaft torque must stay above a curve rising from 0.031 N m to 0.477 N m at 1485 rpm. At a fixed speed less torque means less flow, a higher TSR and a stronger suction peak on the tip's pressure side. This rule classifies all 306 sweep points from the C_P peak to runaway correctly (0 misclassified). A simple rpm ceiling would be a poor substitute: it would cost 70.6 % at 2.5 m/s. The rotor must never be left to free-wheel when towed at 1.6 m/s or more; it must stay loaded or be braked electrically (Section 6.4).

4.7 Loads, structure and the hinge geometry

The flap moment about the hinge is the dominant blade load and it acts in the direction the folding concept needs. In the 22 × 8 mm lug that carries it the bending stress is 2.3 MPa^{est} at 1.5 m/s and 6.5 MPa^{est} at 2.5 m/s (bending across the 8 mm depth, no stress concentration). The blade prints with its span in the layer plane so this stress runs along the filament. The study first used a straight 12° pitch axis; the layout (Section 5) then solved a curved, hull-following axis. Flying the same blade on that axis leaves the C_P peak unchanged (0.3802 against 0.3804 at 2.5 m/s^{est}) and lowers the largest flap moment from 1.379 to 1.343 N m^{est}; the hub and stop are sized with the latter.

4.8 Verification and sensitivity

The solver was checked against theory and against itself:

- **Glauert optimum rotor** (ideal polar, no losses, TSR 4): BEM C_P 0.51847 against the theoretical 0.51847.
- **Betz limit**: the largest C_P in any sweep is 0.3804, below 0.593.
- **Induction**: mid-span axial induction 0.32 to 0.37 at the design point, near the ideal 1/3; the Buhl branch acts on 12 of 24 elements, at the root and tip where F is small.
- **Convergence audit**: 19 unconverged element solves out of 30,648 (1277 operating points), all in the two root elements, which carry at most 1.15 % of the torque in any power-producing point. None is at a design or maximum-power point of the final blade.
- **Polar confidence**: NeuralFoil's own confidence is at least 0.981 over the design-point elements; the angle of attack at mid-span is 2.6° to 4.3°, at least 7.2° from stall.

The sensitivity study (Table 7) shows where the uncertainty lies. Cone angle, the curved axis and the element count move C_P by 0.2 % or less. Transition modelling dominates: a clean inflow ($n_{crit} = 9$) and forced transition at 5 % chord (a rough print) cost 6.1 % and 5.8 %. The nominal C_P should therefore be read as 0.367 with an uncertainty of about 6 % from the polars alone, before any effect of the hull, which the model ignores (no blockage, no hull wake at the rotor plane). The hub-loss placement is conservative: without it C_P would be 5.0 % higher.

5 Fold kinematics, pockets and detents

5.1 Hinge and the stowed blade

The hinge position was chosen by a sweep over its axial station, its radius and the end of the static tail (ovo/layout.py). The rule admits only geometries in which the pin keeps at least 2 mm of hub material around it, the folded heel stays inside the spinner, the heel face holds a magnet, the hub wall over the shaft bore keeps 1.2 mm and the folding root clears the static tail by 0.8 mm; among those it maximises the smallest margin. The result

Table 7: Sensitivity of the peak power coefficient (BEM estimates^{est}).

Case (peak C_P at 1.5 m/s)	C_P peak	TSR	change
nominal (12° cone, n_{crit} 5)	0.3670	3.82	+0.0 %
cone 0° (straight axis)	0.3663	3.84	-0.2 %
cone 8° (straight axis)	0.3667	3.83	-0.1 %
cone 15° (straight axis)	0.3673	3.81	+0.1 %
hull-following curved axis (estimate)	0.3668	3.83	-0.1 %
hub-loss radius 23.4 mm (hub surface)	0.3809	3.79	+3.8 %
no hub loss	0.3856	3.76	+5.0 %
n_{crit} 9 (clean inflow, smooth blade)	0.3446	4.03	-6.1 %
forced transition x/c 0.05 (rough print)	0.3459	3.69	-5.8 %
48 elements (discretisation)	0.3667	3.82	-0.1 %

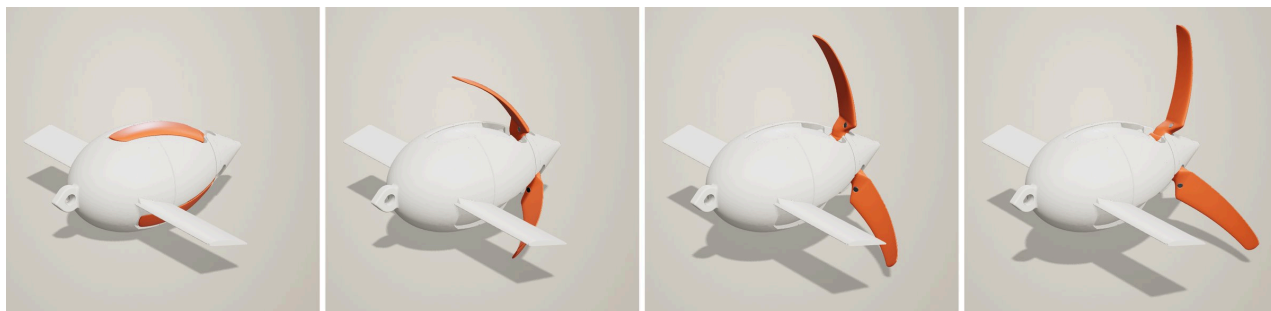


Figure 8: Fold sequence rendered from the CAD assembly at the stowing index: stowed, 60°, 30° and deployed (fins fitted; camera fixed on the static shells).

is a pin at $x = 116$ mm, $r = 14$ mm, with the static tail ending at $x = 108$ mm: pin cover 2.46 mm, heel 1.35 mm inside the spinner when stowed, 4.88 mm of flat heel face, hub wall 1.24 mm. The blade folds 90° and its tip lands at $x = 31.75$ mm.

The stowed shape is solved rather than drawn. For every station the aft offset of the pitch axis is chosen so that the folded section's top sits on the hull surface: the stowed blade stands at most 0.065 mm proud of the hull. An in-plane lead of the stacking axis, 11.5 mm at the root falling to 4.8 mm at the tip, shifts the sections along the hull so their twist does not dig deep pockets; the BEM model ignores this lead (Section 12). The deployed cone that results averages 12.6° from hinge to tip, close to the 12° the blade was designed with.

5.2 Pockets

Three pockets in the static hull, from $x = 30.75$ to 108 mm, take the folded blades. They are 3.02 mm deep at the equator, at most 6.82 mm deep under the aerodynamic span and 11.52 mm in the root notch, with at least 26.6 mm of hull land between neighbours. Because both shells print with +x down, each pocket floor is cut so that it rises at most 45° going forward, and the pocket's forward end and the root-notch ramp are 45° ramps: the shells print without supports. On the CAD solids the stowed aero span clears its pocket by 0.337 mm and the root clears the static tail by 0.899 mm (Section 8). The pockets also index the rotor in the bag: stowed, it cannot turn.

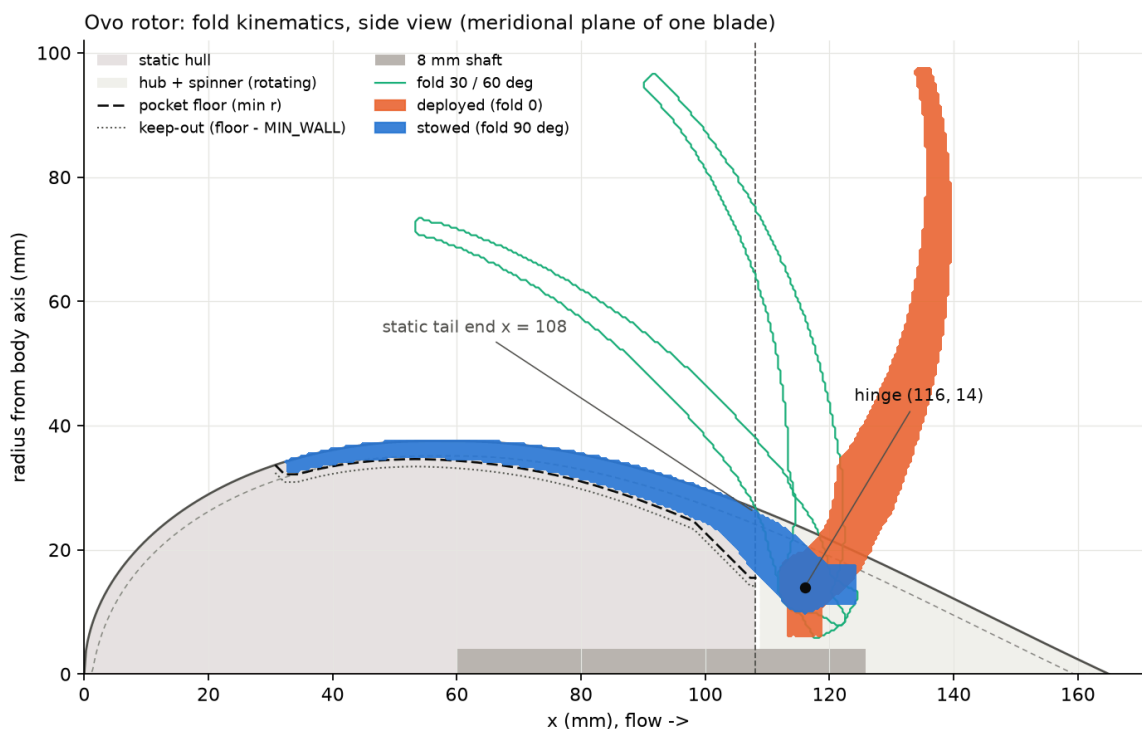


Figure 9: Fold kinematics in the plane of one blade (layout study, docs/figures/layout_fold_side.png): deployed, outlines at 30° and 60°, and stowed over the pocket floor (dashed) and its keep-out (dotted).

Ovo rotor seen from aft (looking upstream)

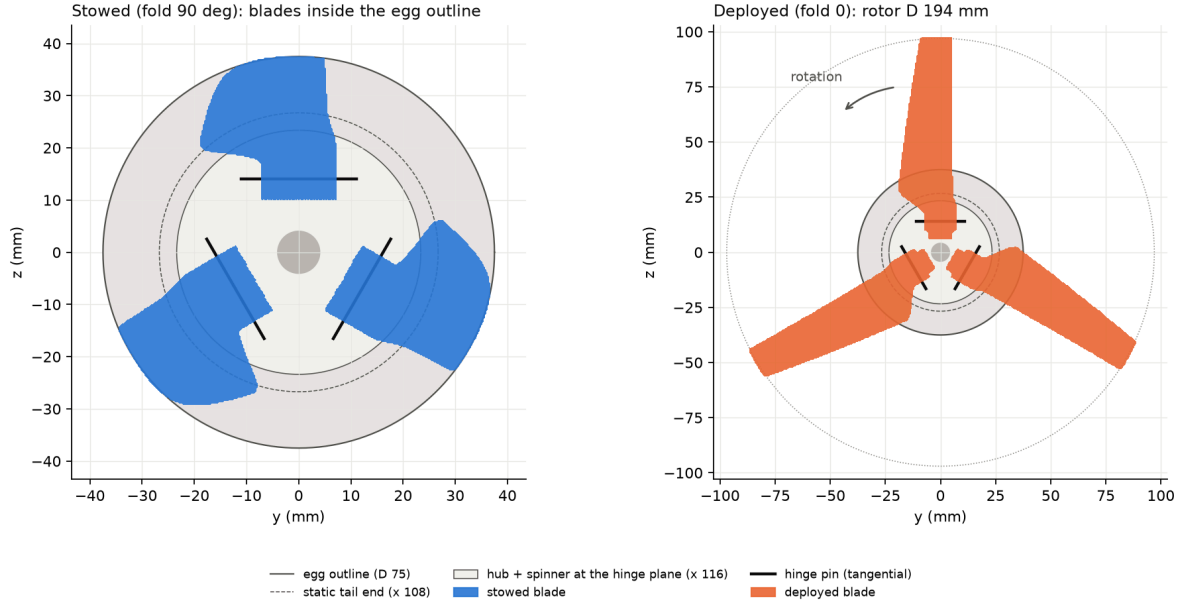


Figure 10: The rotor seen from aft: stowed, the three blades stay inside the egg outline; deployed, they sweep 194 mm. Pins are tangential, so each blade folds in its own meridional plane.

5.3 Detents and hub loads

Magnets replace springs. Deployed, an N52 block ($6 \times 2 \times 2$ mm) in the blade heel faces one in the hub's stop face across 0.4 mm, about $2.32 \text{ N}^{\text{est}}$; stowed, a 4×2 mm disc in the blade shank faces one in a boss of the aft shell, 0.420 mm apart and 1.66° out of line, about $2.27 \text{ N}^{\text{est}}$. Forces come from a dipole model scaled to the catalogue pull of the 4×2 disc (factor 0.63); they must be confirmed on a test print. At 2.5 m/s the deployed stop carries $237 \text{ N}^{\text{est}}$ at 4.86 MPa contact pressure, the 3 mm pin 18.4 MPa in double shear and the printed ears 11.7 MPa in bearing, all from the curved-axis flap moment.

The blades cannot be folded home with the shells being closed over them. A closing sweep on meshes of the CAD solids (bayonet entry, push and 35° twist, both senses) shows that the front shell passes the blades only if they are folded no further than 85° (opened at least 5°) with the fins off, or 50° (opened at least 40°) with the fins on; the assembly sequence encodes this.

6 Layout and drivetrain

6.1 Axial stack

Figure 11 shows the merged layout. The generator sits in the front shell on a twist-lock carrier; the shaft runs from a claw coupling at the equator through two bearings in a tube of the aft shell to the hub. The main stations are: generator bay D50.5 $\times$ L24 at $x = 22.5$ to 46.5, equator joint at 57.5, 608 bearing at 63 to 70, 688 bearing at 96 to 101, lip seal at 102.5 to 107.5, static tail end at 108, hub from 108.6 to the shaft end at 128 mm. The two halves of the layout (rotor and internal) are checked against each other by 30 interface checks; 30 pass.

6.2 Bearings, shaft and thrust path

The rotor is downstream, so its thrust pulls the hub and shaft aft. A set-screw collar (Ruland MSC-8-SS) inside claw B bears on the inner race of the forward 608-2RS, which locates the shaft; the rear 688-2RS floats. Their bores step from 22 to 18 to 16 mm so both press in from the front on crush ribs, and the 16 mm seat of the 688 is also the lip-seal bore. The shaft is an 8 mm SUS440C rod (MISUMI PSSFJ class, about 56 HRC, g6) [10], hard enough for the seal lip.

The bearing span is 32.0 mm against a 37.8 mm overhang to the rotor's thrust centroid, a ratio of 0.85 where a common rule of thumb asks for 1.25. The static tail cannot be lengthened without failing the hinge sweep, so the rule was replaced by the explicit checks of Table 8 for the tow case: thrust $73.9 \text{ N}^{\text{est}}$ from $C_T = 0.8$ at 2.5 m/s, a radial load of 10 % of it and one blade 20 % off its share, all assumptions.

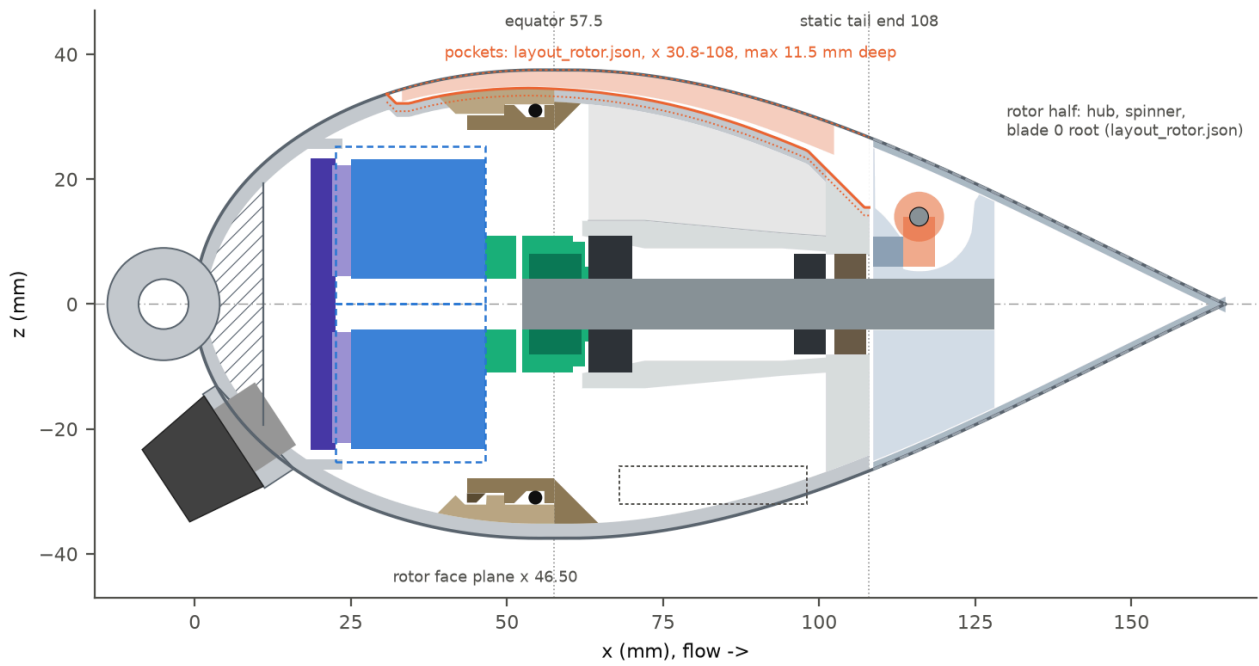


Figure 11: Merged layout, side section (from docs/figures/layout_section.png). Top half in the plane of blade 0 with the deepest pocket floor and the stowed blade (salmon); bottom half between pockets. Blue: GL40 generator in its $D50.5 \times L24$ bay (dashed); purple: twist-lock carrier and adapter; green: claw coupling with the thrust collar; charcoal: 608 and 688 bearings; brown: lip seal; tan: socket and spigot with the O-ring (black); light grey: bearing tube and ribs; orange: blade knuckle on its hinge pin. The section shows the v2 nose with the M12 cable gland; the display build has a clean nose with the tow eye only.



Figure 12: Cutaway render of the CAD assembly: generator replica (blue) on its carrier, claw coupling, shaft, 608 and 688 bearings, lip seal, and the v2 cable gland in the nose.

Table 8: Bearing checks replacing the span-to-overhang rule (layout study; loads are estimates^{est}).

Criterion	limit	value
bearing span	at least 4 shaft diameters, 32 mm	32.0 mm
blade-tip axial runout from 0.05 mm seat slop	at most 0.5 mm	0.30 mm
static safety C_0/P_0 , 608 / 688	at least 4	28.5 / 22.1
shaft deflection at the load plane	at most 0.05 mm	0.015 mm
span / overhang	1.25 (rule of thumb, reported)	0.85

The loads are small for these bearings: their static ratings exceed the largest thrust at 2.5 m/s 19.7 times (608, $C_0 = 1383$ N [11]) and 8.3 times (688 [12]). The open question is stiffness, not strength: rotor wobble should be measured on the first print (Section 12).

6.3 Seals, joint and bayonet

The v2 shaft seal is a TC 8×16×5 NBR lip seal fitted 0.5 mm inside the static tail face. Its lip runs at 0.314 m/s at 750 rpm, so wear is negligible; the installation guideline asks for a shaft of at least 45 HRC [13], which the hardened rod meets.

The equator joint is a radial static gland. The aft shell carries a spigot with the O-ring groove, the front shell a socket bore. The ring is an ISO 3601-141 (AS568-141, 58.42 × 2.62 mm, EPDM 70 for water) [14], the largest 2.62 mm section whose socket keeps at least 1.2 mm of wall under the blade pockets with a 1 mm reserve on the pocket depth: the wall under the pockets is 2.44 mm. Groove bottom D59.88, spigot OD 63.89, bore D64.09, radial gland depth 2.105 mm. A rectangular groove would leave an unprintable flat overhang on the spigot, which prints tip-up, so the groove has a radial aft flank and a 45° tip-side flank (bottom 2.9, top 4.905 mm). With squeeze $s = (d_2 - h)/d_2$ (d_2 the cross-section, h the gland depth) and fill the ring area over the groove area, the worst case combines the ring tolerances, printed diameters within ±0.10 mm and full eccentricity:

- squeeze 18.7 % nominal, 7.7 % to 29.0 % worst case (guidance 15 to 30 % [15]);
- fill 63.4 % nominal, 79.7 % worst case (guidance 60 to 85 %);
- stretch 1.4 % to 3.6 % on the groove bottom.

The worst-case minimum squeeze is below the 14 % minimum of the gland table [14]. It is the price of printed tolerances; at a few centimetres of submersion the joint sees well under 0.1 bar, and the squeeze must be checked on a printed test ring. The layout study recomputes the gland independently and agrees with the hardware study to 0.0 mm. Three bayonet lugs lock the shells after a 35° twist against a hard stop that also aligns the pockets across the joint; three flexure tines in the spigot wall click into dimples at the lock angle.

6.4 Generator trade study and the GL40

Although v1 carries no electronics, the bay is sized for a real generator, so a v2 needs no redesign. The study evaluated 12 off-the-shelf motors (Table 9): gimbal outrunners, low-KV drone motors, a frameless torque motor and an industrial flat motor. Each was coupled to the BEM rotor and its operating point searched for maximum DC

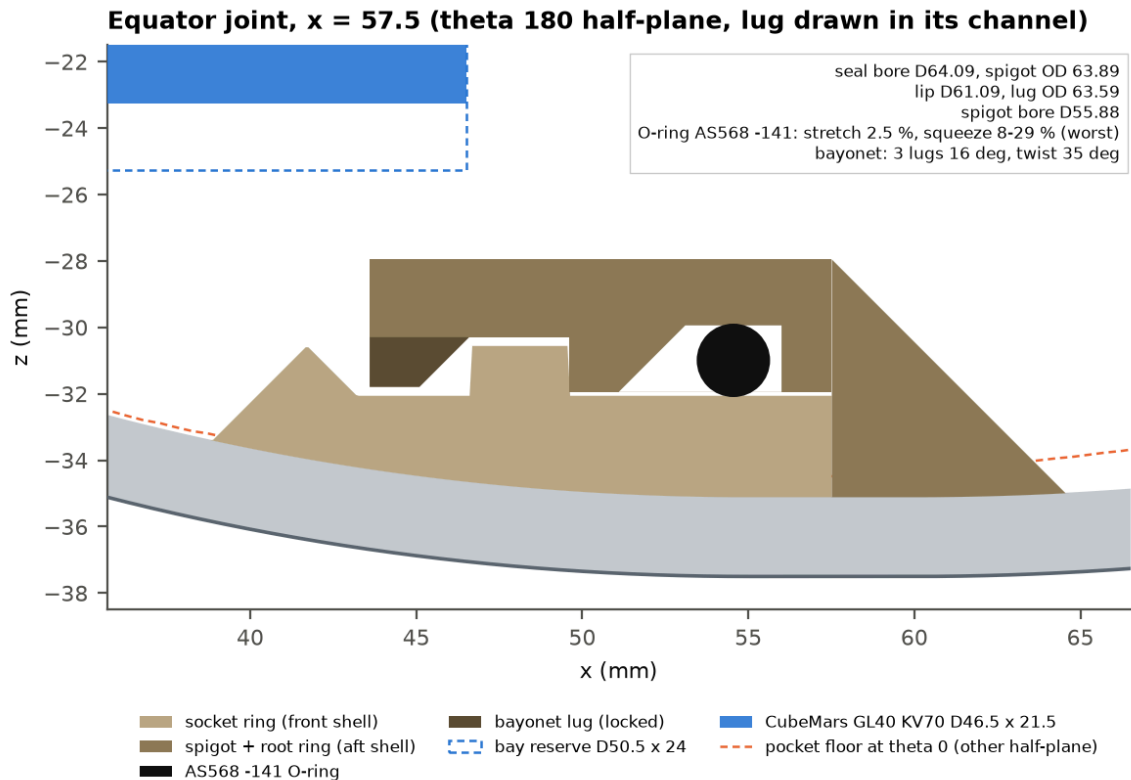


Figure 13: Equator joint section with the -141 O-ring, the bayonet lug and the socket (layout study).

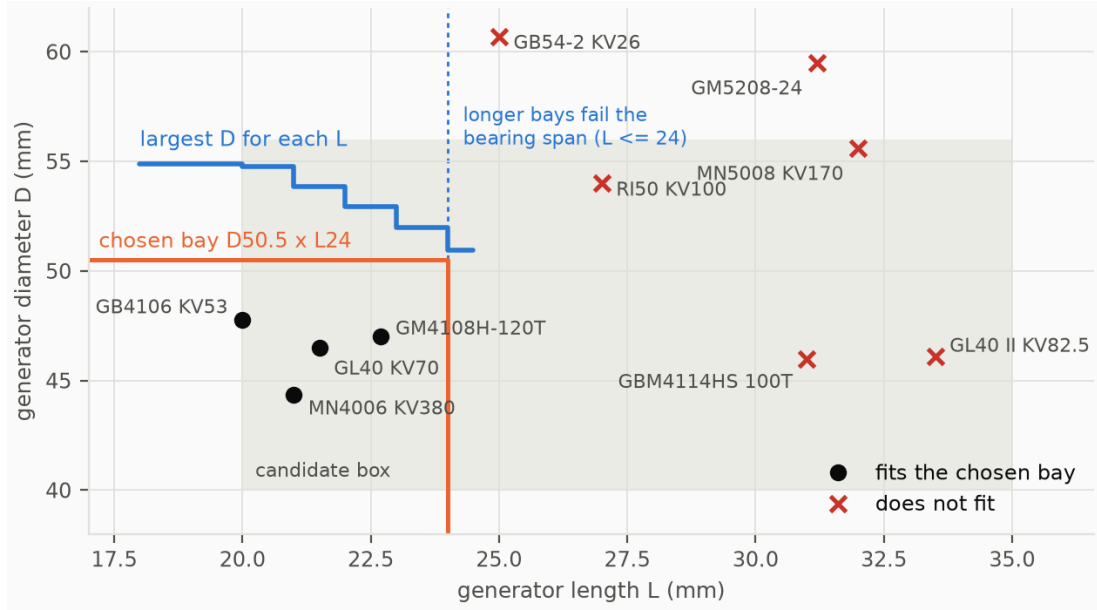


Figure 14: Generator bay frontier: the largest bay diameter for each length (set by the nose wall and the carrier ring), the chosen D50.5 × L24 bay and the catalogue motors; the shaded candidate box spans D 40 to 56 mm and L 20 to 35 mm (layout study, data/layout_internal.json).

power, with three rectifier models: an analytic passive six-pulse bridge (block currents), an ideal active rectifier, and for motors with a published inductance a time-domain bridge model with a capacitor link. Iron and friction losses enter as a constant drag torque from the published no-load input; where none is published the GM4108H value is assumed.

Why the GL40 KV70. It is the only in-envelope candidate whose datasheet is complete and self-consistent [16], [17]: K_V 63.5 rpm/V and the stated torque constant agree ($K_{t0} = 0.1504$ N m/A), the rated 0.25 N m at 1.62 A matches, resistance (4.5 Ω) and inductance (1.8 mH) are given line to line, and a manufacturer drawing gives both mounting faces. Coupled to the rotor at 1.5 m/s it delivers 7.66 W^{est} DC through a passive diode bridge (time-domain model, 7.96 V, 0.962 A, TSR 6.05) and 8.92 W^{est} with active rectification, at 52 % of its rated torque. Its 46.5 × 21.5 mm body and 8 mm through bore fit the bay and the shaft.

An open conflict on the towing range. At 2.5 m/s the electrical optimum would exceed the GL40's continuous rating. Held at the rated current (1.62 A), the rotor would run at TSR 7.51 and give 29.47 W^{est}, but at 2.5 m/s the tips are cavitation-free only up to TSR 6.14 (Section 4.6): the rated-current point cavitates. The cavitation-capped load line avoids that by loading the rotor harder, and from about 2.1 m/s it needs more current than the rating allows (1.65 A, 1.04 × the rated copper loss); at 2.5 m/s it needs 3.0 A, inside the 3.3 A peak rating but 3.44 × the rated copper loss^{est}. Above about 2.1 m/s the GL40 therefore cannot stay inside its continuous rating and keep the tips cavitation-free at the same time. This is an open v2 issue (Section 12).

Honest uncertainty. The ranking among the in-envelope motors is not sharp: with K_m between 0.07 and 0.11 they land within a few watts of each other, and what separates them is voltage, current and data quality. The

Table 9: Generator candidates coupled to the rotor at 1.5 m/s (estimates^{est}; bridge: analytic passive bridge; active: ideal active rectifier). Ranges cover the conflicting datasheet readings. The bay is the D50.5 × L24 envelope of the layout. Recommended choice in bold.

Candidate	D × L (mm)	K_V	R_{ll} (Ω)	K_m (N m/ $\sqrt{W}$)	bridge (W)	active (W)	fits bay	price
iPower GM4108H-120T	47.0 × 22.7	27/32	11.1/22.2	0.063-0.106	7.26-11.23	7.98-11.97	yes	USD 35.90
Rctimer GBM4114HS 100T	46.0 × 31.0	32	15.2/30.4	0.054-0.076	6.07-8.76	6.71-9.56	no	USD 34.99
CubeMars GL40 KV70	46.5 × 21.5	63.5	4.5	0.071	7.78	8.92	yes	USD 72.99
CubeMars GL40 II KV82.5	46.1 × 33.5	82.5/86.8	3	0.064-0.067	6.72-7.13	8.00-8.43	no	USD 133.99
T-motor GB4106 KV53	47.8 × 20.0	53	15.3	0.046	4.80	5.50	yes	USD 45.90
EMAX GB4008 66KV	n/a × n/a	66	7.5	0.053	5.61	6.52	n/a	USD 23.49
CubeMars RI50 KV100	54.0 × 27.0	79.6/100	1.42	0.080-0.101	8.24-10.08	9.92-11.63	no	USD 61.90
T-motor MN5008 KV170	55.6 × 32.0	170	0.27	0.108	9.24	12.19	no	USD 89.99
T-motor MN4006 KV380	44.4 × 21.0	380	0.19	0.057	4.01	7.61	yes	USD 149.90
maxon EC 45 flat 70 W 36 V (411815)	45.0 × n/a	179	1.74	0.040	3.31	4.61	n/a	EUR 226.94
iPower GM5208-24	59.5 × 31.2	20.8	13.7/27.4	0.088-0.124	9.98-12.28	10.63-12.90	no	USD 51.00
T-motor GB54-2 KV26	60.7 × 25.0	26	15	0.095	10.48	11.20	no	USD 74.90

cheaper iPower GM4108H-120T (USD 35.90 against USD 72.99) could deliver 7.26 to 11.23 W^{est} , but its product page [18] contradicts itself: the no-load line implies K_V 27 to 32, which predicts 2.5 to 4.5 times the load torque listed on the same page; if the load point is right, its output at 1.5 m/s collapses to 0.8 to 3.1 W^{est} . The GL40 has its own open points: no no-load loss is published (its drag torque of 0.0248 N m is assumed equal to the GM4108H value), and which part of it carries the 8 mm bore is inferred from the drawing. A ten-minute bench test (line resistance, line inductance, K_V from the open-circuit voltage at a measured speed, spin-down) settles all of it before v2 is frozen.

Towing state. Electrically, an unloaded rotor is harmless: the GL40 reaches 35.2 V^{est} line-to-line peak at the 2.5 m/s runaway speed (the GM4108H 69.9 to 82.8 V), so a charger input rated for 60 V survives open phases. Hydrodynamically it is not: a free-wheeling rotor cavitates from 1.6 m/s (Section 4.6) and lets the fins roll the hull (Section 7). When towed, the charger should therefore keep the rotor loaded. The hardware study sizes a 10.02Ω dump resistor across the DC link for a full battery, absorbing 29.47 W^{est} at 2.5 m/s outside the motor; it is sized for the rated-current point (TSR 7.51), which lies in the cavitating region, so it inherits the conflict above and is not a settled design. A continuous phase short is not an acceptable brake: at 2.5 m/s it would dissipate 69.6 W^{est} in the windings, $5.89 \times$ the rated copper loss, inside a sealed hull.

7 Roll stability and fins

The generator's reaction torque acts on the hull, which hangs on an axial tow line and has almost nothing to resist roll: its centre of gravity is within a fraction of a millimetre of the axis, so the pendulum righting moment is at most 0.0002 N m against a generator torque of 0.1764 N m^{est} at 1.5 m/s. Two fins with opposite incidence supply a counter-torque:

$$Q_f = 2 \cdot \frac{1}{2} \rho V_f^2 S C_L r_{cp}, \quad Q_f/Q_{gen} \in [1.10, 1.155] \text{ at } 1.5 \text{ m/s}, \quad (13)$$

with V_f the local flow speed at the fin, S the area of one fin and r_{cp} the radius of its centre of pressure. The band asks for 10 % more torque than the GL40 load line needs, a margin for a printed Clark Y at low Reynolds number and for the estimated generator drag. The fin study (ovo/layout_internal.py) picks the smallest tip-to-tip span that meets it and clears the pockets and the folding blades: two Clark Y fins of 30×70 mm (20.7 cm² each, aspect ratio 2.30) at $\theta = 87.5^\circ$ and 267.5° , set at 12.5° incidence, 215.2 mm tip to tip in the study (214 mm across the fins in plan as built). Their couple is 0.1946 N m^{est}, $1.103 \times$ the load-line torque, with C_L 0.75 against an estimated three-dimensional maximum of 1.13 at Re 41,300.

A fixed fin set cannot balance every flow speed: the load line raises the TSR with flow, so the generator torque does not grow with V^2 as the fin lift does. The couple ratio is 0.88 at 1.0 m/s and 1.18 at 2.5 m/s^{est}, and since nothing restores roll, any mismatch becomes a slow hull rotation: 4.8 rpm^{est} at 1.5 m/s, 14.1 rpm^{est} at 2.5 m/s. With no electrical load the fins alone would roll the hull at 51 rpm^{est}. Two consequences for v2: the generator must stay loaded, and the tow line needs a roll reference, a two-leg bridle on two nose eyes (about 7.9° of heel^{est} at 1.5 m/s) or a swivel.

The fins slide aft into half-dovetail rails sunk into bosses on the inside of the front shell wall, so nothing stands proud of the hull, and an N52 block pair holds each one. Flush blanking strips close the rails when the fins are off. As built, the tongue has 0.25 mm side clearance, and the measured yaw play of a seated fin ($+1.44^\circ / -1.28^\circ$) spreads the couple ratio over 1.04 to 1.23^{est} around the nominal 1.142: outside the band at both ends. That has no effect on the display build; for v2, crush ribs on the tongue would locate it. Printed tip-down, the fin root sees 4.0 MPa^{est} in the tow case, 0.22 of an assumed 18 MPa interlayer strength.

8 CAD architecture and verification

8.1 Code-CAD from a single source of truth

Ovo is modelled entirely in Python with build123d [19], a boundary-representation library on the OpenCascade kernel. There is no hand-edited CAD file. Every dimension lives in ovo/params.py; the design studies read it and write their results as JSON; the CAD modules read both; one script builds, checks and exports everything (Figure 16). Deviations found during CAD reviews are written back into the parameter file, so parameters and solids agree.

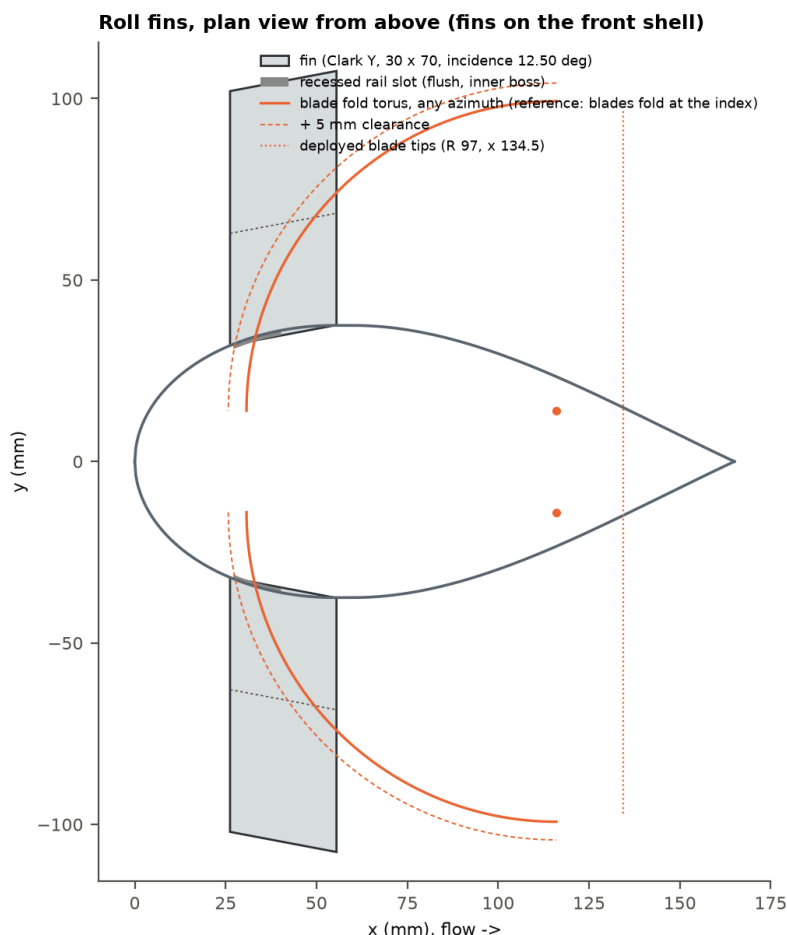


Figure 15: Roll fins from above with the blade fold torus (layout study, docs/figures/layout_section_fins.png, which also shows the view from aft). Folding at the stowing index, the blades clear the recessed rail slots by at least 6.4 mm and the fins by 5.5 mm in 3D.

The part modules are independent: each builds its solids from the parameters and the study tables, runs its own checks and writes them to `out/cad/checks/<module>.json`. The assembly module collects the parts, poses them in each state, adds the detent magnets that no builder models, and runs the integration checks. The blades carry their hinge as data (origin, axis and deployed angle), so the same solid is posed stowed, folded or deployed by one rotation; the integrator checks that this rotation reproduces the deployed pose to 10^{-13} in the transform matrices. A part cache keyed by a SHA-1 hash of the part modules, the parameter and profile files, the layout code and every `data/*.json` file makes stale geometry impossible to reuse, and a `--fresh` build deletes every output the current parts no longer produce. The studies are deterministic in the same sense: re-running the rotor layout reproduced its JSON byte for byte.

8.2 Parts and states

The assembly has 72 parts, 27 printed (from 25 STL files: the three blades share one) and 45 hardware parts (bought items and their sub-parts): printed shells, hub, spinner and cap, three blades, fins and their blanking strips, a printed claw coupling, motor carrier and generator replica, and the bought parts (bearings, shaft, collar,

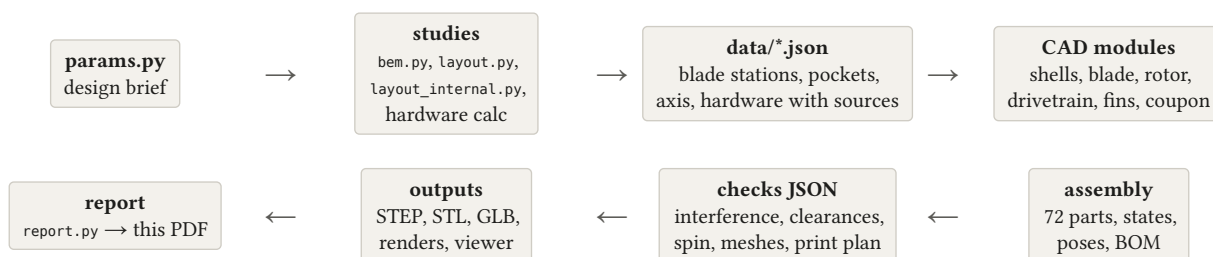


Figure 16: Build pipeline. `scripts/build.py` runs the six part modules, the assembly with its checks and exports, and the renders; `scripts/report.py` turns the JSON outputs into this report.

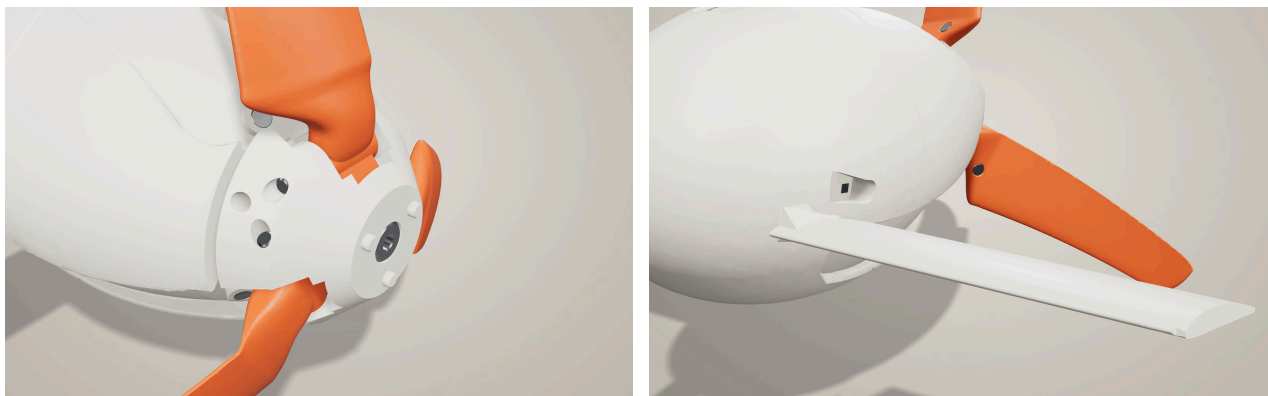


Figure 17: Details of the CAD model (renders, staged poses). Left: the hub with the spinner, its cap and screw hidden and the blades part-folded, showing the ears, the tangential hinge pins, the shaft bore and the stop faces that carry the deployed-detent magnets. Right: a fin slid partly out of its recessed rail to show the half-dovetail slot and the floor magnet; seated, nothing stands proud of the hull (the rail check finds 0.000 mm³ of shell more than 0.05 mm outside it).

pins, screws, magnets, O-ring, lip seal, cable gland). Seven states are modelled: stowed, deployed, folded 30° and 60°, cutaway (a quarter of the shells removed), packed (fins off, strips in) and v2 (front shell with the gland seat and the gland). The display and v2 builds differ only in the front shell: the display shell has a clean nose with the tow eye, the v2 shell adds the M12 gland seat and tap bore.

8.3 Verification

Clearances and overlaps are measured on fine meshes of the CAD solids with manifold3d [20] booleans and minimum-distance queries (0.02 mm chord, 0.03 mm on the shells). In every state, every pair of parts whose bounding boxes come within 3 mm is tested (Table 10). Intended overlaps (press fits on crush ribs, pins in printed ears, screws in their threads) are listed in the code with their reason and reported separately; anything else above 0.5 mm³ is a defect.

Further automatic checks:

- **Spin sweep.** The rotating parts are turned through 360° in 10° steps against every static part nearby: 23 pairs, 0 rubbing; the closest running gap is 0.298 mm (generator_bell / generator_stator).

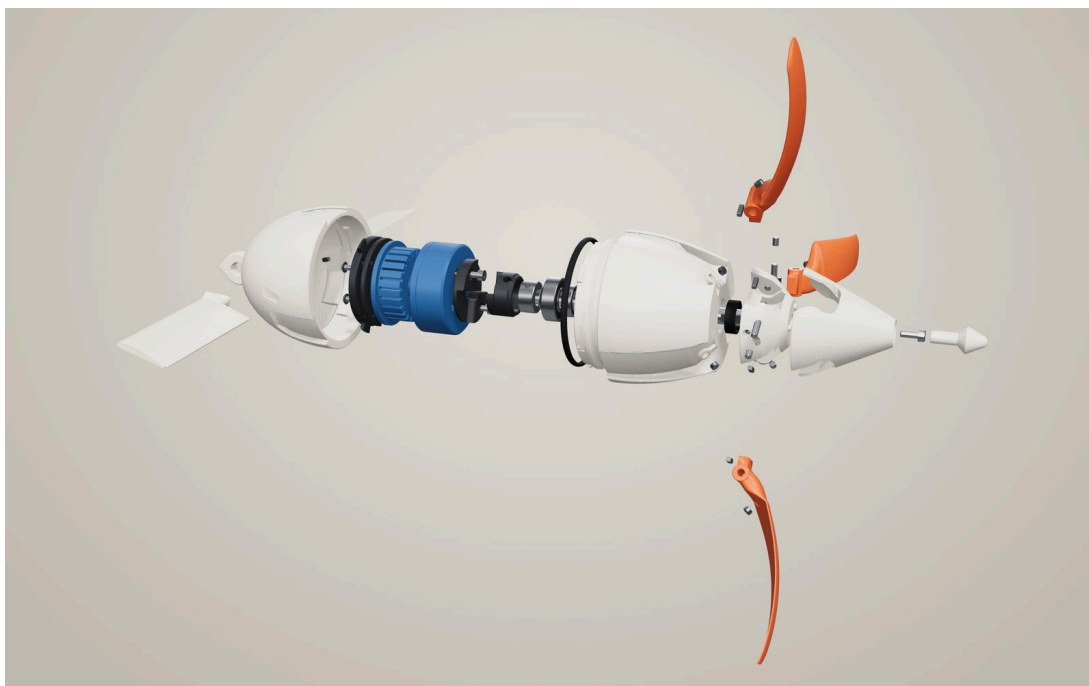


Figure 18: Exploded view in assembly order: front shell and fins, generator stack and claw A, claw B and bearings, aft shell with the shaft, hub with the blades and pins, spinner and cap.

Table 10: Interference matrix by assembly state (out/cad/checks/assembly.json). Static pairs are computed once in the stowed state; the other states add the pairs that involve a part that moves or changes.

State	fold (°)	parts	pairs within 3 mm	intended overlaps	zero-gap contacts	defects
stowed (fold 90)	90	56	261	12	36	0
deployed (fold 0)	0	56	57	0	9	0
fold 30	30	56	69	0	6	0
fold 60	60	56	98	0	6	0
cutaway	0	56	107	2	17	0
packed (fins off)	90	56	14	0	4	0
v2 nose	90	57	24	0	6	0

Table 11: Selected key clearances (14 of 25 checks; all 25 of 25 pass).

Check (CAD meshes)	measured	limit	result
Hub gap, static tail to hub front	0.600 mm	≥ 0.6	pass
Stowed aero span to pocket floors and walls	0.337 mm (blades 0.337, 0.337, 0.337)	≥ 0.3 (-0.05 tessellation)	pass
Stowed root to static tail	0.899 mm (blades 0.899, 0.899, 0.899)	≥ 0.8	pass
Stowed detent magnet pair: gap, angle, force	0.420 / 0.365 mm, 1.66°, 0.00 mm; about 2.27 N (layout 2.32 N at 0.4 mm)	face gap ≤ 0.45 mm, axis angle $\leq 2.0^\circ$, force ≥ 2.0 N	pass
Deployed detent pair: gap, force	0.400 mm, about 2.32 N	0.4 ± 0.15	pass
Deployed heel on hub stop face	0.000 mm / 0.000 mm ³	contact, no overlap	pass
Folding blades to fins	5.635 mm at fold 90	≥ 2.0	pass
Folding blades to hub (run clearance)	0.296 mm at fold 6	$\geq \text{RUN_CLEAR } 0.3$ (-0.01 tessellation)	pass
Knuckle bore on hinge pin	0.149 mm at fold 60	> 0.1	pass
Neighbouring blades while folding	≥ 3 mm at every fold	≥ 1.0	pass
Claw coupling dog flanks	0.300 mm	≥ 0.25	pass
Shells locked at the bayonet stop	0.0000 mm ³ , 0.000 mm (lugs on the hard stop)	no overlap	pass
Closing the shells over folded blades	fins off: closes at fold $\leq 85^\circ$ (unfold $\geq 5^\circ$); fins on: fold $\leq 50^\circ$ (unfold $\geq 40^\circ$)	0 mm ³ overlap	pass
Front shell proud of the hull at the fin rails	0.000 mm ³ more than 0.05 mm outside the hull	flush egg	pass

- **Fold and closing sweeps.** Blades are folded from 0 to 90° in 3° steps against the hub, spinner, pins, shells and neighbours, and against the fins at the stowing index; the shells are closed over folded blades along the real bayonet path (Section 5).
- **Print meshes.** The STL audit covers 27 printed parts in 25 files (the three blades share `blade.stl`): 27 of 27 are watertight with consistent winding, have 0 folded sliver pairs in all, match their B-rep volume within 1 % (27 / 27) and fit the bed (27 / 27).
- **Web model.** The viewer scene has 67 meshes and 313,639 triangles (10.48 MB) with 0 folded pairs; a headless browser check confirms that the viewer's folded and deployed blade poses match the CAD within 0.0105 mm.
- **Module checks.** 176 of 176 module interface and feature checks pass (shells 47/47, rotor 25/25, drivetrain 39/39, fins 20/20, coupon 45/45), on top of the 30 layout interface checks.

These checks found real defects during development: blades classified inside-out by the B-rep classifier because of a hooked surface parameterisation, a gland boss that reached into the generator bay, a fin pad that overlapped a pocket once its yaw was modelled, and a coupon check that read the wrong part. Each was fixed at its root and is now guarded by a check.

9 Printability and print plan

The design rules are those of a 0.4 mm nozzle at 0.2 mm layers: walls of at least 1.2 mm (three perimeters), overhangs of at most 45° from vertical without support, and fits taken from printing guides [21], [22]: -0.05 mm per side for press fits on three crush ribs, 0.25 mm per side for sliding fits, 0.15 mm for the blade knuckle on its pin and 0.30 mm between moving bodies. Two tolerance coupons are printed first, in each material and profile; they carry the bearing seats, pin holes, magnet seats in the three print orientations, the hub D-bore, an M12 tap drill and keys for the bayonet and the fin rail, and their result adjusts the fits before the real parts are printed.

Every part has a planned orientation. Both shells print with +x down without supports: the pocket floors and ramps respect the 45° rule, the O-ring groove has its 45° flank, and the only large bridge is a 38.6 mm disc inside the nose (the three bayonet-click tines of the aft shell print with short breakaway ties across their slits). The blades print chord-vertical with the trailing edge down on a sacrificial printed cradle whose rails stand 0.2 mm under the blade, which holds the 0.600 mm trailing edge straight without slicer supports. Display parts are PLA;

Table 12: Print plates on the 256 mm bed (out/cad/checks/assembly.json). Filament masses are estimates from the shell area, the wall count and the infill.

Plate	content	parts	max height (mm)	filament (g)
0	tolerance coupon (PLA; print a PETG copy too)	coupon, coupon_b, coupon_key_tongue_up, coupon_key_tongue, coupon_key_lug	22.0	34
1	shells and strips (ivory PLA)	front_shell, aft_shell, fin_blank_0, fin_blank_1	74.6	128
2	rotor (PLA, each blade on its cradle)	spinner, hub, blade_0, blade_1, blade_2, spinner_cap	42.9	51
3	drivetrain + fins (PETG)	fin_0, fin_1, motor_carrier, adapter_spacer, claw_a, claw_b	73.7	31
4	cutaway shells (PLA, optional display variant)	front_shell_cutaway, aft_shell_cutaway	74.6	101
5	generator replica (blue PLA)	generator_bell, generator_stator	20.0	24
6	v2 front shell (PETG, v2 wet build only: replaces front_shell)	front_shell_v2	74.6	58

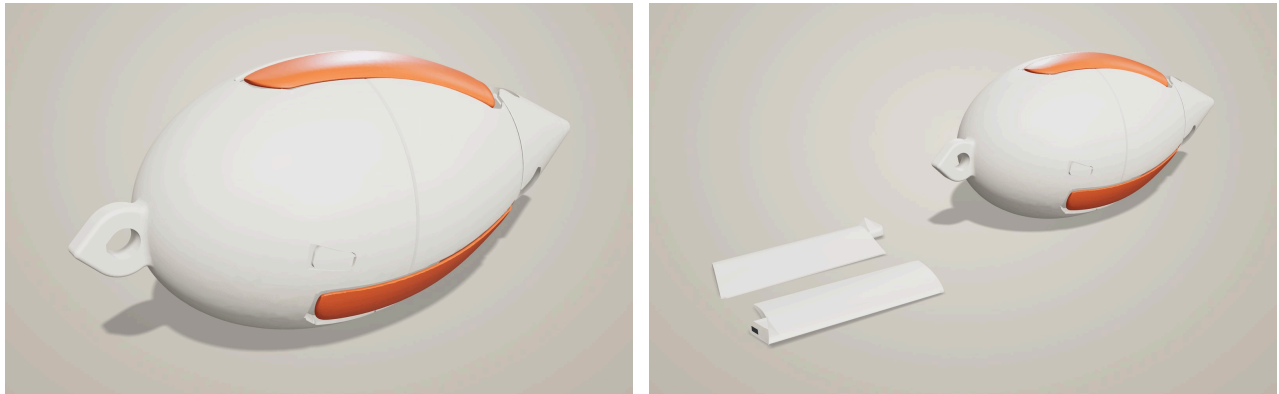


Figure 19: Packed state. Left: blades in their pockets, fins removed and blanking strips in place. Right: the packed egg with the two loose fins.

parts that carry torque, slide or are clamped by a set screw, and the fins, are PETG, the material recommended for wetted parts [23]. A v2 wet build moves the shells and rotor to PETG.

One display build needs about 235 g of filament (204 g PLA, 31 g PETG^{est}), plus brims and the coupons; the cutaway shells and the v2 front shell are alternatives outside that total.

10 Bill of materials and cost

Table 13 lists the bought parts of the as-built assembly, grouped, with unit prices where the hardware study sourced one (USD converted at 1.1225 USD per EUR, ECB reference rate of 2026-10-02 [24]). Printed parts and filament are not priced.

The priced display-build lines of this as-built list total EUR 11.88 pro rata; 11 lines have no sourced price yet (pins, block magnets, shaft, collar, shim, screws, O-ring, lip seal and the seal-fitting sleeve). The hardware study's

Table 13: Bought parts of the as-built assembly (out/cad/checks/assembly.json), priced where a source exists (data/hardware.json).

Bought part	qty	unit (EUR)	total (EUR)	build	price basis
Dowel pin ISO 8734 3 x 22	3	not priced		display + v2	
Dowel pin ISO 8734 3 x 4 (pin retention)	6	not priced		display + v2	
N52 block magnet 6 x 2 x 2 (deployed detent)	6	not priced		display + v2	
M3 screws (grub, cap, button head)	8	not priced		display + v2	
Shaft 8 x 75.5, SUS440C g6, machined flats	1	not priced		display + v2	
Set-screw collar Ruland MSC-8-SS	1	not priced		display + v2	
Shim 8 x 12 x 0.5, stainless	1	not priced		display + v2	
Ball bearing S608-2RS, 8 x 22 x 7, stainless	1	5.02	5.02	display + v2	USD 5.63
Ball bearing SS688-2RS, 8 x 16 x 5, stainless	1	5.02	5.02	display + v2	USD 5.63
Lip seal TC 8x16x5 NBR (v2)	1	not priced		v2	
O-ring ISO 3601-141, 58.42 x 2.62	1	not priced		display + v2	
Heat-set insert M3 x 5.7	2	0.12	0.24	display + v2	EUR 12.09 per 100
Cable gland Hummel HSK-K M12x1.5 (v2)	1	0.93	0.93	v2	USD 10.48 per 10
Brass seal-fitting sleeve (tool, v2)	1	not priced		v2	
N52 block magnet 4 x 2 x 2 (fin rails)	6	not priced		display + v2	
N52 disc magnet 4 x 2 (stowed detent)	6	0.27	1.60	display + v2	USD 0.30 each, MOQ 50
Generator CubeMars GL40 KV70 (v2; display: printed replica)	1	65.02	65.02	v2	USD 72.99

Table 14: Prior art (sources: Estream [25], WaterLily [26], [27], Blue Freedom [28], [29], Aquair UW [30]). The last column assumes an electrical power coefficient of 0.30 to 0.20; for Ovo it is inverted, like the products, from its rated electrical power, the 7.66 W DC estimate (Ovo’s own model reaches that power at 1.5 m/s, an electrical power coefficient of 0.153).

Product	rotor D	rated power	mass	stowed / body	stowage	price	flow for rating (m/s)
Enomad Estream	210 mm	2.5-5.0 W	800 g	245 x 65 mm	folding blades, separate turbine and 6400 mAh battery modules	USD 180	0.99-1.13
WaterLily	n/a (unit 178 mm)	15 W	1300 g	n/a	rigid ducted wheel, tethered or towed, 10 ft cable	USD 159	1.59-1.82
Blue Freedom Portable	120 mm	5 W	690 g	220 x 55 mm	propeller stored inside the body, flexible shaft to the generator body ashore	EUR 299	1.43-1.64
Ampair Aquair UW	318 mm	70 W at 6 mph, 100 W at 9 mph	13,600 g	n/a	fixed prop, pole or tow mount	n/a	1.81-2.07
Ovo (this work)	194 mm	7.66 W DC ^{est} at 1.5 m/s (GL40, bridge)	n/a	75 × 161 mm	blades fold forward into tail pockets	not a product (Section 10)	1.20-1.37

own bill of materials, drawn up before the CAD, gives EUR 15.28 pro rata and EUR 35.48 cash outlay for the display build (minimum orders: 50 magnets, 100 inserts) and EUR 66.01 / 74.42 for the v2 additions, with 11 lines unpriced. It predates three CAD changes: one 608 and one 688 instead of two 608s, 18 magnets of three kinds instead of sixteen discs, and two inserts instead of eight. The generator dominates the v2 cost.

11 Prior art

Table 14 compares Ovo with the portable and towed water generators for which data could be found. Few publish the flow speed at their rated power, so the last column inverts Equation 1: the flow each rotor needs for its rated power at an electrical power coefficient of 0.20 to 0.30.

Read with care: the data quality varies (some values come from reviews, and the WaterLily rotor size is bounded by its housing). The comparison supports three statements. First, per swept area Ovo is in the class of the Enomad Estream, the closest product: both have about 0.2 m rotors and folding blades; the Estream’s rating speed is unpublished. Second, Ovo’s distinctive point is stowage: a closed 0.40 L egg (a 0.71 L bounding cylinder) against the Estream’s 245 × 65 mm tube (0.81 L). Third, its predicted rotor performance is plausible but not exceptional: a full-scale 1.58 m river turbine measured C_P 0.4087 [31], against the 0.367^{est} predicted here for a far smaller, printed rotor at lower Reynolds number. Portable hydrokinetic prototypes are also studied in the literature [32]. What Ovo adds is an open, fully parametric and documented design.

12 Limitations and v2 roadmap

12.1 Limitations

Models, not measurements. Every power, load and mass figure marked ^{est} comes from a model:

- BEM is a quasi-steady annulus model: no hull blockage or hull wake, no yaw, shear or turbulence, rigid blades, coning by the independence principle, and no in-plane lead of the stacking axis.
- The polars are an XFOIL-trained network in a transitional Reynolds range (36,014 to 146,312 over all maximum-power points); transition modelling alone moves C_P by about 6 %, and printed roughness is the main unknown.
- The generator figures rest on datasheet values and an assumed iron and friction drag for the GL40; rectifier switching and cable losses are not modelled.
- Cavitation inception is taken at $\sigma_v = -C_{p,\min}$ with no nucleation delay and no tip-vortex cavitation.
- Structure is checked by bending stresses at the lug and the fin root only: no stress concentration, fatigue or printed-strength data were sourced.

Design margins that are thin or open.

- Bearing span to overhang is 0.85; the explicit checks pass, but wobble must be measured.
- Worst-case O-ring squeeze is 7.7 %, below the 14 % table minimum.
- The v2 hull is nearly neutral (0.009 N^{est} net buoyancy for 389 g^{est}): reaching the +0.3 N target needs 29.7 g less mass or about 31 cm³ of foam float (four-perimeter shells alone save 22.5 g^{est}).
- The fin couple ratio spreads over 1.04 to 1.23^{est} with the rail play, and residual roll needs a bridle or swivel.

- Detent forces are modelled; the stowed hold might resist the flow opening the blades, so flow-driven deployment is a design intent until a water test shows it.
- Towing above about 2.1 m/s is unresolved: the cavitation cap needs more torque than the GL40's continuous rating there ($3.44 \times$ the rated copper loss at 2.5 m/s^{est}), and the rated-current point with its 10.02 Ω dump resistor cavitates. Options for v2: a larger generator, accepting a short-time overload inside the 3.3 A peak rating, a mechanical brake, or limiting towing to about 2.1 m/s.
- The cost has 11 unpriced lines; the hardware study's bill predates the CAD.

12.2 Roadmap

1. **Print and measure the display build.** Coupons first, then the shells, rotor and drivetrain. Measure the fits, the detent forces and the bayonet click; check that the blades fold and deploy by hand, the rotor spins freely and wobble at the blade tips stays near the predicted 0.30 mm.
2. **Bench-test the generator.** Line resistance and inductance, K_V from the open-circuit voltage, no-load loss from a spin-down; re-run the coupled study with the measured values.
3. **Electronics (v2).** A three-phase rectifier and a buck-boost stage to 5 V USB with an input rated for 60 V; maximum-power tracking along the load line, with the cavitation cap as a minimum shaft torque above 1295 rpm, a current limit at the GL40 rating, and a dump load instead of a phase short. The cap and the current limit conflict above about 2.1 m/s, so the towing strategy has to be settled first (Section 12).
4. **Water test.** Tow tests from 1.0 to 2.5 m/s logging flow speed, rotor speed, DC voltage and current to measure the electrical power coefficient; observe blade deployment, roll angle with and without a bridle, and cavitation near the surface.
5. **Design updates from the data.** Float or thinner shells for v2 buoyancy, crush ribs on the fin tongues, and a re-run of the blade design with measured polars or a corrected transition model.

13 Conclusion

Ovo shows in CAD that a rotor $2.6 \times$ the diameter of its body can fold flush into a printable egg when the rotor is placed downstream and its blades fold forward, so that the flow is meant to deploy them and the thrust to lock them; whether the flow does deploy the blades past the stowed detent is the first thing the water test must show. The design is quantified rather than sketched: the cost of every printability constraint is known in power coefficient, the generator choice and its operating law follow from a coupled loss model with a cavitation limit (which leaves the towing range above about 2.1 m/s as an open issue), and the geometric claims about the 72-part assembly are checked by the build itself on fine meshes of the CAD solids. Its predictions are still predictions; the next step is to print it, bench-test the generator and put it in the water.

Colophon. © 2026 Luca Ramírez Olivo. This report, the CAD models and their exports are licensed under CC BY 4.0 (the code under MIT). Built 2026-10-06 by `scripts/report.py` from repository commit `df74660` and CAD build `d821a10403a2`.

References

- [1] D. F. Myring, "A theoretical study of body drag in subcritical axisymmetric flow," *Aeronautical Quarterly*, vol. 27, no. 3, pp. 186–194, 1976.
- [2] T. Burton, N. Jenkins, D. Sharpe, and E. Bossanyi, *Wind Energy Handbook*, 2nd ed. John Wiley & Sons, 2011.
- [3] M. O. L. Hansen, *Aerodynamics of Wind Turbines*, 3rd ed. Routledge, 2015.
- [4] M. L. Buhl, "A New Empirical Relationship between Thrust Coefficient and Induction Factor for the Turbulent Windmill State," technical report NREL/TP-500-36834, 2005.
- [5] S. A. Ning, "A simple solution method for the blade element momentum equations with guaranteed convergence," *Wind Energy*, vol. 17, no. 9, pp. 1327–1345, 2014.
- [6] P. D. Sharpe and R. J. Hansman, "NeuralFoil: An Airfoil Aerodynamics Analysis Tool Using Physics-Informed Machine Learning," *arXiv preprint arXiv:2503.16323*, 2025, [Online]. Available: <https://github.com/peterdsharpe/NeuralFoil>
- [7] M. Drela, "XFOIL: An Analysis and Design System for Low Reynolds Number Airfoils," in *Low Reynolds Number Aerodynamics*, in Lecture Notes in Engineering, vol. 54. Springer, 1989, pp. 1–12.
- [8] R. Gasch and J. Twele, *Wind Power Plants: Fundamentals, Design, Construction and Operation*, 2nd ed. Springer, 2012.
- [9] J. Carlton, *Marine Propellers and Propulsion*, 4th ed. Butterworth-Heinemann, 2019.

- [10] “MISUMI, linear shafts catalogue (PSSFJ, SUS440C, g6).” Accessed: Oct. 03, 2026. [Online]. Available: https://us.misumi-ec.com/pdf/fa/2019/2019_US_0202_003.pdf
- [11] “BearingsDirect, S608-2RS stainless steel bearing 8x22x7 (load ratings, price).” Accessed: Oct. 03, 2026. [Online]. Available: <https://bearingsdirect.com/s608-2rs-stainless-steel-miniature-ball-bearing-8x22x7mm-sealed/>
- [12] “BearingsDirect, 688-2RS miniature bearing 8x16x5 (load ratings, price).” Accessed: Oct. 03, 2026. [Online]. Available: <https://bearingsdirect.com/688-2rs-miniature-ball-bearing-8x16x5-sealed-mr688-2rs/>
- [13] “Oil Seal Stocks, TC rotary shaft seal installation guideline.” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.oil-seal-stocks.com/all-oil-seals/oil-seal-tc-rotary-shaft-seal/>
- [14] “Apple Rubber, ISO 3601 metric size O-rings and static radial gland design.” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.applerubber.com/src/pdf/iso-3601-metric-size-o-rings.pdf>
- [15] “Global O-Ring and Seal, O-ring groove design guide.” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.globaloring.com/o-ring-groove-design/>
- [16] “CubeMars GL40 KV70 product page (KV, resistance, inductance, ratings).” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.cubemars.com/goods-1132-GL40.html>
- [17] “CubeMars GL40 gimbal motor, 2D drawing.” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.cubemars.com/data/cms/202602/gl40-gimbal-motor-2d-drawing.pdf>
- [18] “iFlight iPower GM4108H-120T product page.” Accessed: Oct. 03, 2026. [Online]. Available: <https://shop.iflight.com/ipower-motor-gm4108h-120t-brushless-gimbal-motor-pro217>
- [19] “build123d: a Python parametric boundary-representation CAD library on the OpenCascade kernel.” [Online]. Available: <https://github.com/gumyr/build123d>
- [20] “Manifold: geometry library for topological robustness.” [Online]. Available: <https://github.com/elalish/manifold>
- [21] “Prusa Knowledge Base, Modeling with 3D printing in mind.” Accessed: Oct. 03, 2026. [Online]. Available: https://help.prusa3d.com/article/modeling-with-3d-printing-in-mind_164135
- [22] “Creative3DP, Press-fit tolerances for 3D printing.” Accessed: Oct. 03, 2026. [Online]. Available: <https://tools.creative3dp.com/blog/press-fit-tolerances-3d-printing/>
- [23] “Prusa Knowledge Base, PETG.” Accessed: Oct. 03, 2026. [Online]. Available: https://help.prusa3d.com/article/petg_2059
- [24] “European Central Bank, euro foreign exchange reference rates (2026-10-02).” Accessed: Oct. 03, 2026. [Online]. Available: <https://www.ecb.europa.eu/stats/eurofxref/eurofxref-daily.xml>
- [25] “New Atlas, Estream portable water turbine charger (Enomad).” Accessed: Oct. 03, 2026. [Online]. Available: <https://newatlas.com/enomad-estream-portable-water-turbine-charger/44897/>
- [26] “WaterLily Turbine, paddling charger product page.” Accessed: Oct. 03, 2026. [Online]. Available: <https://get.waterlilyturbine.com/paddling-charger/>
- [27] “World Water Reserve, WaterLily turbine review.” Accessed: Oct. 03, 2026. [Online]. Available: <https://worldwaterreserve.com/waterlily-turbine-review/>
- [28] “Blue Freedom, Portable hydro power plant product page.” Accessed: Oct. 03, 2026. [Online]. Available: <https://blue-freedom.net/hydro/portable/>
- [29] “Gigazine, Blue Freedom review (machine translated).” Accessed: Oct. 03, 2026. [Online]. Available: https://gigazine.net/gsc_news/en/20160418-blue-freedom/
- [30] “No Outage, Ampair Aquair UW water generator.” Accessed: Oct. 03, 2026. [Online]. Available: <https://nooutage.com/aquair-uw.htm>
- [31] E. Chica, L. Velásquez, and A. Rubio-Clemente, “Full-Scale Experimental Assessment of a Horizontal-Axis Hydrokinetic Turbine for River Applications: A Challenge for Developing Countries,” *Energies*, vol. 18, no. 7, p. 1657, 2025, doi: 10.3390/en18071657.
- [32] J. Riglin, F. Carter, N. Oblas, W. C. Schleicher, C. Daskiran, and A. Oztekin, “Experimental and numerical characterization of a full-scale portable hydrokinetic turbine prototype for river applications,” *Renewable Energy*, vol. 99, pp. 772–783, 2016, doi: 10.1016/j.renene.2016.07.065.