HUMAN POWER

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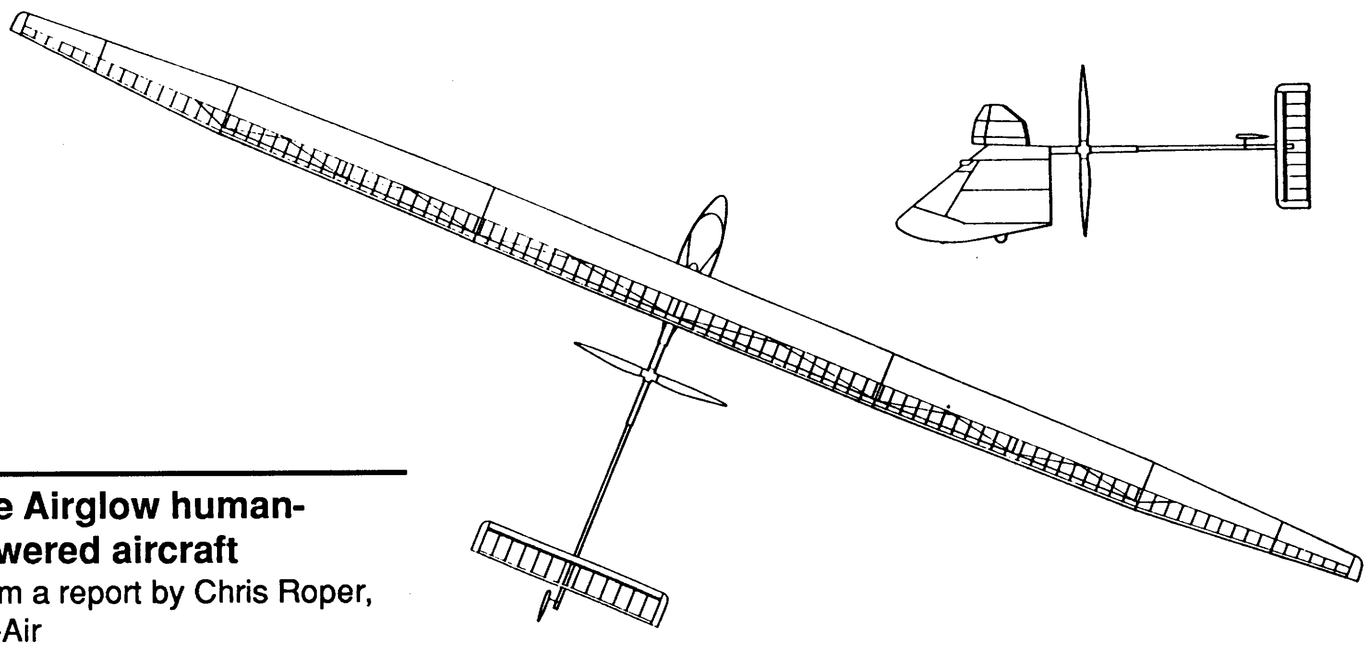

The Airglow human-powered aircraft

From a report by Chris Roper, VP-Air

Airglow was designed and built by John and Mark McIntyre of Cambridgeshire, UK. It flew on its first day of testing, July 20, 1990, piloted by Nick Weston. John has a degree in marine engineering; he accompanied the Daedalus on its epic flight. Mark is a professional model-builder, and he has used remote-control model-aircraft servos for the flying controls. The wingspan is 26m and the total airframe mass around 34 kg. It was designed to withstand 2g. The wing section is DAI1335 and DAI1336, produced by Mark Drela for Daedalus. The propeller is coaxial on the tailboom like that for Bionic Bat, and transmission is via bevel gears and a short length of untwisted 6mm-pitch chain.

(This is taken from a manuscript Chris Roper has written covering all known HP flight; part will appear in a forthcoming book on HPVs in general, and he hopes to find a publisher for his complete manuscript-Ed.) ❑

Pedaling with paddlewheels, or how to build your own

by Robert B. Fearing

The paddlewheel pedal craft outperforms other human-powered water craft in many conditions. Compared with a conventional canoe, it is faster at less effort. It can travel in very shallow water. It can maintain a constant speed against wind and current. It uses the more powerful leg muscles from a comfortable seated position, rather than the arm muscles as in a canoe or kayak.

I have experimented with many paddlewheel drive systems on various water craft. The most successful arrangement is a catamaran 4.9-m (16-ft.) long that can be pedaled by one or two people. I believe that such a craft can be constructed in many home workshops. Design details are given for this arrangement. The historical sequence given below may save others from "reinventing the wheel". The supreme recumbent seat, using proportions recommended by David Gordon Wilson, was used on the second craft and on all the rest, except as noted. All paddlewheels were large enough for the job, unlike the usual "toy" pedal boats. Easy adjustment of paddlewheel dip was recognized early on as an essential design feature.

(continued on page 9...)

Human Power
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Members may purchase additonal copies of *Human Power* for $3.50 each. Non-members may purchase copies for $5.00 per copy.

Special thanks to the authors, Marti Daily, Carolyn Stitson and Leah McGavern, without whom this issue would not have been possible.

Editorials

A sense of (human) power

In no way can the editorship of Human Power be regarded as a bully pulpit. We are not read by the masses, but by a small, highly intelligent, far-sighted group of individuals. (It's always easy to be complimentary about one's own group - but I think that these descriptions are not too far wide of the mark). Therefore it is perhaps not too remarkable that within a week of the last issue coming out I had two responses, from Doug Milliken and Phil Thiel, to an editorial suggestion about the need for more research on various topics. I hope that we will have room for both comments in this issue. In the same week Toshio Kataoka responded to the suggestion that we have more special issues of Human Power by sending material for one that could be devoted to topics on human-powered helicopters. In view of the great efforts that are being made by several groups to win the HP-helicopter prizes, a special issue would be highly appropriate. We need more material: please respond as enthusiastically as you did to the last editorial by sending in contributions. And Michael Eliasohn has almost finished putting together what should be nearly a special issue on front-wheel-drive front-steering recumbents. We have a feast in store!

Rapid and generous responses like these could go to an editor's head. One wonders what hobby-horses one could mention that would produce instant results? Here are two.

In the old days of wind-up watches there were people who maintained that they could not wear wrist watches because something about their bodies' static electricity or magnetic fields prevented the watches from ticking. I found it difficult to believe them. There has been a long correspondence in the magazine of the FCOT–the Fellowship of Cycling Old Timers–on the old equivalent of cyclo-computers, the Lucas odometer, mounted on the front-wheel spindle with a five-pointed star wheel that was driven around by a striker on one of the spokes. Some people had no trouble with them, while others remembered that they never got more than a few hundred miles out of them before they failed. What is relevant to these stories is that my sixth cyclo-computer has just frozen up. I have repeatedly bought new makes and models hoping that I will arrive at one that will last a year and give me good data on how long my tires and chains last, but there always comes a time when whatever new model I've bought just refuses to respond to any input. With some of them I've been able to restart them by taking out the cells and losing all my data, and with others nothing will revive them. Sending them back to the manufacturers usually gets a friendly and uninformative letter and an equally disappointing replacement. Am I the equivalent of the people who couldn't wear wrist watches, or do other people have similar experiences? If so, why?

My second topic on which I would enjoy having reaction, and perhaps a full paper, is on aluminum-alloy components. My crank snapped as I was riding home tonight. I have had many aluminum-alloy components fail similarly on many bicycles, regular and recumbent, over my many years of riding, but someone wrote to me recently who far surpassed me in crank failures. I have preached before about the dangers from fatigue failure of aluminum-alloy frames and particularly forks. We know that aluminum has no fatigue limit as does steel: eventually all aluminum components should, therefore, fail in fatigue, however low the loading. "Alloy" solid cranks were developed as a lighter-weight substitute for solid steel cranks. One conclusion might be to develop tubular lightweight steel-alloy cranks. But then sometimes I find myself flying in a DC3, and reflecting that these were first built in around 1936 and have been used for short-hop trips involving many take-off-and-landing stresses and low-level bumpy flights, and yet the wings don't break off. Nor does anyone propose steel airplanes, despite events like the Aloha tragedy in which part of the fuselage blew away after a fatigue failure. High-performance aircraft are using

In this issue–

carbon-graphite composites increasingly, and yet the fatigue performance of these materials isn't good, either. I need guidance in this jungle of conflicting data, and I suspect that I'm not alone. Input would be appreciated.

–Dave Wilson ❑

Letters to the editor

HP hydrofoils in Russia

It gave me great pleasure to read the article by Alec Brooks, Allan Abbott and David Wilson in one of the 1987 Russian issues of Scientific American. In the early 70s, as an amateur inventor, I was both lucky and unlucky to work on the same problem. While comparing the specific power of various vehicles and craft, I noted that hydrofoils had a relatively small specific-power requirement. Therefore I designed and built a sports hydrofoil craft that promised greater speed on water. In 1978 I built a new version of the rowing craft. I would like to know more about the Flying Fish, and I would like to have expert IHPVA opinions on my ideas.

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Low-energy boats and steam power

With reference to the article on low-energy boats by Theo Schmidt, I, as an active steamboater as well as an HPV nut, although admitting that steam plants don't take any prizes for efficiency, would take exception to his comments that boilers are heavy and large. In fact, the state-of-the-art has been progressing steadily for some time. Examples of current thinking in steamboating can be found in the pages of Model Engineer and Model Boats magazines, both published in the UK.

A curious aspect to steam technology is that at present all steam marine speed records are now held by models, some of which have gone in excess of 100 mph using lightweight flash steam boilers and enclosed-crankcase engines turning at over 10,000 rpm. I know that HPVs are [our primary interest] but experimentation with steam is not standing still. Perhaps a hybrid HPV/steamboat is not out of the question, eh? Good luck!

Edward T. Haas, Jr.
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Definition of Caster

To the best of my knowledge the following definition has never appeared in any cycling-related publication.

CASTER: any wheel that follows its steered axis is a caster. This applies to all machinery whether in a plane, car or HPV.

The lesson from this statement is understanding that it is possible to have more than one caster from a steered axis. A bicycle has two casters. The front one is often referred to as "trail", while the rear caster is rarely dicussed.

In order to develop the measurement for the front caster the physical properties of the rear caster must be identified.

(This topic is covered in a lecture on steering dynamics and rider control, given at the IHPSC in Portland, August 1990).

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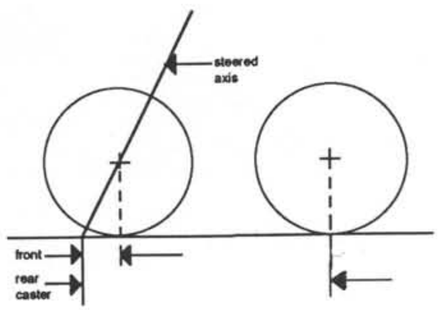

Review

Cycling Science, vol.2/ 4, December 1990.

The first two full articles in this issue are on aerodynamic handlebars–tests showed that they are metabolically as efficient as standard bars–and on soft-shell and hard-shell helmets, which appear to offer different degrees of safety in different conditions, neither type giving an overwhelming advantage over the other. The issue is completed by three articles on food intake and energy expenditure during extreme sustained exercise, including the Tour de France. For me the data in these articles were a revelation. For instances, I did not know that human beings can work for days at a higher level than can be supported by the consumption of regular food alone. Special high-energy low-fibre food is therefore not merely a convenience but a necessity. High-carbohydrate diets for athletes should not include fat (I thought, in my ignorance, that fat is fine if one burns it off fast). The old carbo-loading regimen has been found to be at fault, and a new one is specified. Drinks need to be fortified with glucose, glucose polymers or sucrose, but probably not fructose, and should include electrolytes for ultramarathon exercise. These new findings give me a satisfactory and even exciting explanation of why I have felt so exhausted after past multicentury rides: I have made every possible error in eating and drinking.

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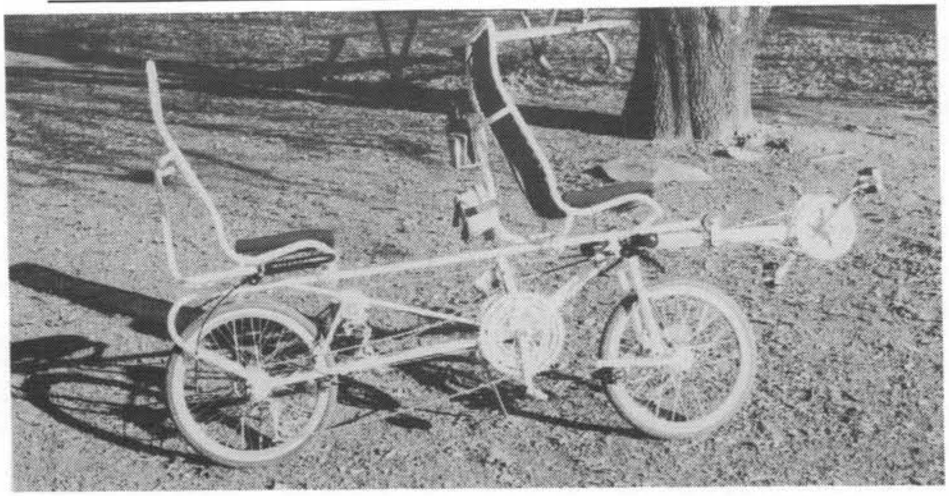
—Dave Wilson ❑



Pedal-power on the French canals

by Philip Thiel

With a ruling depth of less than two meters and a speed limit of six kilometers per hour the smaller canals of France are ideally suited for low-power, slow-speed cruising. Thus, this project for a pedal-powered, screw-propelled, two-person "penichette", classe "escargot"; or snail-class mini canal-barge. It is intended for easy construction at canal side by a group of like-minded people who would enjoy sharing a spring of boat construction and a summer of leisurely fluvial explorations as part of a small flotilla.

Here are my preliminary specifications: a simple, essentially flat-bottomed, square-ended hull with dimensions about 5.2 m (17') by 1.8m (6') wide to be built of exterior-grade plywood and softwood framing. Accommodated under 1200-mm (4') sitting headroom are two berths forward, with an access-hatch over; followed by toilet and hanging space with two louvered doors to provide several different arrangements for privacy; and then a "salon" with table-seating and food-preparation counter. Aft of this is an open cockpit sheltered with a folding Bimini top. An outboard swing-up rudder is controlled by tiller from either of the two side-by-side pedaling positions, and propulsion is provided by two swing-up Seacycle drive units in wells built into the hull and transom.

Essential equipment includes a folding bicycle, to be used for procuring fresh bread, fruit, cheese and wine from the nearest village.

The first phase in this program involves the construction in Seattle of a prototype, to test performance and to check out construction details, time, and costs.

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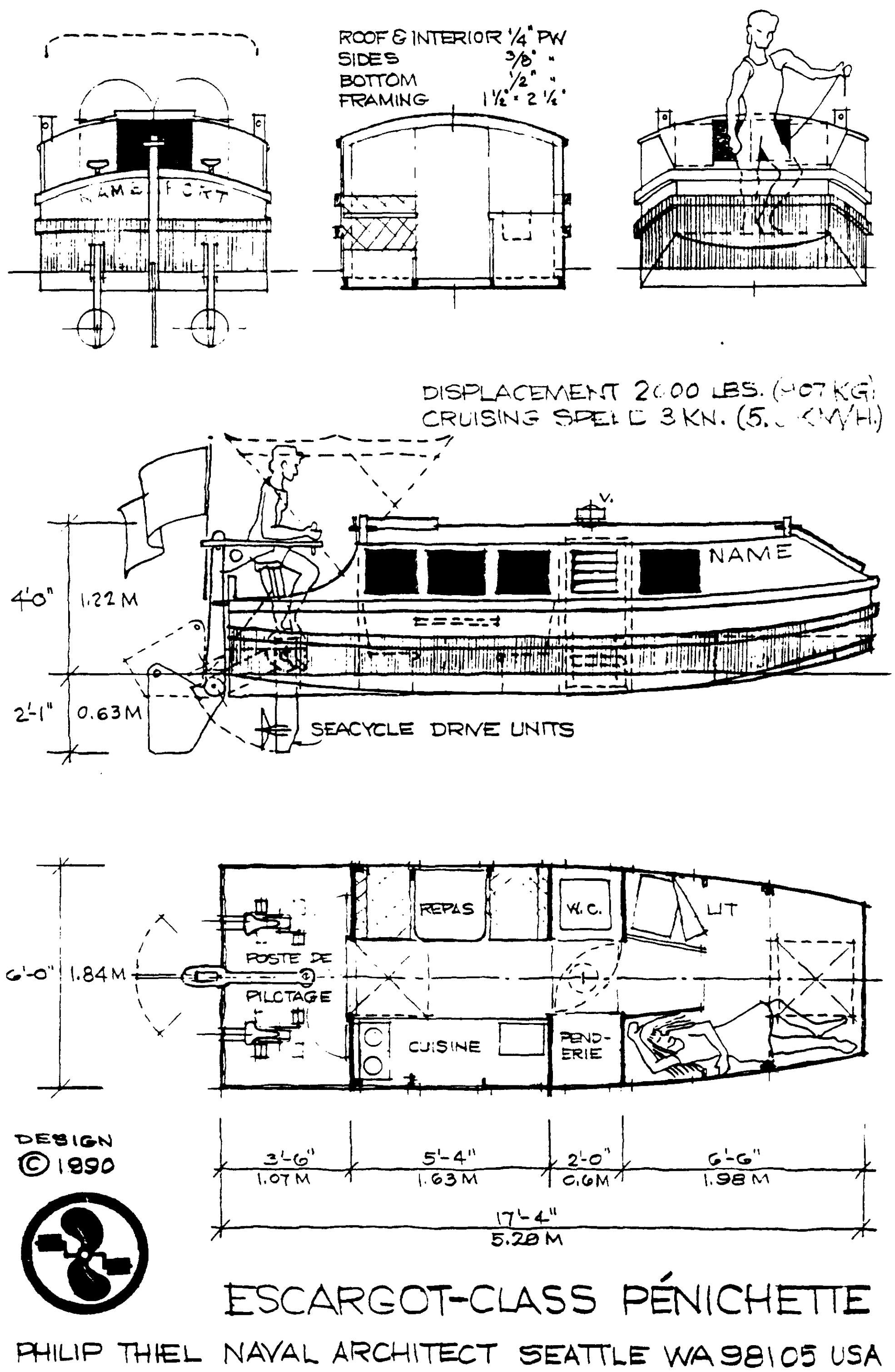

CORRECTION

We made an error in Philip Thiel's "How to make a wooden propeller" article in HP 8/4/91/15. Left column on p. 15, note 2, last two lines: "...the disc area of the propeller, πR^2, where R is the radius of the propeller." Apologies!

Choosing a hull shape for a pedal-powered boat

by Shields Bishop

I have chosen the twin-hulled or catamaran arrangement for most of my experimental boats because it provides the best combination of stability and wetted area. Monohulls can provide the necessary stability only by very wide, flat bottoms or deep, ballasted keels, which in each case results in more wetted area. In order to minimize wetted area for a given displacement, I use a hull shape that has semicircular underwater sections from bow to stern. The radii of these sections are a function of the sine of angles proportioned by the distance from the bow. For example, let's choose a hull with a maximum beam of 500 mm (20 inches) at a point halfway from bow to stern and a waterline length of 5.5 m (18 ft.) At the bow the radius is 250 mm (10 inches) X sin 0° = 0 inches; at 6 inches aft of the bow the radius is 10 X sin (0.5/18 X 180°) = 10 X sin 5° = 0.872 inches; at 12 inches from the bow the radius is 10 X sin (1/18 x 180°) = 10 X sin 10° = 1.736 inches and so on to the stern when the radius is again 0 inches. This example gives a hull which is the same fore and aft. One variation which I have used is to bias the maximum beam toward the stern. To do this, proportion the sine function accordingly. For example, on a hull that is 5.5 m (18 ft.) long, put the maximum beam at 3.7 m (12 ft.) aft. Then from stem to 3.7 m (12 ft.) aft, make the radii = 10 X sin (distance aft/12 X 90°) and from 3.7 m (12 ft.) aft to stern make the radii = 10 X sin (distance from stern/6 X 90°). This relationship gives the hull shape which has the minimum wetted area for a given displacement of any geometrical shape without hollows or reverse curves and at the same time with a smooth transition from the entry and exit to the flanks of the hull. The longitudinal coefficient of this shape is 0.5, which means that we can determine the displacement volume by multiplying the waterline length by the immersed cross-sectional area at the point of maximum beam X 0.5. This is the displacement of one hull at the design depth. A handy feature of this shape is that the wetted area is exactly equal to the maximum beam X the waterline length. There are other useful relationships between wetted area, beam, and displacement which make it easy to consider different sizes and types of craft. The 0.5 longitudinal coefficient is not optimum for speeds in knots in excess of about the square root of the waterline length in feet. For example, with 16 ft. hulls, this would be 4 X 1.15 mph = 4.6 mph. At speeds greater than this, the longitudinal coefficient should be increased to about 0.65. This means that the maximum beam must be decreased and the ends of the hull should be fattened to keep the displacement constant. The narrower hull with the fatter ends will have less resistance than the wider hull with thinner ends at speeds in excess of 4.6 mph. but more resistance at slower speeds. An easy way to change the shape that the sine formula gives is to use exponents on the sine function. If you apply fractional exponents, you will find that the shape gets fuller at the ends. For example, in the above case of an 18-foot boat, suppose that for the point 12 inches from the bow we wrote the formula 10 X (sin (1/18 X 180°))0.8 = 2.465.

So you see that with an exponent less than 1, the effect is to fatten the ends while keeping the maximum beam the same. In this case, we must diminish the maximum beam to compensate. Now we are going to get into Simpson's rule or maybe some computer program. Unless you are going to split hairs for a small increase in speed, I believe it is best to keep it simple.

Second only to speed with minimum effort, the feature people like the most in a human-powered boat is maneuverability: that is, quick, positive steering and quick starting and stopping. As an aid to quick steering, the hull shape described above has lots of "rocker"--deep near the center and shallow toward the ends with no sharp keel or skeg. It helps to keep everything about the boat as light weight as possible without sacrificing durability. Extra weight will make the boat feel sluggish starting and stopping as well as steering.

There are features of this hull design and catamaran arrangement that are disadvantageous. Catamarans, especially light ones without masts, centerboards or keels, are very uncomfortable in beam sea. Unless you are pedalling along at least about 30° to the waves, you will get a sharp, snappy ride. Most boats aren't too good with all the waves abeam, but a light catamaran must be the worst. So when you "tack" in big waves, change heading as quickly as you can and advise anyone new on board to "hold on". It is not unsafe, it is just uncomfortable. You might even get seasick on Golden Pond.

A disadvantage to the 0.5-longitudinal-coefficient shape is that it is "tender" fore and aft. The crew and cargo must be carefully positioned to avoid burying the bow or stern. On the other hand, this shape will not pound in choppy water.

Note

For each single hull of the catamaran, some useful formulas are:

Displacement(lb) $D=.0845(B^2L)$

Beam(inches) $B=3.44\ D/L$

Wetted area(sq.ft.) = $BL/12$

D= displacement in lb

B= beam in inches

L= waterline length in feet

Please pardon the archaic units, but don't they seem to somehow make sense? (Your editor had to give up his efforts to translate them!)

Shields Bishop has worked as a metallurgist for many years, and as an avocation has acquired extensive experience in designing and building a variety of pedal-powered watercraft.

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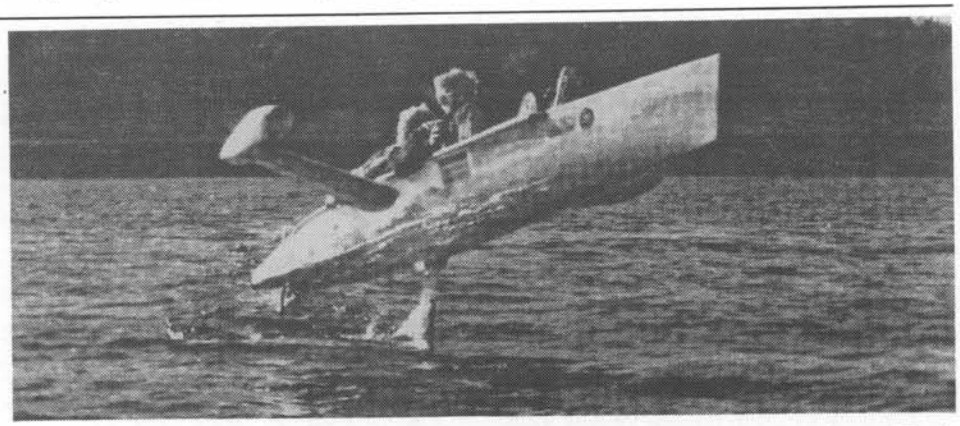

Bjorn Regnstrom in his hydrofoil "Af Chapman II" at almost 6m/s

A propeller design process for human-powered marine vehicles

By Patrick K. Poole

Abstract

A propeller blade design process for human-powered marine vehicles is presented. The process described is based on simple momentum theory modified to account for the energy lost in the rotational motion prevalent in low-speed propellers and typical of those in human-powered propulsion. The process gives the designer a complete mechanism to generate relatively high-efficiency blades specifically suited for their individual application while allowing for a variety of design options.

Introduction

High-efficiency screw-type propellers suitable for human-powered marine applications are of limited availability. This fact became apparent during the design process of the U.S. Naval Academy's entry in the 1st Annual International Human-Powered Submarine Race sponsored by the H.A. Perry Foundation. The event was held at Riviera Beach, Florida in June of 1989. An effort to locate a propeller that was ideally suited for our submarine proved fruitless and necessitated an original design. At the competition it was noted that only a few other entries had specifically designed propellers while many resorted to modifying available off-the-shelf propellers with little regard to efficiency. The Naval Academy's entry, SQUID, incorporated a contra-rotating propeller of in-house design which proved to be extremely efficient. The design method was based on momentum theory, modified to account for the rotational energy lost in the swirling motion of the propeller wake. That process is described in the following paper giving a complete mechanism for developing blade shapes from inputs of engine power, E_p, mechanical efficiency, N_m, and blade hub and tip diameters, D_{hub} and D_{tip}, and propeller RPM.

Discussion

Generally, propeller designers attempt to minimize the propeller's non-thrust flow energy. Two major losses of energy occurring in the conversion of shaft power to thrust power in screw type propellers are 1) the inability of the exit flow to adequately diffuse its energy to the surrounding medium (kinetic energy loss) and, 2) the swirling motion of the flow caused by the rotation of the blade (rotational energy loss). Momentum theory accounts for the kinetic energy loss and designs based on this theory are generally accurate for high-speed applications, but less so otherwise. Using momentum theory for low-speed applications will result in actual efficiencies less that those predicted by the theory. The design process presented herein modifies momentum theory to account for the rotational energy loss making it more applicable in the mid and low-speed ranges. Following the design flow chart, figure 1, and using the presented equations should result in a propeller efficiency as predicted for a specifically identified vehicle operating at design conditions.

Throughout the process there are some general comments that the designer should keep in mind:

1) The larger the propeller diameter the greater the efficiency.
2) Slow-speed propellers have high rotational energy losses
3) High-speed propellers have high kinetic energy losses.
4) A sufficiently large diameter can overcome items 2 & 3.

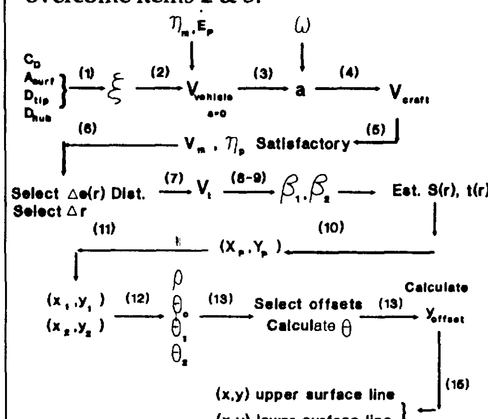

Figure 1. Design process flow diagram

Design process and equations

Following the step-by-step process below will result in the blade shape geometry in the form of (**x**, **y**, **z**) coordinates of the high and low pressure surfaces. The x-coordinate is in the direction of blade motion, the y-coordinate is the axial direction with the positive being forward and negative aft, and z-coordinate is in the radial, or **r**, direction from blade hub to tip. Flow and blade vector components are as shown in figure 2.

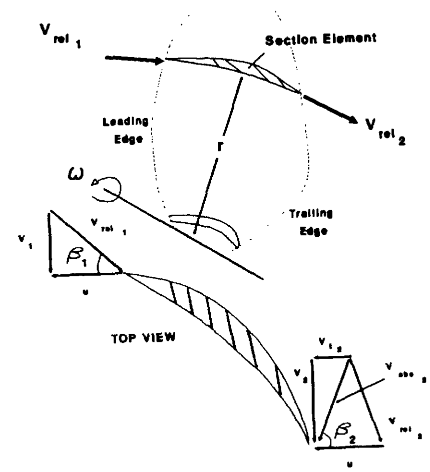

Figure 2. Blade section flow vector definitions

Step 1 Estimate the equivalent drag coefficient, C_D, of your craft. C_D represents the total force required to move the craft (with all appendages in place) through the medium as a function of velocity. It can be estimated by scale model testing and Froude similitude analysis or from semi-empirical data for similar hull forms. Comstock (1967), Allmender (1990), and Saunders (1957) give detailed procedures for both methods.

Step 2 Calculate the total surface area, $A_{surface}$, (again including appendages) of your craft.

Step 3 Calculate the propulsive coefficient, ξ, for the propeller size anticipated.

$$\xi = \frac{V_2}{V_1} = \left\{ \frac{4 C_D A_{surface}}{\pi (D_{tip}^2 - D_{hub}^2)} + 1 \right\}^{\frac{1}{2}}$$

Step 4 Calculate the vehicle speed (assumed to be the inlet axial velocity) when rotational energy is ignored, $V_1 |_{a=0}$, for inputs of your estimated engine power, E_p, fluid mass density, ρ, and mechanical efficiency of the shaft and gearing between the engine and the propeller shaft,

$$V_1 |_{a=0} = \left\{ \frac{16 \eta_m \dot{E}_p g_c}{\rho \pi (D_{tip}^2 - D_{hub}^2)} \quad \frac{1}{\xi^3 + \xi^2 - \xi - 1} \right\}^{\frac{1}{3}}$$

Step 5 Choose propeller design RPM and calculate angular velocity, $\omega = \frac{2\pi}{60}(RPM)$

Step 6 Calculate the rotational energy coefficient, **a**,

$$a \approx \frac{8 g_c \eta_m \dot{E}_p}{\rho \omega (D_{tip} + D_{hub}) C_D A_{surface} (V_1 |_{a=0})^2}$$

Step 7 Calculate the craft speed, V_c,

$$V_c = \left\{ \frac{16 \eta_m \dot{E}_p g_c}{\rho\pi (D_{tip}^2 - D_{hub}^2)} \frac{1}{2(\xi^2 - 1) + (\xi^3 - \xi^2 - \xi + 1)(1 + a^2)} \right\}^{\frac{1}{3}}$$

Step 8 Calculate the propulsive efficiency, η_p

$$\eta_p = \frac{1}{1 + \frac{1+a^2}{2}(\xi - 1)}$$

Step 9 Adjust RPM and blade diameters to optimize η_p and V_c repeating steps 2-7.

Step 10 Choose a blade loading distribution, $\Delta e(r)$, specifying the specific work as a function of radius. The following distributions, shown graphically in figure 3, are all "constant work" distributions that maintain a uniform axial velocity profile in the annulus of the propeller.

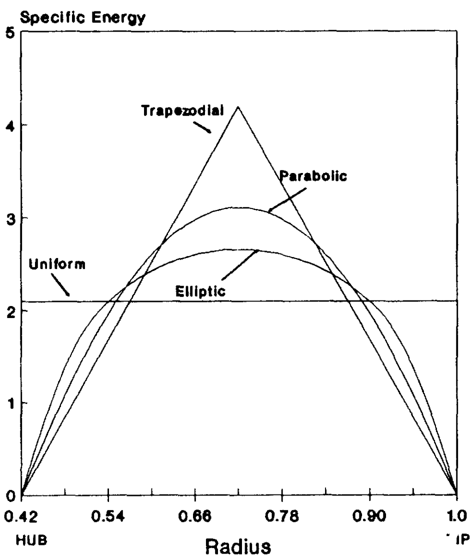

Figure 3. $\Delta e(r)$ Distributions having uniform radial distributions of axial velocity

Uniform

$$\Delta e(r) = \hat{E} = \frac{\eta_m \dot{E}_p}{\frac{\rho\pi}{8g_c}(D_{tip}^2 - D_{hub}^2)(1 + \xi) V_c}$$

Trapezoidal

[Mid ≥ r ≥ Hub] $$\Delta e(r) = \frac{4\hat{E}}{D_{tip} - D_{hub}}(2r - D_{hub})$$

[Tip ≥ r ≥ Mid] $$\Delta e(r) = \frac{-4\hat{E}}{D_{tip} - D_{hub}}(2r - D_{tip})$$

Parabolic

$$\Delta e(r) = 24\hat{E}\left\{ \frac{2(D_{tip} + D_{hub})r - 4r^2 - D_{tip}D_{hub}}{(D_{tip} - D_{hub})^2} \right\}$$

Elliptic

$$\Delta e(r) = \frac{4\hat{E}}{\pi}\left\{ 1 - \left(\frac{4r - D_{tip} - D_{hub}}{D_{tip} - D_{hub}} \right)^2 \right\}^{\frac{1}{2}}$$

Step 11 Choose sufficiently small increment of blade radius subdivision, Δr. Let

$$r = \frac{D_{hub}}{2} + \Delta r.$$

Step 12 Calculate blade exit tangential velocity, $V_{t2}(r)$,

$$V_{t2}(r) = \frac{1}{\omega}\left[\frac{\Delta e(r)}{r} \right]$$

Step 13 Calculate the relative flow velocity angles, β_1 and β_2

$$\beta_1 = \tan^{-1}\left(\frac{V_c}{\omega r} \right)$$

$$\beta_2 = \tan^{-1}\left(\frac{\xi V_c}{\omega r - V_{t2}} \right)$$

Step 14 Choose a maximum chord length and thickness. Choose the radial distribution of chord and thickness from hub to tip, **S(r)** and **t(r)**

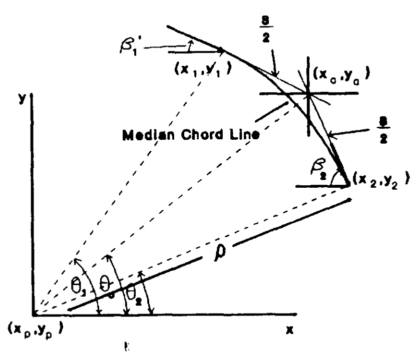

Figure 4 Median chord line

Step 15 Calculate the center of the blade Mean Chord Line, X_p, Y_p), (figure 4)

$$X_p = -\frac{S(r)}{2}\left\{ \frac{\left[\frac{\cos^2\beta'_1}{\sin\beta'_1} + \frac{\cos^2\beta_2}{\sin\beta_2} \right] + [\sin\beta'_1 + \sin\beta_2]}{\frac{\cos\beta'_1}{\sin\beta'_1} - \frac{\cos\beta_2}{\sin\beta_2}} \right\}$$

$$Y_p = \frac{\cos\beta'_1}{\sin\beta'_1}(X_p) + \frac{S(r)}{2}\left(\frac{\cos^2\beta'_1}{\sin\beta'_1} + \sin\beta'_1 \right)$$

where $\beta'_1 = 1.1\beta_1$ allows for approximately 10% incidence angle at inlet.

Step 16 Calculate the leading and trailing edge coordinates, (x_1, y_1) & (x_2, y_2)

$$x_1 = -\frac{S(r)}{2}\cos\beta'_1 \qquad y_1 = +\frac{S(r)}{2}\sin\beta'_1$$

$$x_2 = +\frac{S(r)}{2}\cos\beta_2 \qquad y_2 = -\frac{S(r)}{2}\sin\beta_2$$

Step 17 Calculate the arc radius ρ, and the angles to the leading and trailing edges and the mid-point location relative to the, $\theta_1, \theta_2, \& \theta_o$,

$$\rho = \sqrt{(x_1 - x_p)^2 + (y_1 - y_p)^2}$$

$$\theta_1 = \tan^{-1}\left(\frac{x_1}{y_1} \right) = \frac{\pi}{2} - \beta'_1$$

$$\theta_2 = \tan^{-1}\left(\frac{x_2}{y_2} \right) = \frac{\pi}{2} - \beta_2$$

$$\theta_o = \frac{\pi}{2} - \frac{(\beta'_1 + \beta_2)}{2}$$

Step 18 Select wing section offsets parameters for either NACA sections 16 or 66, Table 1, or any other appropriate section shape.

x/S(r)	Sec 16 y/t(r)	Sec 66 y/t(r)
0.000	0.0000	0.0000
0.0125	0.1077	0.1155
0.025	0.1504	0.1530
0.050	0.2091	0.2095
0.075	0.2527	0.2540
0.100	0.2881	0.2920
0.200	0.3887	0.4002
0.300	0.4514	0.4637
0.400	0.4879	0.4952
0.450		0.5000
0.500	0.5000	0.4962
0.600	0.4862	0.4653
0.700	0.4391	0.4035
0.800	0.3499	0.3110
0.900	0.2098	0.1877
0.950	0.1179	0.1143
1.000	0.0100	0.0333

Table 1. NACA section 16 and 66 offset parameters (from Comstock 1967)

Step 19 Subdivide the angle between the leading and trailing edges such that each angle θ corresponds to the x section offset locations. Calculate θ and its corresponding y_{offset}.

$$\theta = \theta_1 - (\theta_2 - \theta_1)\{x/S(r)\}$$

$$y_{offset} = t(r)\{y/t(r)\}$$

Step 20 Calculate upper and lower blade surface coordinates, (x_i, y_i), figure 5.

$$x_i = \rho\cos\theta - X_p \pm y_{offset}\cos\theta$$

$$y_i = \rho\sin\theta - Y_p \pm y_{offset}\sin\theta$$

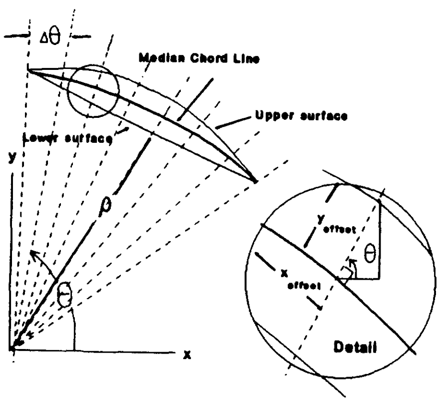

Figure 5. Line generation from NACA offsets

Step 21 Increment θ, repeat steps 18 & 19 until $\theta = \theta_2$.

Step 22 Increment Δ r. repeat steps 9 through 20 until $r = \frac{D_{tip}}{2}$

Contra-Rotating Propellers. Generally, C-R design assumes that the exit flow from the first, or forward, propeller is met by the inlet of the second, or after propeller. The exit from the after propeller is also assumed axial. A major advantage to this propeller is that side, or torsional forces are eliminated. This does not mean that there are no "swirl" or rotational energy losses, in fact, the magnitude of the rotational energy coefficient remains unchanged. A second advantage of C-R is that the amount of specific work done by each blade is reduced. C-R propellers are designed by dividing the specific work between the forward and after blade in whatever manner you feel will give you the best performance. Usually, the specific work is split evenly between the two. Steps 1 through 9 are done as in single propellers so that the inlet axial velocity to the forward propeller (craft speed) and the final exit axial velocity from the after propeller are determined. The axial velocity occurring between the two C-R blades is then taken as the average of the two. The forward blade is design exactly as given above, but the after blade requires using appropriate equations for the relative blade angles.

Step 23 Calculate relative blade angles for the after C-R blade

$$\beta_1 = \tan^{-1}\left(\frac{V_c}{\omega r + V_{t1}}\right)$$

$$\beta_2 = \tan^{-1}\left(\frac{\xi V_c}{\omega r + |V_{t2prop1}|}\right)$$

Results

Figure 7 shows a 3-D graphical drawing of a one-third horsepower, two foot diameter propeller with a six inch hub diameter whose design operation speed is 120 RPM. Parabolic specific work and chord length distributions were chosen with NACA 16 section offsets and a linear blade thickness variation with a half-inch maximum at the hub and one-sixteenth-inch at the tip.

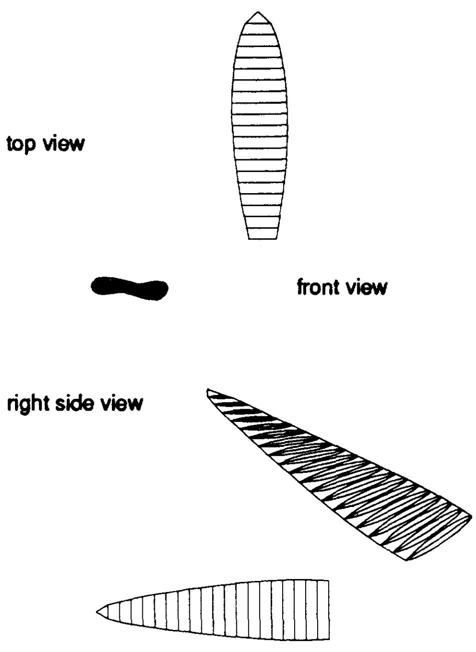

Figure 7. Three-dimensional graphic output

Conclusions

There are many option open to the designer which will result in an optimum (maximum η_p) blade design. The task of

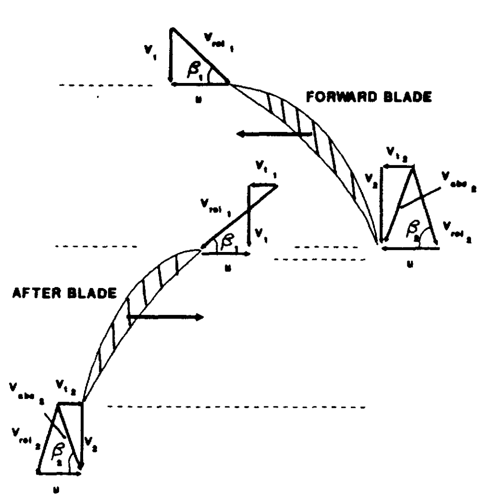

Figure 6. Contra-rotating propellers

the designer is to find the propeller parameters such that losses are minimized and the maximum fraction of engine power is delivered to the fluid in form of thrust. Ideally, a maximum diameter, minimum RPM solution is desired so that non-thrust losses are minimized while reducing the occurrence of cavitation.

Computer Programs

Pat Poole has listings of Theo Shmidt's propeller-design program with some typos corrected (HP 88/7/2/8), and of one modified account for hub diameter. Write him for the listings.

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Pedaling with pedalwheels

(continued from page 1)

Development sequence followed by the author

1. Fore-and-aft bicycle seating and chains driving side wheels on a monohull, or a single wheel on a catamaran configuration with chain drive is OK on short center distances if the frame is rigid.

2. A bicycle chain driving a stern wheel on a monohull 4.3-m (14-ft) long (the "Blue Boat"). Total length of chain was 8m (25 ft). The wood structure was not rigid enough for chain drive. "Blue Boat" made a trip from Syracuse to Schenectady with camping gear for two. It towed two canoes and one pedal boat (with propeller retracted) through a very shallow section of the old Erie canal.

3. Direct drive (pairs of pedals on each end of wheel) using a pair of canoes as a catamaran - a combination of one canoe and one "John Boat" worked well. Three-meter (9-ft) beam required assembly of frame to canoes at launch site. Spray deflectors are needed. Wheel-dip adjustment complicated an otherwise simple system. Decked-over hulls needed dry pockets to accommodate pedaling feet.

4. Rocker drive (oscillating levers with pedals) with wire connecting rods driving stern wheel for 6.1m (20 ft) x 1.2m (4 ft) flat-bottom square-stern "Rose Pedal" (boat was painted rosie pink color). Paddlewheel shaft had two pairs of 180-degree cranks offset 90 degrees. Twice as many cranks are needed since these connecting rods work only in tension. Three passengers sit on center bench, waiting their turn to pedal. The same drive was applied to two canoes, catamaran style. The rockers work, but rotating pedals are here to stay!

5. Direct drive from bicycle seat combined with rocker drive with recumbent seat for side wheels on canoe. Outriggers are needed to obtain stability for high seating. A flat-bottom boat of same overall beam would be more stable.

6. Rotating pedals with wire connecting rods driving a catamaran. Pedallers sat side-by-side. The at-rest wire tension is not critical. The only apparent disadvantage of this arrangement is need for extra cranks on the paddlewheel shaft.

7. Connecting-rod drive for "Rose Pedal". Four wires replaced with four connecting rods (total of lengths =11m (37 ft). Wheel could be raised or lowered by lever at aft seat while underway. Open hulls were a nuisance to keep dry and clear of rubbish and leaves.

8. Direct drive to side wheel by aft pedaller coupled to fore pedaller with connecting rods - this drove a decked, double-ended, flat-bottom, 7.3m (24 ft) x 1m (3ft) hull with "foot pit" for near pedaller. For the necessary stability a monohull has excessive wetted area, wave-making resistance and hull weight compared with a catamaran.

9. Connecting-rod drive for catamaran: seating arrangement is side-by-side. This is the configuration we will study in detail.

Use of bicyle parts

At the beginning of this development sequence, we utilized all parts of a bicycle except the wheels. The latest designs use no bicycle parts, except two wheels! A pair of retractable wheels is built into the craft. This makes cartopping easier than using a trailer. One person can easily handle a catamaran weighing 900-1350 N (200-300 lbf). (See figure 1.)

Hull calculations

Before getting into details of the drive system, let us discuss the equally important hulls. The powering calculations are based on the hull design of Shields Bishop. The choice of a 4.9 m (16 ft) length for our water craft appears arbitrary. However, it is derived from the following calculations. If we want a constant displacement for a series of geometrically similar hulls from 2.4 m (8 ft) to 7.3 m (24 ft) long, we calculate the power required for expected speeds. Results, for a displacement of 2360 N (530 lbf) with each pedal at 110 watts (0.15 hp) input and 60% propulsion efficiency:

Length	Speed 1 pedal	Speed 2 pedals
2.4 m (8 ft)	1.4 m/s (3.2 mph)	1.7 m/s (3.7 mph)
3.0 (10 ft)	1.7 m/s (3.8 mph)	2.1 m/s (4.6 mph)
3.7 (12 ft)	1.9 m/s (4.3 mph)	2.3 m/s (5.2 mph)
4.9 (16 ft)	2.2 m/s (4.8 mph)	2.7 m/s (5.9 mph)
7.3 (24 ft)	2.2 m/s (4.8 mph)	2.7 m/s (6.1 mph)

Molded fiberglass hulls are not essential for a pedal craft. My first catamaran hulls were of 6-mm (1/4") marine plywood with flat bottoms and parallel sides. The second pair I built took me 80 hours after design time, through painting. Having a hull mold does not restrict the molded hulls to a fixed displacement. I increase displacement by increasing hull depth with mold-side extensions. Spread or squeeze in sides of molded hull, before attaching deck, to change displacement.

Steps for determining drive arrangement

1) Choose the number of people to be carried.

A 4.9-m (16-ft) catamaran can accommodate four people with lunch and drinks or two people with camping gear. There is no need to consider pedals for each

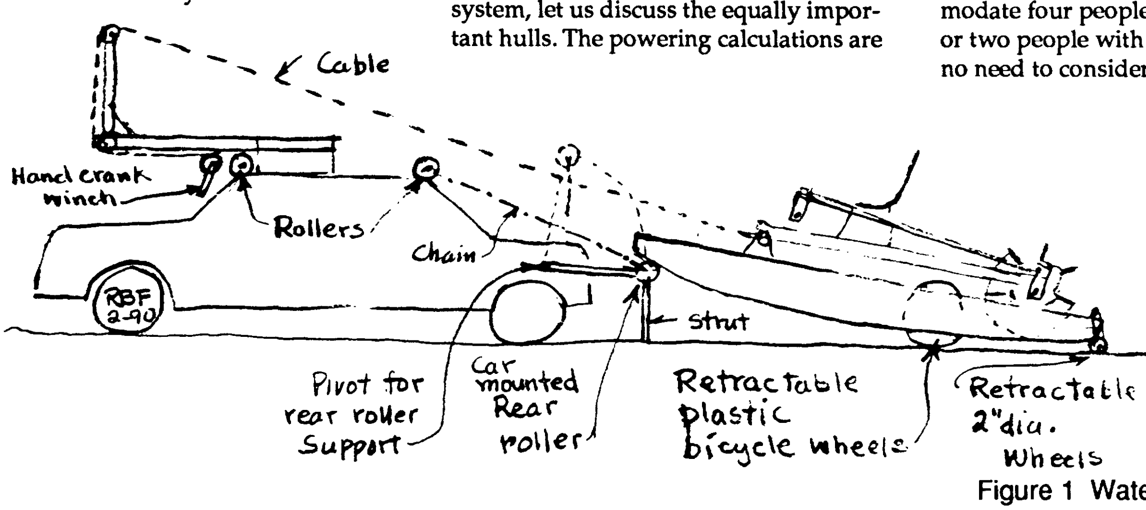


Figure 1 Watercraft delivery system

person. Two people can propel the craft in turns. With each person putting out 110 watts (0.15 hp), adding two more sets of pedals would increase the speed from about 2.2 m/s (5 mph) to 2.7 m/s (6 mph) for double the effort (see figure 2). A single-place craft is not needed. There would be insignificantly better performance over one person on a two-place craft. However, the single-place concept is the ideal platform for experimenting with a feathering paddle wheel. Compare figures 6A and 6B (see page 13). Note that the pedal cranks for both schemes have the same complexity. A two-place craft can be produced for little more effort than for a single-place version.

2) Choose an arrangement for a two-place craft.

The side-by-side is a sociable arrangement and requires a simpler drive (see figures 6B, 6C and 8A). Figure 6C shows a more complex (stiffer) pedal crank, but the fore-and-aft bearings are in a line. With one person or with two of greatly different weights, one hull rides higher in the water. A constant side-to-side paddle-wheel dip is obtained by lowering that side of the paddle relative to the hull. The twisted frame does not affect the pedaling. A tandem seating requires a drive similar to figure 6A, but with two pedal cranks and two more connecting rods. A split wheel, but with connecting rods under the center of the seat, similar to figure 6D, requires a complex wheel-support design (see figure 8).

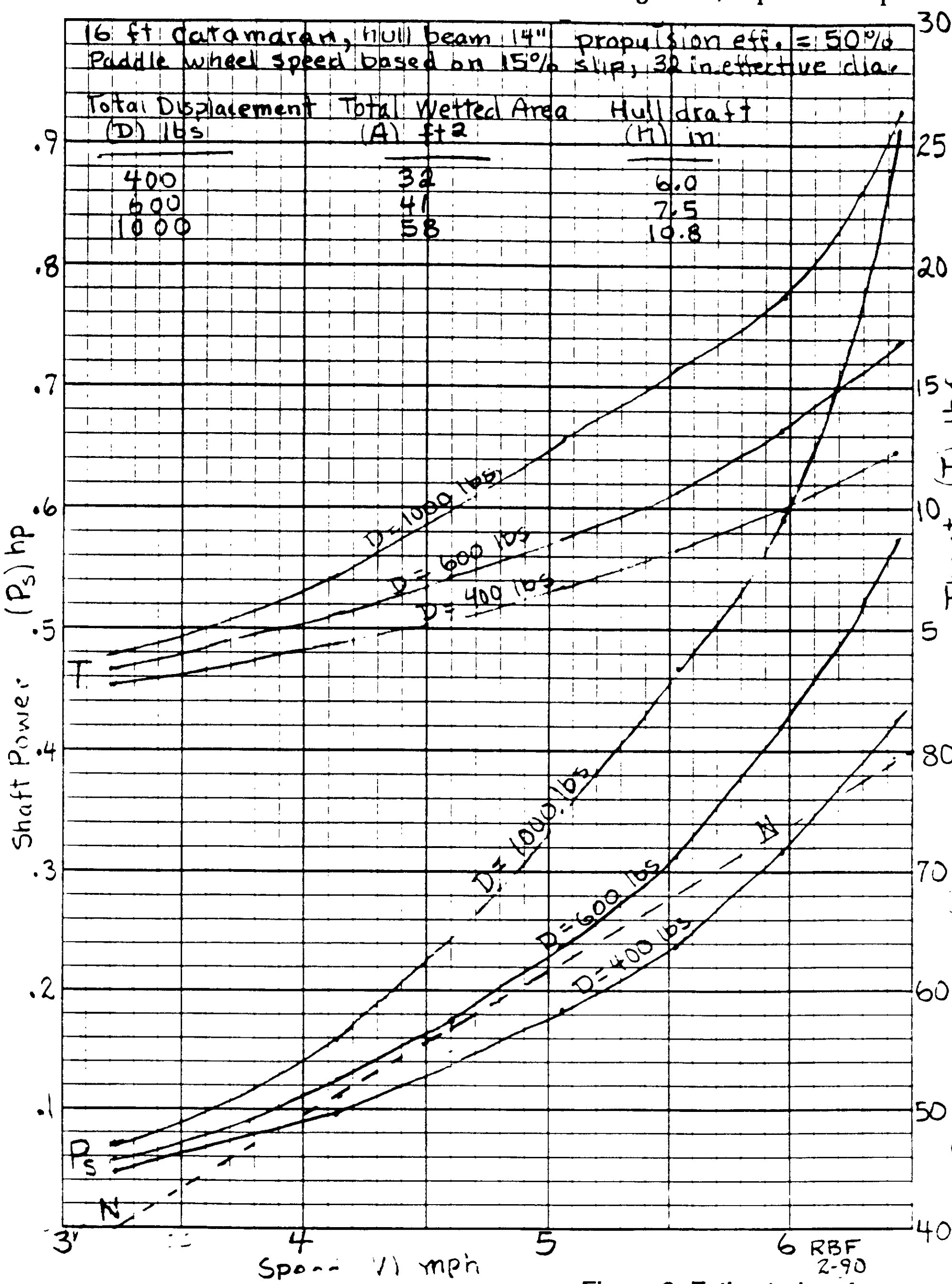


Figure 2 Estimated performance

3) Choose an arrangement for a three-place craft.

A third person can be accommodated on a basic two-place boat by increasing the hull spacing (see figure 6E). The pedal crank is the same as figure 6B but longer between main bearings. There is more space for heavy cargo when the center seat is removed.

4) Choose arrangement for four-place craft.

The seat positions could be similar to figure 6D with pedaling position forward. The drive would be similar to figure 6C, but paddle-wheel length would be greater; or use the center drive of figure 6D. The fore-and-aft minimum spacing for tandem pedallers is 1070 mm (42") with the aft seat 50 mm (2") lower than the forward seat. Without rear pedals, the spacing could be less than 760 mm (30") if the rear-seat position is adjustable to accommodate long-legged forward pedallers.

5) Choose the location of the paddlewheel.

The wheel can be at any fore-or-aft position. Any length of connecting rod can be used. In a more forward position, it is less affected by changes in hull trim. If it is close to the seat, a splash guard may be needed.

Steps for design of paddle-wheels

1) Determine power required.

Consider the powering of three different pedal craft, using the same basic hulls but at different total displacements (see figure 2). The 180-kg (400-lb) craft represents a single-person craft. The 270-kg (600-lbm) and 450-kg (1000-lbm) would be for two and four people, respectively. A catamaran built just for one person would use hulls made from the same mold but with narrower beams. That would reduce the power required. To reconstruct the data shown in figure 1, carry out the following calculations. Calculate the powering requirements due to friction using Froude's equation:

Rf = friction force = Cf.A.V1.825 lbf
A = wetted area - ft² ($1\ m^2 = 10.75\ ft^2$)
V = speed, knots (1 m/s = 1.944 knots)
Cf = friction factor = 0.0106 for 4.88-m (16-ft) length

No simple equation will give the wave-making resistance–see pages 115 and 199 in the same reference. The wave-making resistance can be two to three times the frictional resistance for a 5-m (16-ft) long hull in the 2-3-m/s (4-6-mph) range. But these data are from tests of fatter models having a minimum displacement-length ratio D (tons)/(L(ft)/100)^3 of 50. The Bishop hull was designed at about 30. Estimate the wave-making resistance from figure 3 (until something better shows up). These are extrapolated data. The actual slope of the curves is unknown, but it is certain that the wave-making resistance is zero at zero displacement. The simplification does not take into account other hull proportions: length-to-beam ratio, prismatic coefficient and beam-to-draft ratio.

With regard to other assumptions in figure 1, the literature (Saunders) shows propulsion efficiencies for paddle wheels from 40-55%. The latter is for a radial wheel. The rpm is based on using a wheel 1-m (36") in diameter. This rpm would also be the pedal speed for a 1:1 drive ratio.

2) Determine the paddlewheel blade area.

The swept area of a propulsion device sets the maximum attainable efficiency (see Table 1). Paddle wheels for large ships have a design C_{TL} range could have an actual efficiency over 70%. The reasons for low paddlewheel efficiency will be discussed later. Take C_{TL} = 0.3. If the effective blade width is 75 mm (3"), the blade length then becomes the axial length of the paddle wheel. If more blade area is better, then the catamaran is the ideal application of a paddle wheel. The practical spacing of the hulls and seating provides more than enough paddle-wheel length (see figure 6). The single-place scheme, figure 3A, provides space for over 760-mm (30") long, using a hull center-to-center spacing of 1220 mm (48"). A hull spacing of 1800 mm (70"), used for a craft similar to figure 3D, allows two wheels 660 mm (26") long.

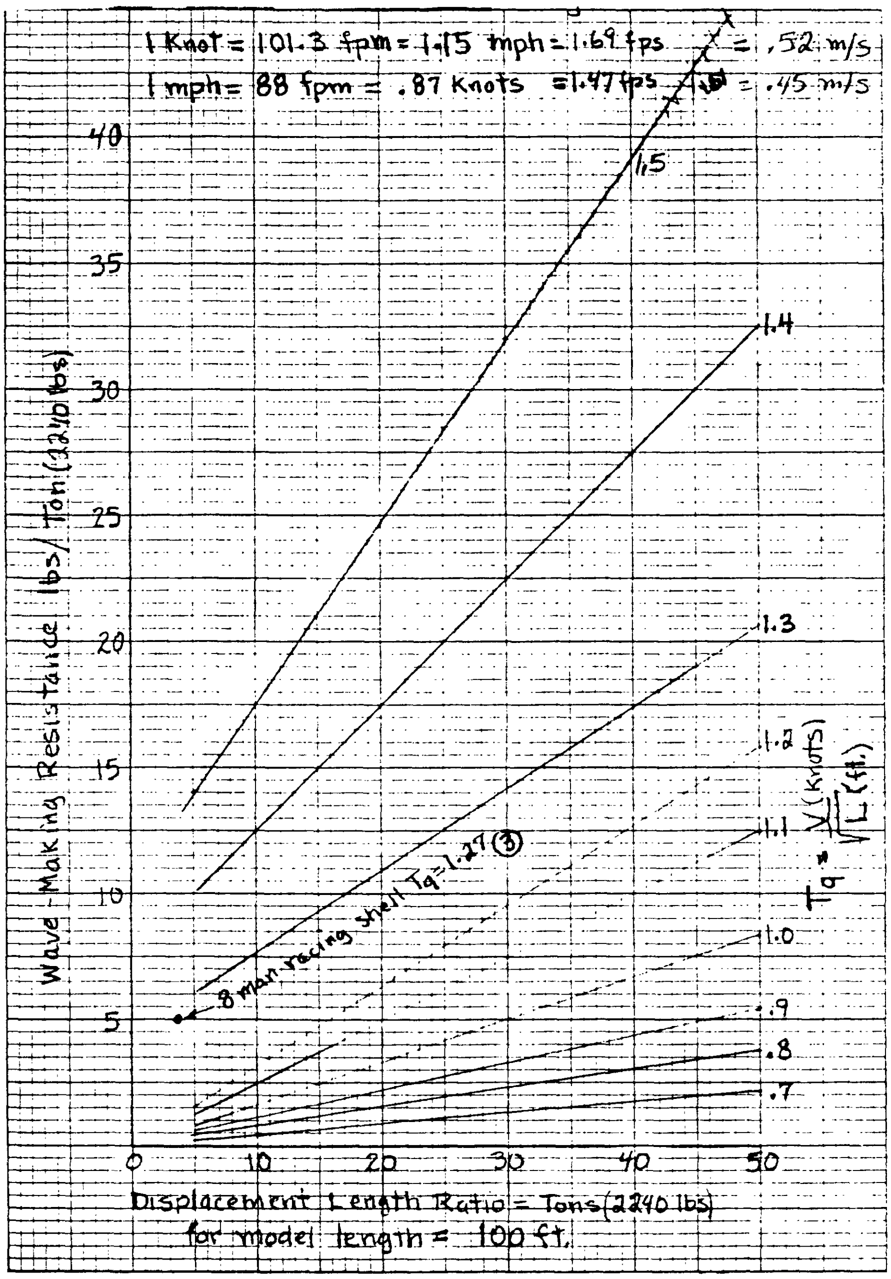


Figure 3 Wave-making resistance

3) Determine paddlewheel geometry.

The angle orientation of the blades entering and leaving the water is a major source of paddlewheel energy loss. It can be reduced by using a large ratio of wheel radius to blade width and a small dip ratio. The use of a feathering paddle wheel does this. The literature refers to a feathering paddle wheel as being equivalent to a radial wheel twice the diameter. But the geometry shows the feathering wheel to be actually much better (see figure 4). The angles for the feathering wheel were measured from a model. The impact angle at the entering side can be reduced by using a curved blade or a lead angle (see figure 5). Reducing the impact angle this way increases the exit losses (throwing water upward). The rationale is that the total loss is somewhat less. A good reason for not using a feathering paddlewheel is that more pedalling effort is healthier. (Eat more peanut butter instead.) Another source of energy loss is in the vortices generated at submerged blade edges. Long narrow blades increase this loss. A conventional paddlewheel design would need a dip ratio of typically 1.5 to assure adequate effective blade area to accommodate varying hull draft. Adjustable blade dip is an essential feature of a human-powered paddlewheel drive. It provides a dip ratio of 1.0, allowing use of stronger wider blades that are immersed only to the depth needed. The upper portion of the blade, being out of the water, suppresses the vortex on that edge. Excessive blade dip is readily noticed by added pedal effort and vibration. The other geometric feature is the number of blades. Increased blade pitch ratio as defined in figure 5 (page 13) improves efficiency. It reduces the effect of an excessive amount of blade area in the water at once and the interaction between closely-spaced blades.

With a dip ratio of 1.0 and 10 blades, the swept area varies from 100% to about 70%, ten times per revolution (see figure 4). The effect of this is not felt at the pedals.

6) Make bearings, connecting rods and pedals.

Teflon and nylon and other plastic bearings are used (see figure 11). Consider well-oiled wood bearings. The

Table 1 Blade size

Use thrust load coefficient (C_{TL}) method

$$C_{TL} = \frac{F}{\frac{P}{2} A V_a^2} \text{ (dimensionless)}$$

$$A(in^2) = \frac{69T \text{ (lb)}}{C_{TL}\ V^2 \text{ (mph}^2)}$$

F = propulsion force, thrust (lb)
A = sweep area of propulsion device (ft^2)
Va = velocity of advance (assume = hull speed) (fps)
P = density of water = P $\left(\frac{62.4}{g} \frac{lb\ sec^2}{ft^4} \text{ for fresh water}\right)$

$$\text{Ideal efficiency (i)} = \frac{2}{1+\sqrt{C_{TL}+1}}$$

C_{TL}	i
0.3	0.93
0.5	0.90
1.0	0.83

Examples

Displace. lb.	No. of persons pedal	Power =50% hp)	Speed (mph)	Thrust (lb)	Blade Area C_{TL}=.3 (in^2)	Blade Length (width=3") (in)	Equivalent Propellor Dia. (in)
400	1	.15	4.7	6	62		
400	1	.25	5.6	9	66	23	9.2
600	1	.15	4.4	7	83		
600	2	.30	5.5	11	84		
600	2	.50	6.2	15	90	30	10.7
1000	2	.30	4.9	12	115		
1000	2	.50	5.7	17	120		
1000	4	.60	6.0	19	121		
1000	4	.90	64.	23	129	3	13.4

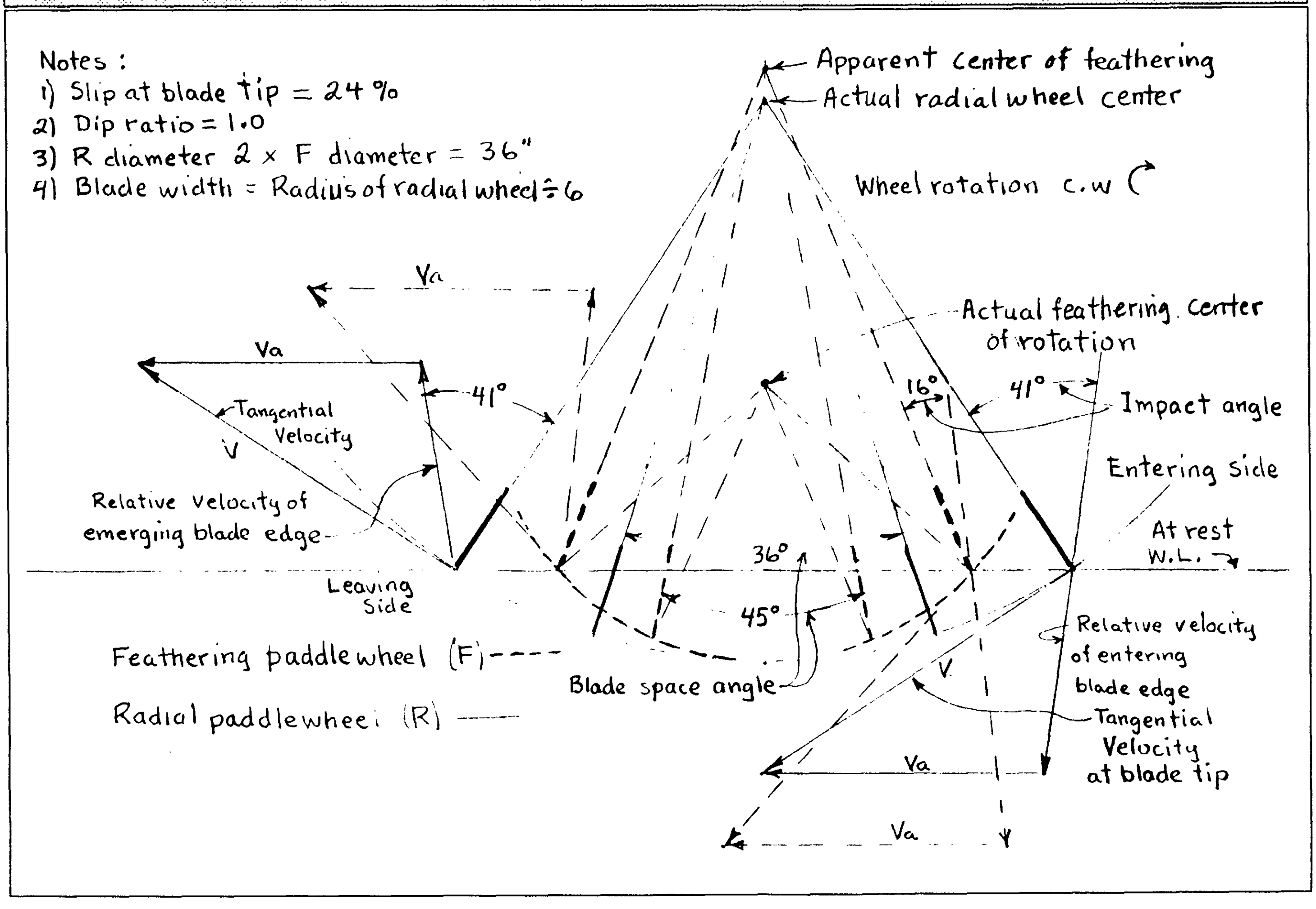


Figure 4 Critical paddlewheel blade angles

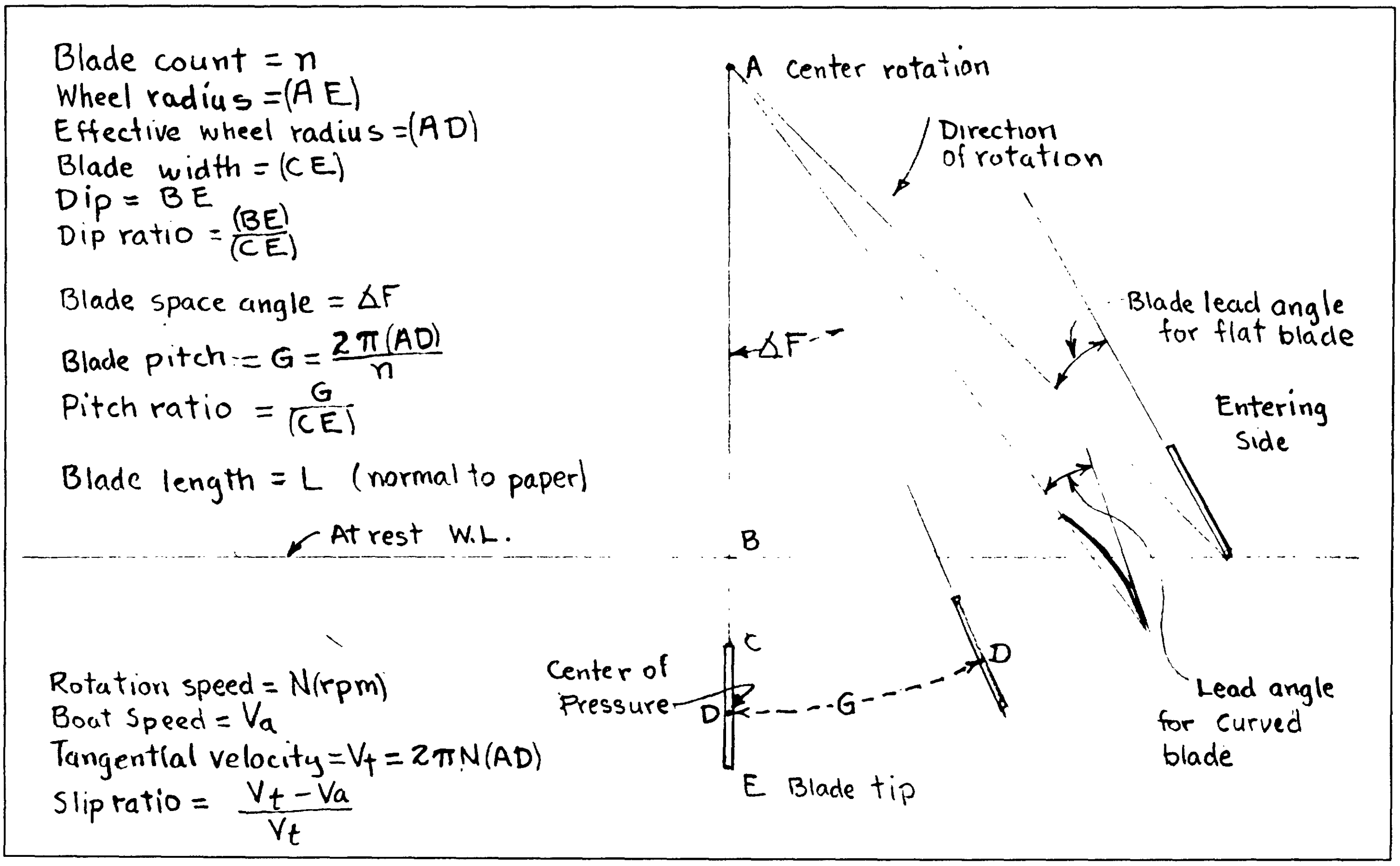
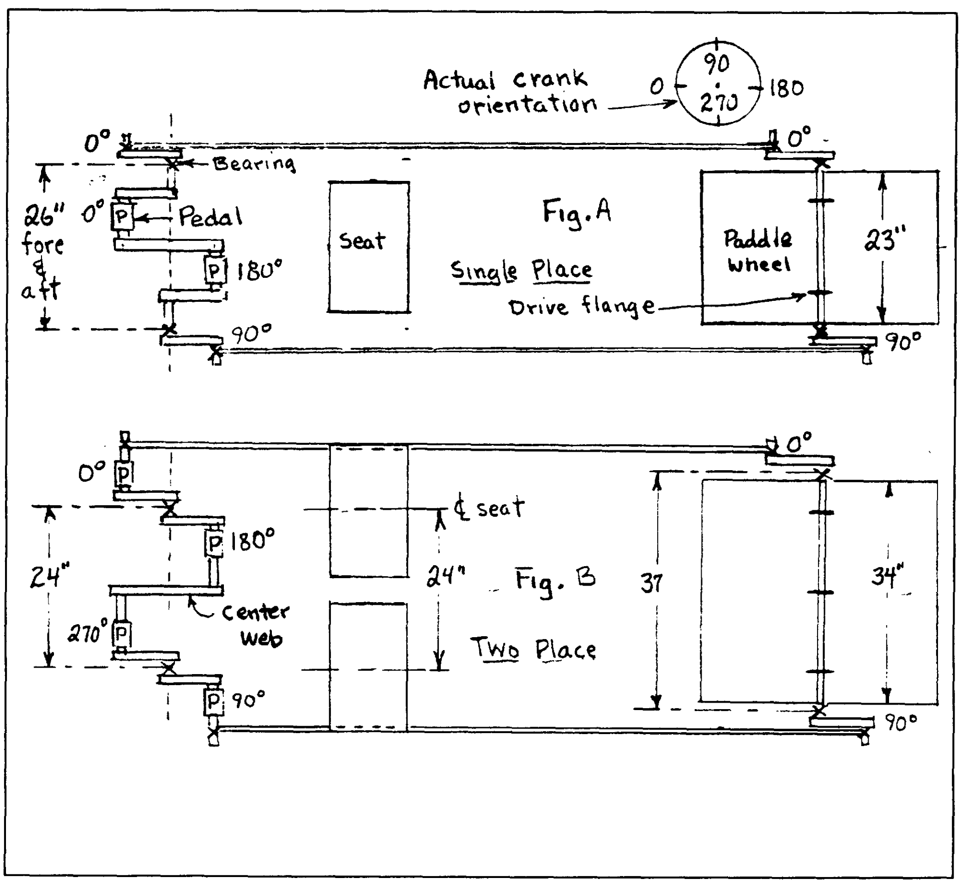


Figure 5 Definitions for radial paddlewheel



Figures 6a and 6b Crank and connecting rod schemes

pedals could be made from two pieces of plywood, glued or screwed together. The connecting rods are 20-mm (3/4") wide by 40-mm (1-1/2") to 50-mm (2") deep, stiffened with 10-mm (3/8") thick, 50-mm (2") wide flanges, top and bottom to make an "I" beam section.

7) Put it all together.

Use aluminum or steel tubing (EMT), and wood for structure and seats. Hold it all together with sheet-metal brackets fastened with hose clamps, screws, and wire. The molded hull has a horizontal flange at the deck edge for bolting the support structure to the hulls. All mounting brackets and the hull flanges have 75-mm (3") modular hole spacing. The fore-and-aft position of the frame can be adjusted without major rework.

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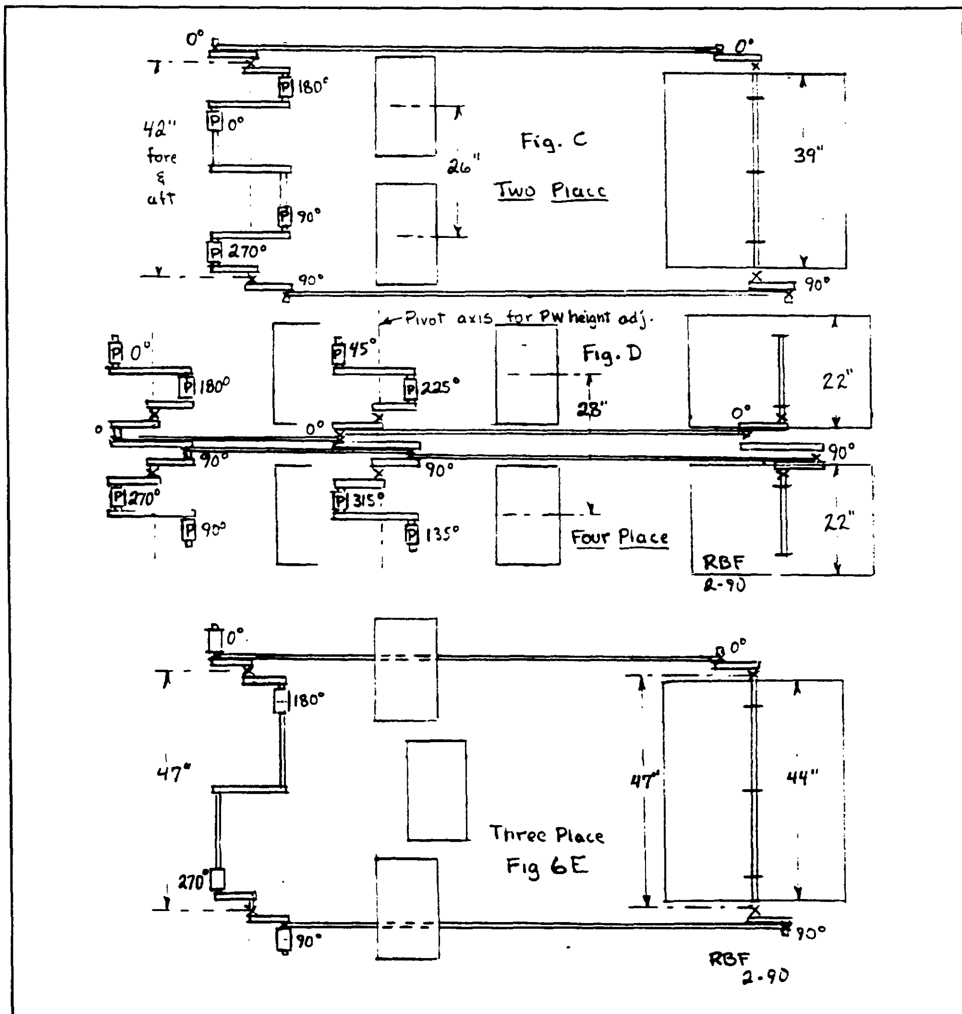
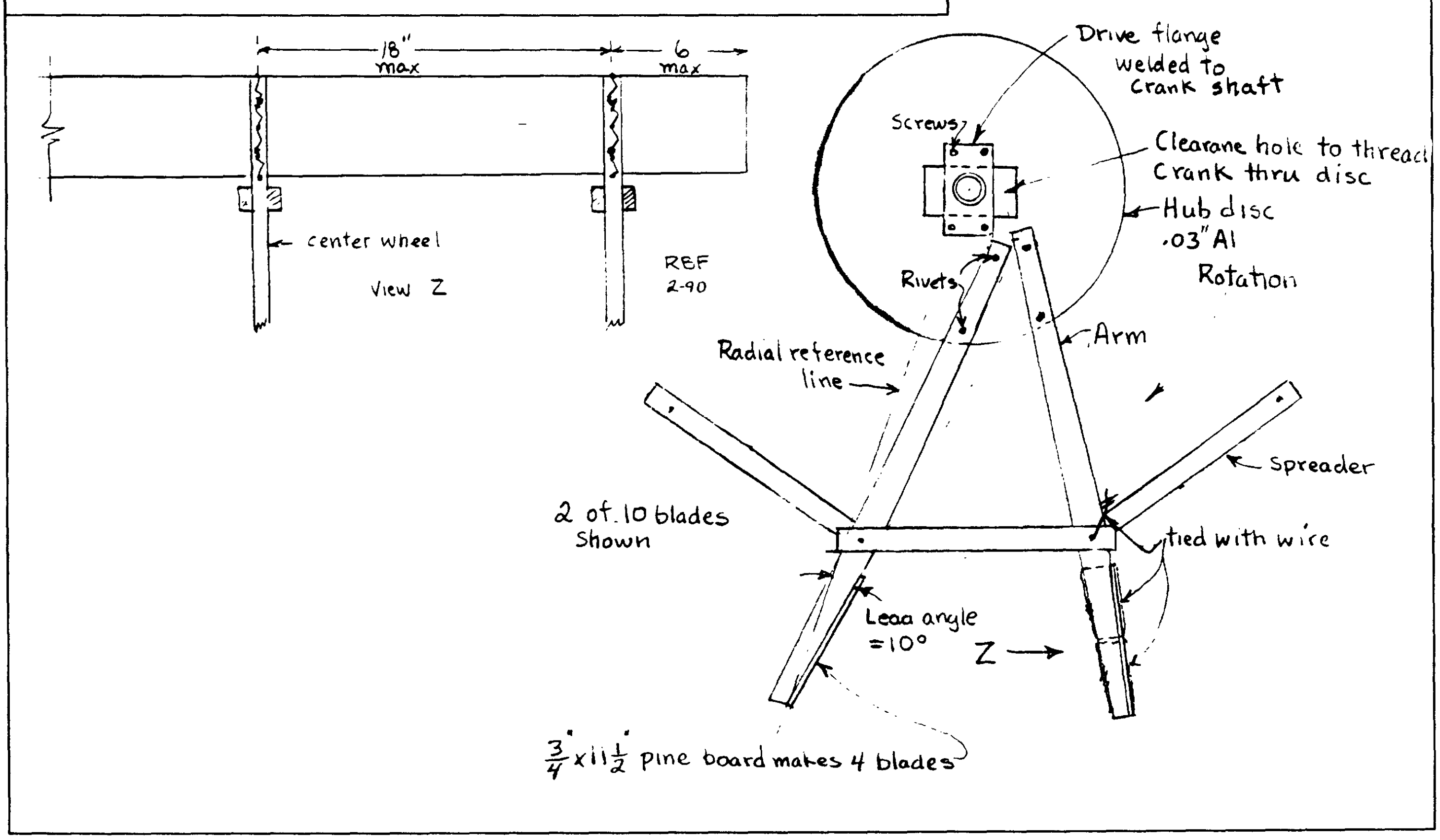


Figure 6c, 6d and 6e (right) Crank and connecting rod schemes

Figure 7 (below) Paddlewheel construction



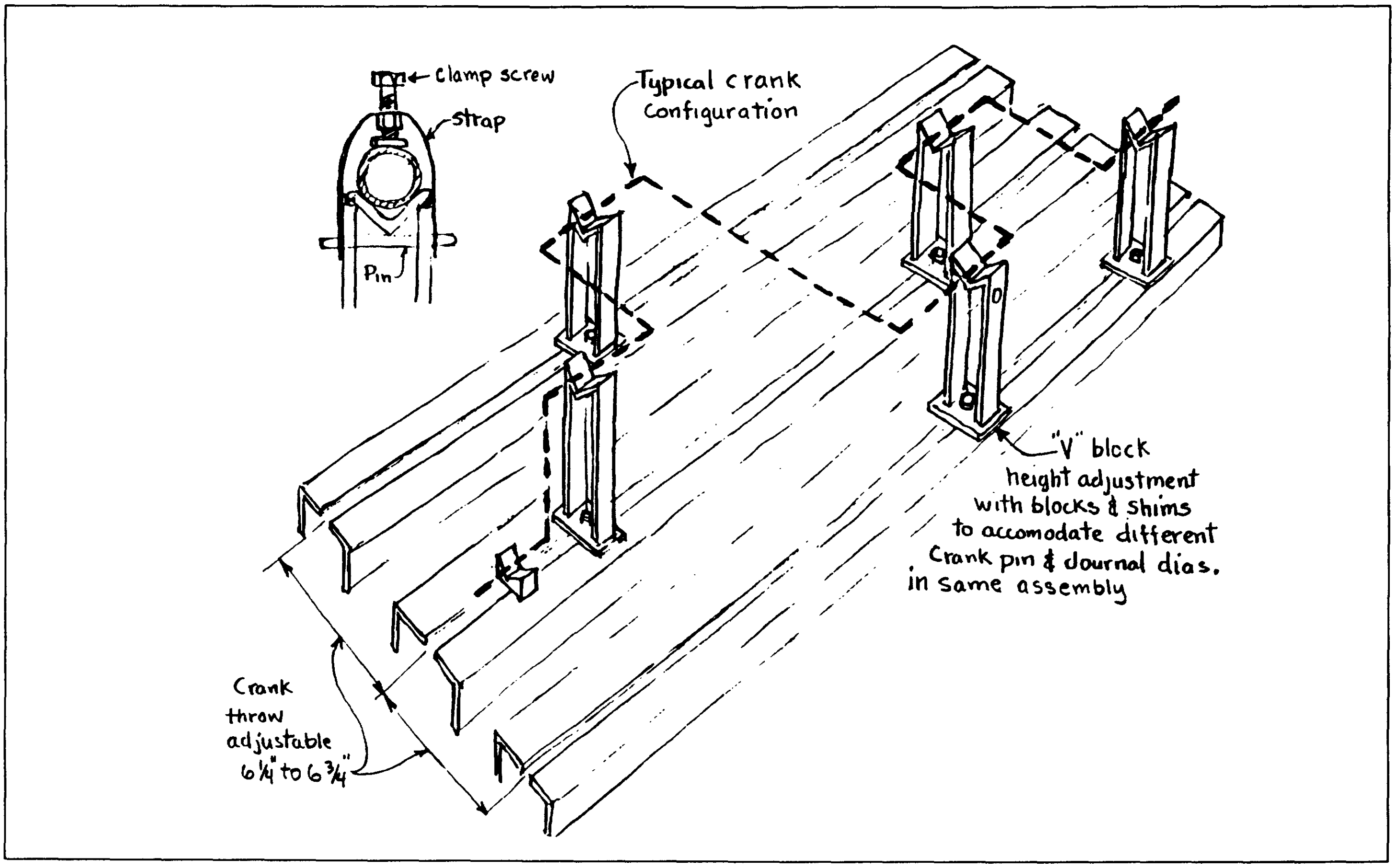


Figure 8 Crank alignment fixture for welding and checking

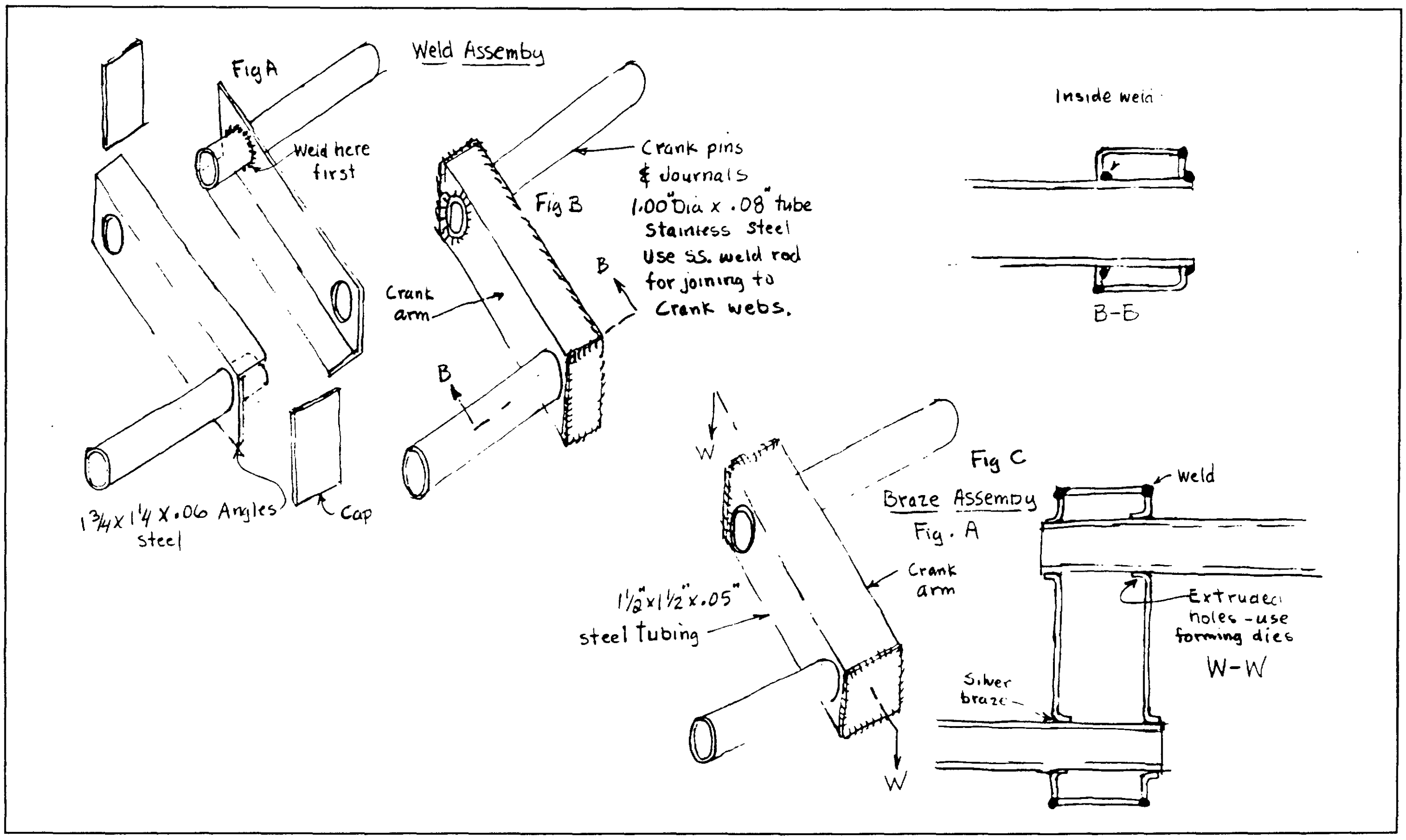


Figure 9 Crank joints

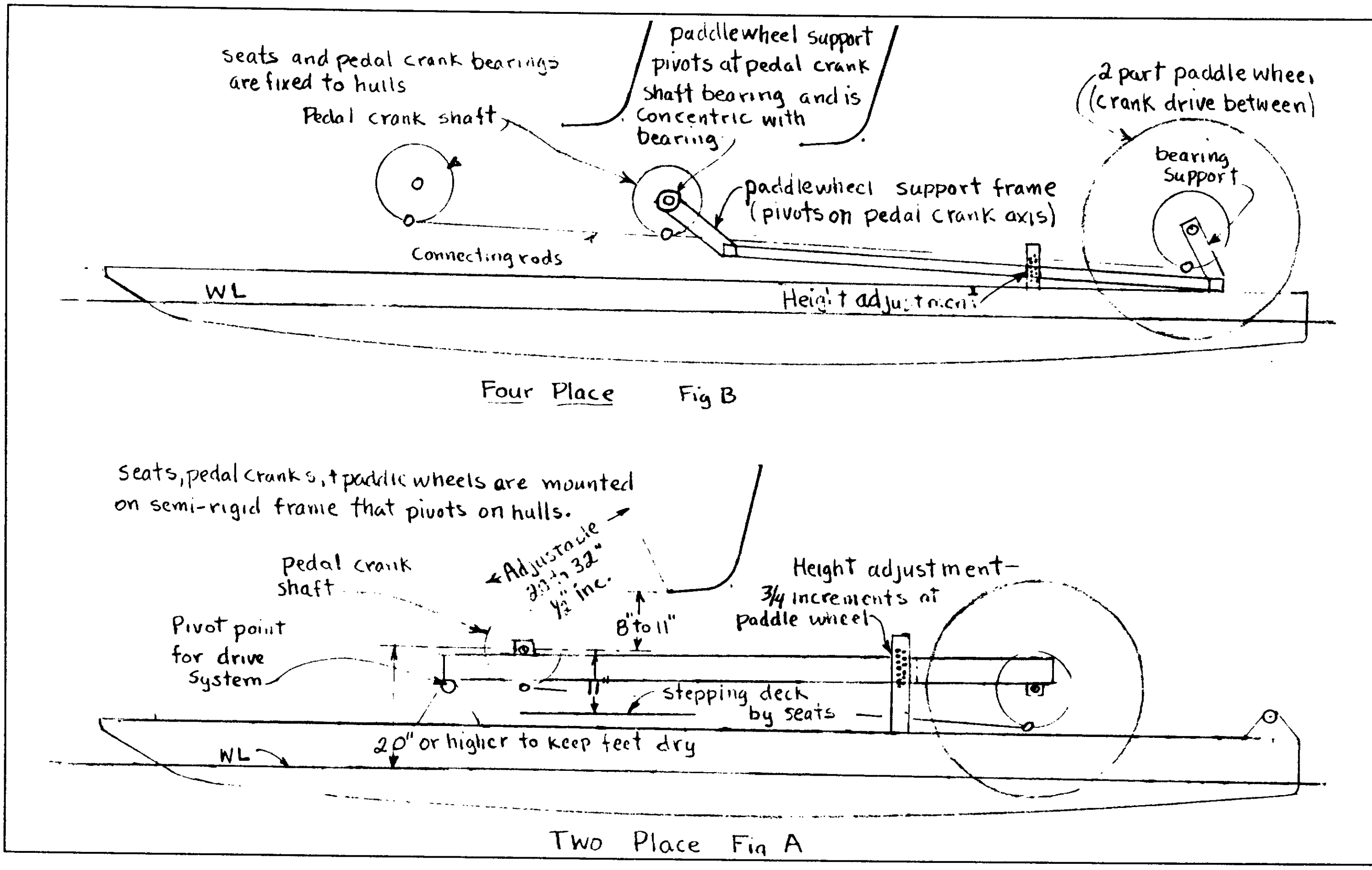


Figure 10 Drive support structure

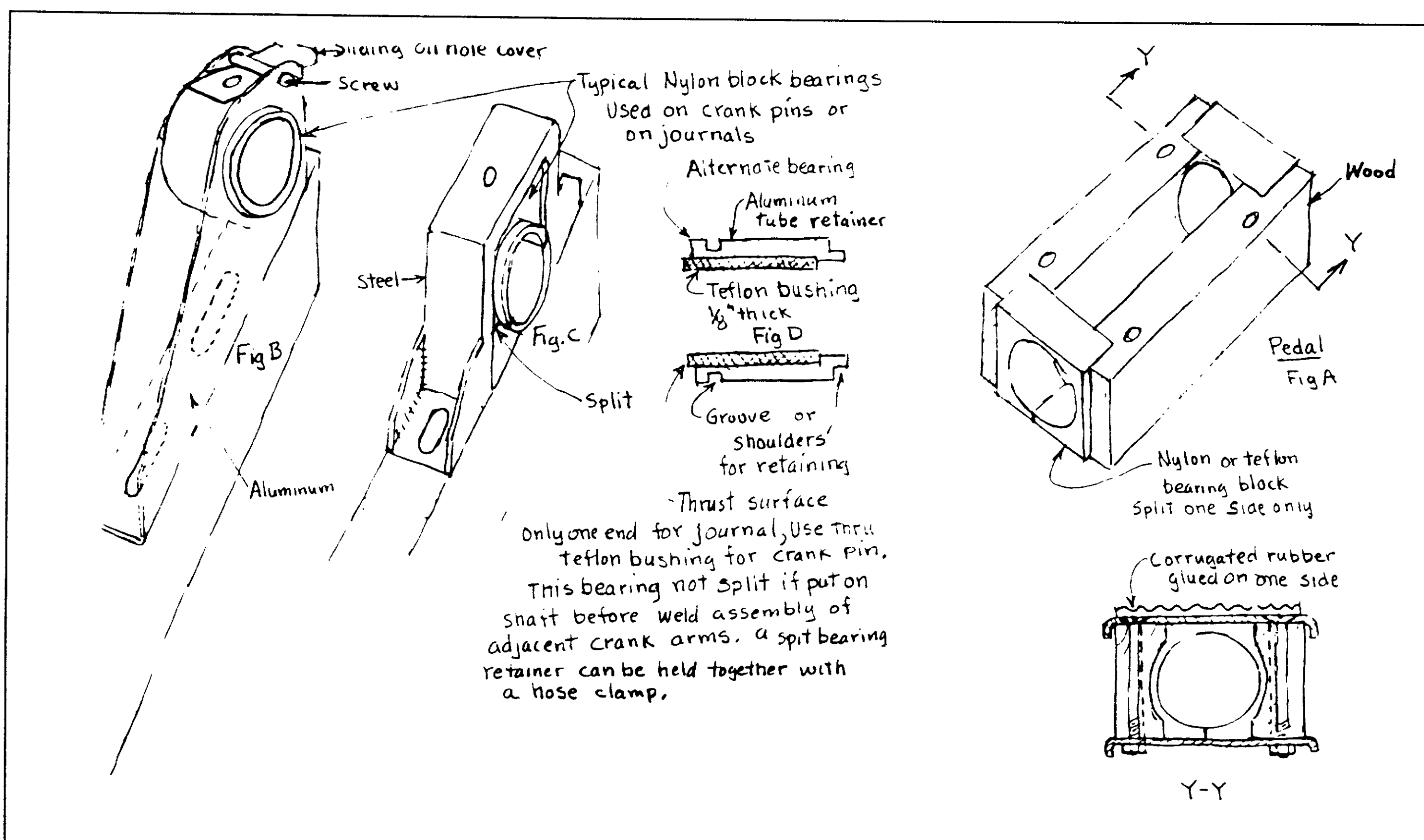


Figure 11 Bearings and pedal

Flat-tire directional performance
by Doug Milliken

Some years ago, I spent a very interesting day test riding a large (300 kg) fast motorcycle on one type of run-flat tire. The work was for a large tire manufacturer who, unfortunately, has stopped making M/C tires. The test consisted of riding with the tubeless-tire valve removed from the rear tire (zero inflation pressure). A tire of suitable size for the front wheel was not available for this test.

This particular tire had what could be described as a bead- retention-system consisting of a flap of rubber outside the rim as well as the usual wire bead on the locating-bead seat inside the rim. Two designs were tested with different dimensions of external flap.

a) The tire with the thinner flap allowed the bead wire to slip off of the bead seat at low speed, into the drop center. After this failure of the bead-retention system, it performed much as any standard tire–the tire flopped back and forth across the rim inducing a great deal of undesirable "rear steer". The M/C was just barely controllable at low speed. The symptom was unpredictable crab-tracking (dog-tracking) randomly alternating left and right. This required large (~ 10 degree) front-steering-angle compensations and large rider body motions.

b) The other tire design with a thicker flap dimension was remarkable. With no pressure it could be ridden safely at 80 kph with only a slight 'twitchy' feeling from the rear end of the M/C. At the prompting of the test engineer, I attempted to dislodge the tire bead by riding in small circles while lifting the front wheel off the ground with engine power (popping wheelies in first gear). Even with this abuse, the tire bead stayed firmly seated. This design of bead-retention system really worked and was an undeniable safety feature.

Unfortunately, this particular run-flat tire design is heavy and requires thick sidewalls. I suspect that it is not suitable for bicycles due to high rolling resistance as well as high weight. Additionally, it was several times more difficult to mount than a standard tire because the flap had to be properly seated over the lip of the cast aluminum rim.

My guess is that run-flat bicycle tires will probably be tubeless, something that is bound to come as HPVs slowly catch up with motor-vehicle technology. First, we have to give up these leaky spoked rims...Certainly no tube would have survived the extended running that I did with the rim squashing the tire flat. In fact, I suspect that there was some proprietary "lubricant" inside the tire to reduce the rubbing friction (and tire destroying heat) when the M/C tire was pinched by the rim. This test convinced me that run-flat tires are possible and that bead retention is one key to retaining control with a flat tire.

Until some tire and rim manufacturer produces "the right thing", I take the advice of Dr. Alex Moulton which is to use tube protectors (Wolber Protex, Mr. Tuffy, etc.) and also to choose a tire with a good bit of air in it. Narrow racing tires don't have much air in them so they go flat very quickly, even with a small puncture, leaving little time to slow down. Of course, this doesn't help in the case of a true "blowout" but, in my experience, instantaneous "blow-outs" are rare with tough, modern, tire materials.

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❑

Bicycle fairings and efficiency
Comment by Doug Milliken *on* Dave Kehoe's *article*

It is good to see research on conventional bicycles with fairings such as Dave Kehoe's "Bicycle fairings and efficiency". We did similar (but less extensive) coast-down testing and also wind-tunnel testing on front fairings over ten years ago, part of which has been published in the "Second International HPV Scientific Symposium" proceedings (available from IHPVA). These comments are meant to expand on testing techniques, not to question his results which all look very good to me.

For coast-down tests, we also used a digital speedometer to measure top speed. Back in the dark ages, this was a DC tachometer generator belt-driven off the rear hub, monitored on a digital voltmeter. It actually calibrated fairly well and was good to about 0.1 mph!

Mr. Kehoe has pointed out the basic problems of coast-down tests nicely–in particular waiting for calm air, including the effects of nearby traffic. Calm air is necessary to get any kind of repeatable results. As air drag is nonlinear with speed, simple averaging of run speed is not valid, unless the runs are all at nearly the same absolute speed.

Unfortunately, there is almost always some wind, which implies a "relative wind" not lined up with the direction of vehicle motion. In our work, sponsored by Dr. Alex Moulton, we concluded that real-world fairing performance must be judged by some kind of wind-averaged performance. Better still are graphs showing performance at different relative wind angles and at different wind and bike speeds.

Here are two ways that we have used to get at this problem:

a) Side-by-side coast-down testing ("racing") is the low-cost method. Two riders of equal stature, weight, equal tires and tire pressure, etc. (but with different aerodynamic configurations) start out at the top of a hill and match speeds before they both start coasting. This way both riders are in about the same cross-wind conditions. They should stay on opposite sides of the (empty) highway to avoid any drafting or wind-shadowing effects.

The last time we tried this, we used a long hill (an extension of the fault line that forms Niagara Falls) that was shielded from a strong cross wind by dense trees on the top half, but was wide open to a gusty wind for the bottom half. The two fairings under test were well matched in the sheltered part of the run but one had markedly better performance in the presence of a cross wind. The latter also was much harder to control in the cross wind...

Of course, these results are only relative; it is hard to imagine any way to characterize each run and to get solid numerical data. Perhaps some future experimenters will have the capability of measuring wind speed and relative direction as well as bike speed. They will also need good data recording and analysis to make any sense at all out of these transient data.

b) The expensive way is to test in a wind tunnel at various steady-state yaw angles (relative wind angles). The tests we ran measured the "x-force", the force along the longitudinal axis of the bike that the rider must overcome by pedaling. We also measured the lateral force ("y-force") at the front and rear wheel to get a feel for the control difficulty associated with different fairings. The raw data were corrected to a standard velocity and density condition and plotted as "x-force" vs. yaw angle.

From the basic data it is possible to make plots of "x-force" drag for combinations of different bike speeds, wind speeds and wind directions. We cross plotted the raw data on polar paper and the plot we came up with is somewhat similar to the "velocity-made-good" plot used to characterize sailboats with different sails, in different wind conditions.

The wind tunnel is a powerful tool. Until some future date when we can afford to truly calculate entire flow fields by super-super-computer, the wind tunnel is still the only way to really measure the stalled-flow aerodynamics of bluff bodies, such as partially streamlined bicycles. There is also a lot to be said for the experience of sitting on a bike in a wind tunnel in the absence of other distractions. In particular, I found that I became very sensitive to any little bit of flapping clothing–by tightening various muscles or shifting slightly I could stop the flapping and reduce my drag (note that our tests were just at the dawn of skin suits).

Incidentally, the upright, fully faired, Alex Moulton 51-mph (23-m/s) run mentioned at the beginning of the article was made by Jim Glover in a fairing designed and built by me, not ZZip Designs. The confusion is easy to understand because ZZip make a special model road fairing to fit the AM Bicycle. The 200-meter run in question was at the 12th IHPSC in Vancouver, Canada, at sea level, and on near-level ground. In all fairness, this should not be directly compared to runs made at high altitude, on a course with IHPVA-legal down-hill slope.

The bike inside the "Moulton-Milliken 'Liner II" was a predecessor of the AM "Jubilee" model, now in production in England. With continuing support from Alex Moulton Bicycles Ltd., the current-generation "M-M 'Liner III" has shown good results, especially in Hull 1989 (Jim Glover riding) and Milwaukee 1990 (with Fred Markham up). Timed runs to date have been plagued by poor wind conditions. 'Liner III is based on the new AM "Speed" model: look for us at the IHPSC this summer in Milwaukee!

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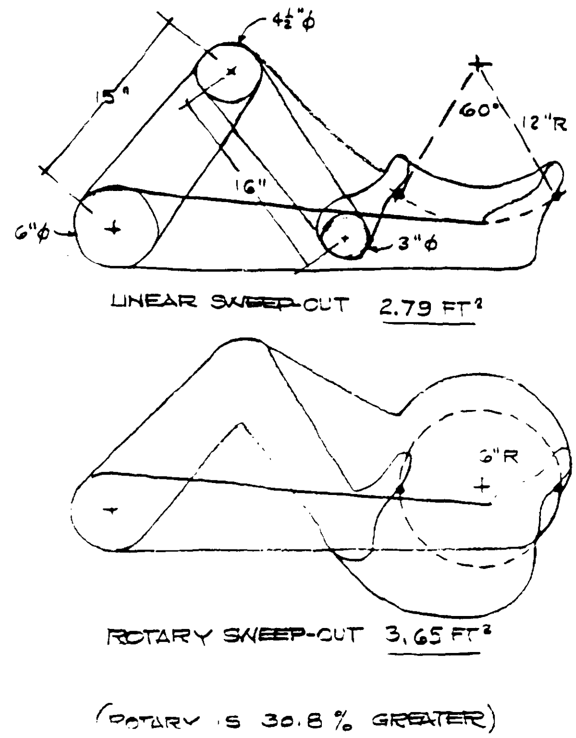
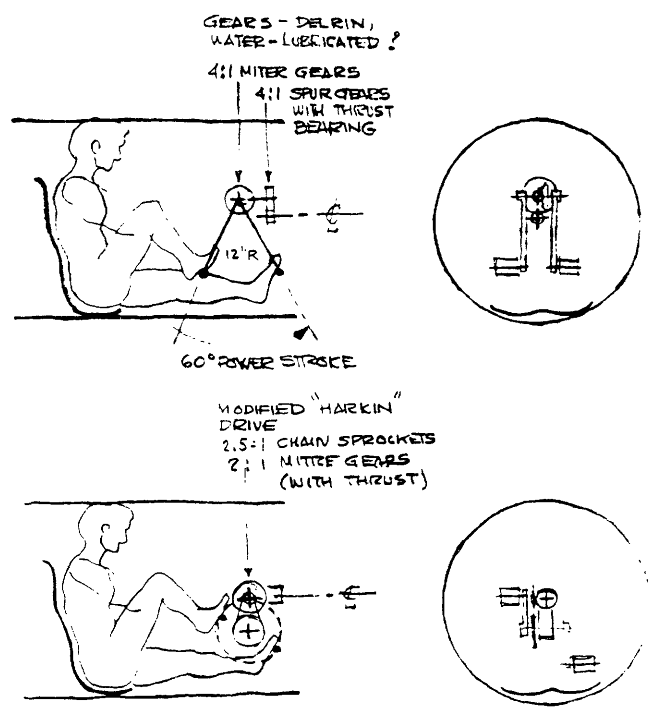
Leg sweepout volume, drag—and cooling?

Philip Thiel sent some sketches he made about HP-submarine transmissions, in which pedalling in water makes reduction of leg friction very advantageous. He showed that rotary pedalling involves the legs sweeping out 30-percent more volume than does oscillating or linear pedalling. The implication—which needs to be tested—is that linear pedalling would incur less water friction. Therefore in air in cold weather it could be that one's feet would not get so cold in linear vs rotary pedalling. Here's another interesting research topic.



linear drive

rotary drive



Paddlin' Madeline

by Philip Thiel

"She was named Madeline and came about her name quite naturally from the old song 'Paddlin' Madeline Home'. This is what her owner shot back at me when I asked what he was gong to call her."

So writes Harold "Dynamite" Payson about the 19', 6" (5.9 m) plywood side-wheel paddle boat he built for Peter Burling from plans by Philip Bolger(1). Seating two people side-by-side on seats that adjust fore-and-aft, the boat is 7', 10" (2.4 m) wide over the paddle-boxes, draws only a few inches, and is adapted for operation in calm, smooth waters. Propulsion by paddle wheels is, of course, the way to go in shallow water, with the further advantages, compared with propellers, of fewer problems with weeds, underwater damage, and mechanical complexity. Disadvantages are weight, bulk, and windage of the paddle wheels.

About performance, Dynamite writes: "I must admit that all during the building of Madeline I was skeptical, thinking she was going to be awkward to pedal and very tiring and that the moment you stopped pedaling, unlike a bicycle, she would simply come to a halt. Phil hadn't been very reassuring, saying that he offered no guarantee that this boat would work...But, I decided that this one, too, deserved a try.

"I'm glad, for the first time we put her in the water one July evening her prospective owner and I pedaled away any negative feelings I'd had about the craft. Pedaling her was easy, not the least bit awkward but a nice, pleasant exercise you could keep up for quite some time; Phil had gotten the amount of paddle dip and foot-pedal leverage exactly right. I liked the chuffing sound of the paddle wheels, a rhythmic accompaniment that reminded me of an old steam train getting up to speed. The speed surprised me, too. Oh, she was no race horse, and she definitely preferred quiet water, but the two of us got her up to a steady seven knots with no strain" About that, designer Bolger comments: "I don't believe it for a minute: possibly five. Anyway, [there were] no mile trials that I know of." But owner Burling reports that he can do a 1.5 mile (2.4 km) distance on Lake Sunapee (New Hampshire) in Madeline faster than he can scull.

He goes on to say she is a "wonderful lake boat", with the best part being her substantial feeling, and the delightful, rhythmic accompaniment of the paddles. He does report, however, that in a chop a paddle will dig in and cause a roll, and that in a wind the bow will swing about the deep dagger rudder (fitted in place of the steering cars shown in the drawings and photograph). But owner Burling is obviously well pleased with her performance and states she is vastly superior to those he has tried in the Caribbean and Europe.

Paddlin' Madeline

Notes

1. Dynamite describes and illustrates the details of construction in his book Build the New Instant Boats (Camden, ME: International Marine, 1984) and offers complete drawings for the construction of this boat, available from him at Pleasant Beach Rd., South Thomaston, ME 04858 USA.

Philip Thiel
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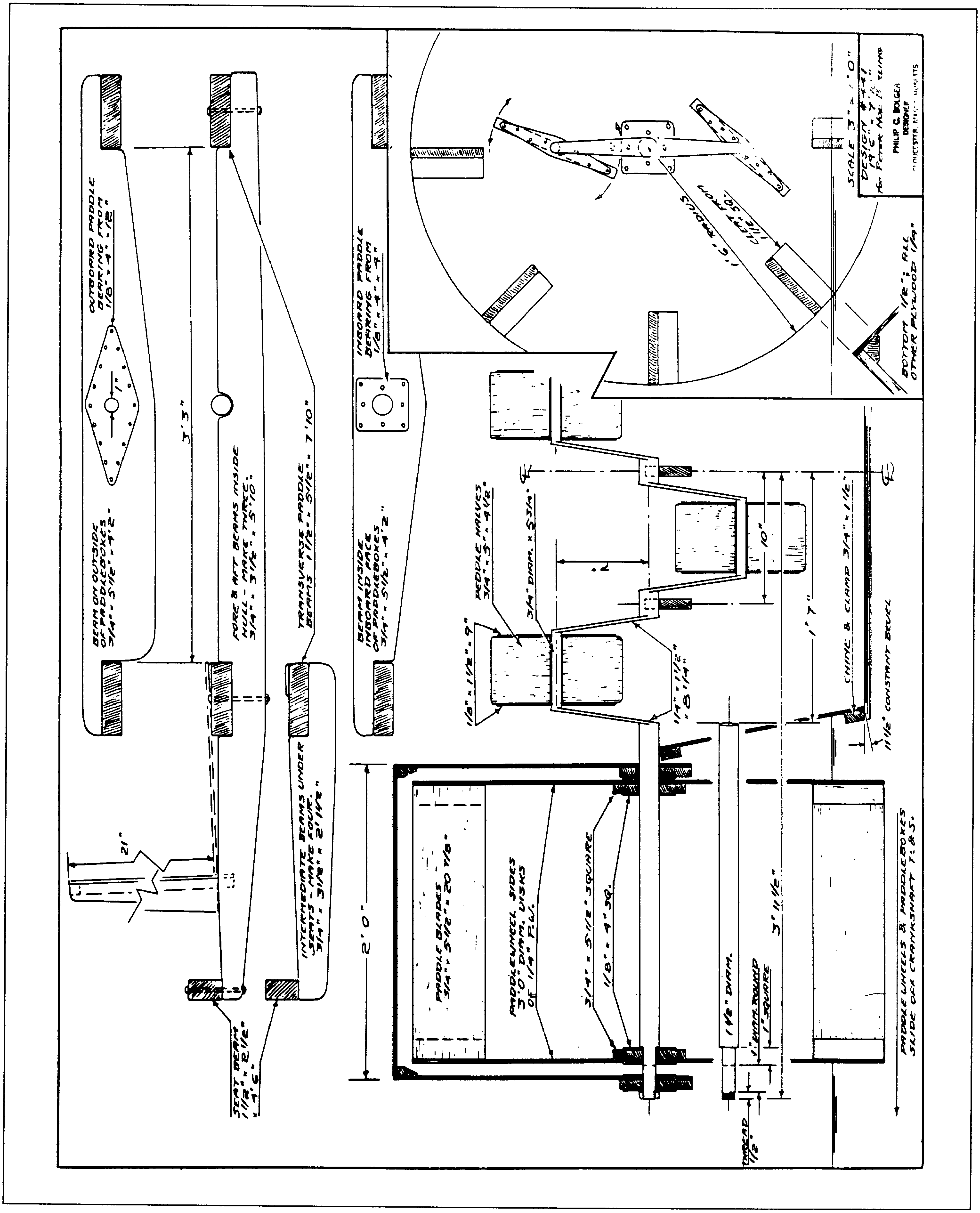
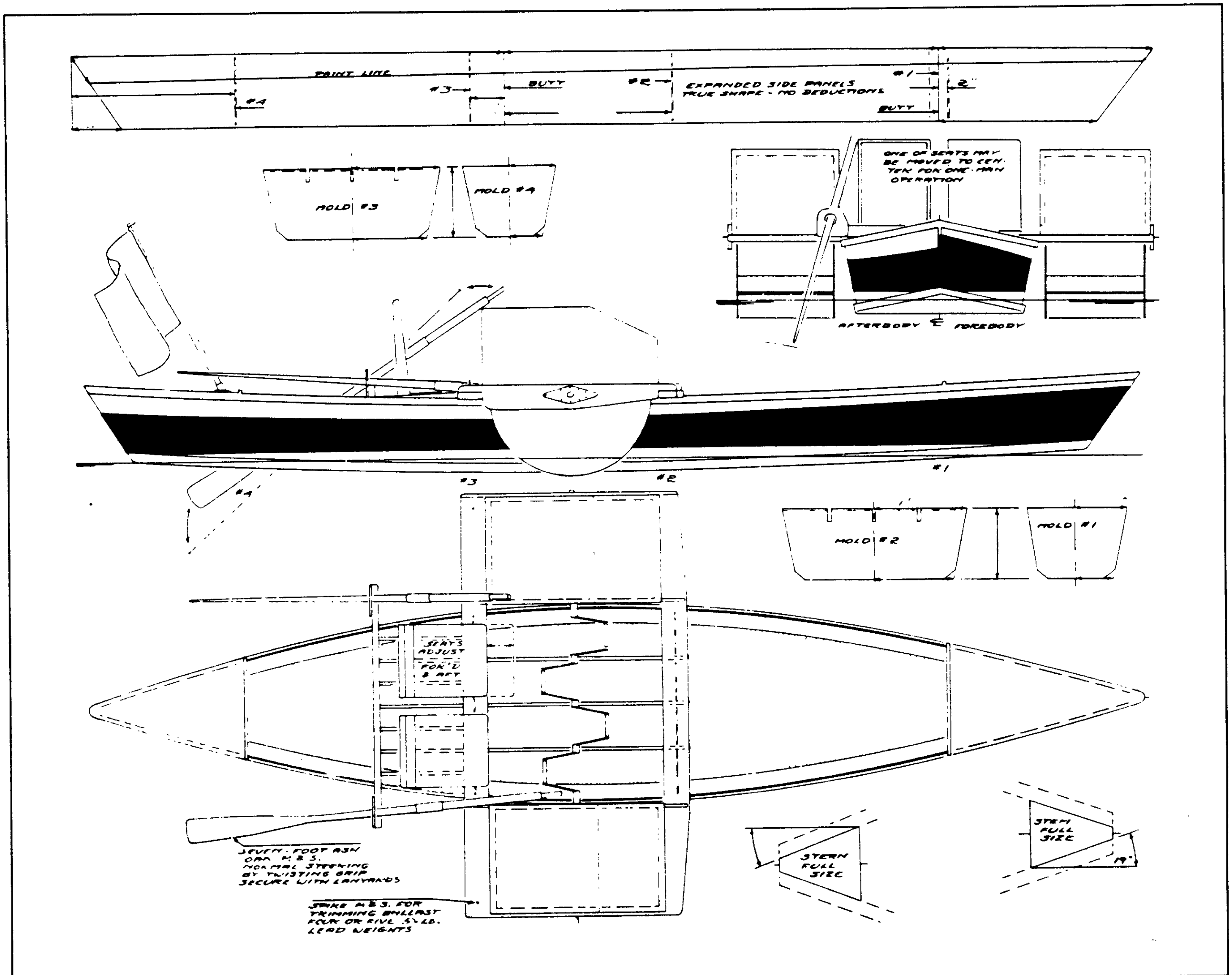


Design for Madeline



Design for Madeline, profile and plan

Conversion Factors*

MASS
1 lbm = 0.4536 kg (kilogram)

FORCE
1 lbf = 4.448 N (newton)

LENGTH
1 inch = 25.4 mm (millimeters)
1 foot = 304.8 mm
1 mile = 1.609 km (kilometers)

AREA
1 sq.ft. = 0.0929 sq.m.

VOLUME
1 cu.ft. = 0.02832 cu.m.

PRESSURE
1 lbf/sq.in. = 6.895 kPa (kilopascals)

STRESS
1 Pa = 1 N/sq.m.
100 kPa = 1 bar = 14.503 lbf/sq.in.

DENSITY
1 lbm/cu.ft. = 16.017 kg/cu.m.

VELOCITY
1 mile/h = 0.447 m/s (meters/second)
=1.609 km/h
1 knot = 0.52 m/s

TORQUE
1 lbf-ft = 1.356 N-m

ENERGY
1 ft-lbf = 1.356 J (joules)
1 Btu = 1054.9 J
1 kcal = 4.186 J
1 kWh = 3.6 MJ (megajoules)

POWER
1 hp = 746 W =746 J/s (watts)
1 kcal/min = 69.78 W
1 ft-lbf/s = 1.356 W

SPECIFIC HEAT
1 Btu/lbm-degR = 4.187 J/kg-degK

HEAT FLUX
1 Btu/sq.ft.-h = 3.154 W/sq.m.
1 kcal/sq.m.-h = 1.163 W/sq.m.

* We will reprint this every few issues

1991 Royal Aeronautical Society Human-Powered Flight Conference, January 30, 1991

Chris Roper, VP-Air

As at such conferences in previous years, those interested in HPF from many countries met and heard a fascinating array of speakers. Chairman Frank Low allowed ample time for discussion.

No prizes were awarded this year: the Kremer Seaplane competition and the Kremer Marathon competition remain open.

John McIntyre of Cambridge, England, gave an account of his Airglow plane which first flew last July. Designed in 1986/7, this aircraft is similar to Daedalus in many ways. Construction has been almost entirely within the McIntyre household by John and his brother Mark. This has included first making the equipment to produce carbon-fibre tubes for the spars and the manufacture of a gearbox. The team had access to a Cambridge University computer and help from Ciba-Geigy and other local firms.

A development is the use of a fly-by-wire control system. Model-aircraft servos are mounted next to the control surfaces, and wires lead between these and the pilot's column.

John said that his design philosophy has been to err on the side of making any component slightly heavier if this will give an aerodynamic advantage. The fuselage frame is very robust to provide stiffness for the transmission, and for pilot safety.

Flight tests so far have shown that the length of the runway at Duxford is covered easily. Pilot Nick Weston states that take-off is quite hard work though; this is blamed on the high rolling resistance of the wheel chosen, a standard glider wheel. With regard to ground crew, McIntyre recommends a launch-team and a separate chase-and-retrieve team. Biggest problem is water-vapor condensation on the wings in the early morning when most flights are made.

Peter Wray of Loughborough University told us about the HPA project that is underway there and which started, he said, at the equivalent RAeS conference two years ago.

This group's constraints are that work must be dovetailed into academic studies, and that the first aircraft must fly in order to get continued support.

Pictures of the oven for carbon-fibre tubes showed it to be a copy of John McIntyre's twin-box-and-blower design, but upgraded to industrial safety standards. This university team's digital foam-cutter is on the principle of that used at MIT, but the wire-holders are not just graph-plotters: they have been engineered from scratch for their purposes.

Wayne Bleisner was expected, but his family had persuaded him not to fly to Britain for fear of reprisals against the US and UK during the Iraq/Kuwait war. However, his paper was read, and we heard of his Marathon Eagle project, the only known direct attempt for the Kremer Marathon prize.

This will have a retractable undercarriage and "rudder" at the back of the pod in addition to a rudder in the usual place. This may "give" when flying with sideslip, or perhaps be active and "push".

The wing-structure will be a carbon-box spar 50mm square at the root, with a monocoque sandwich skin.

Much computer analysis of wing-fuselage interference has led to an unusual planform at the root.

Peter Selinger of the German HVS team described this project. The "H" is for Hutte, glider designer; the "V" for Frank Villinger, who was responsible with Helmut Haessler for the Mufli of 1935. First sketch for the HVS was in 1975, and design was revised in 1978.

Selinger admitted that this project had not been a success. These grandfathers of HPF had built a machine which excessively copied glider principles. The control system alone, while beautifully constructed, must have weighed more than the transmission on some HPAs. The wing was all-molded monocoque and was a perfect shape: later, to try to save weight, the team cut holes in it which they covered with plastic film. Treadle-movement pedals drove the variable-pitch propeller and a spool-drive to the ground-wheel. Rigging procedure involved the pilot being seated first, then the wing was added on top.

The first flight was in June 1982 (wrongly quoted elsewhere), and flights of up to 700 [m]? were made. In 1985 the group disbanded, and the plane now shines down from its perch in a museum.

Peer Frank, now living and working in Virginia, USA, told us of flight tests and pre-flight tests in Germany of his Velair.

For the flight-tests, he developed an 8-channel data logger that was adaptable for many tasks. It recorded pilot heart-rate, prop-revs, control deflection, cabin temperature, stress in the spar and other parameters. This quantitative information was compared with the subjective assessment of the pilot, for instance, that flying in gusty conditions is much harder work.

On the ground an attempt had been made to test the propeller by mounting it on the roof of a car. So far, these tests have been of limited value, since vibration has caused wide scatter in the measurements of thrust.

Other tests have included a proof-test of the spar with deflection-vs-load recorded, and fully-instrumented ergometer tests with four human aero-engines.

The Velair programme is temporarily shelved since "the experience is now being applied to the development of a very-high-altitude research airplane."

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UNITED KINGDOM

Chris Roper is a pioneer in and an enthusiast for HP aircraft, having finished his first design in 1961 (the "Hodgess Roper"). He, and many volunteers, started the construction of his second design, Jupiter, in 1963. (He also married one of the volunteers, Susan Jones). The plane flew many times in 1972. One flight was a then-record distance, 1070 m. Chris has just finished the MS of a book about HP flight, giving technical details and comments on every known flight attempt; we are anxious to see it published, and Chris has given us permission to publish extracts in HP. We are delighted to have such a knowledgeable enthusiast as VP-Air.

Around the world in the human-powered yacht *California*

by Donald W. Spaulding

The goal: to build a boat capable of circling the world powered by one human being–with no motor or sails. HPY *California* is designed to cruise all waters, from the humid leaden stillness of the doldrums to the icy mountainous breaking seas of a Cape Horn gale. No other boat has been designed like her–no other boat had to be. HPY *California* will help re-define how far one average human being can go through an alien environment, using only his wits and muscle.

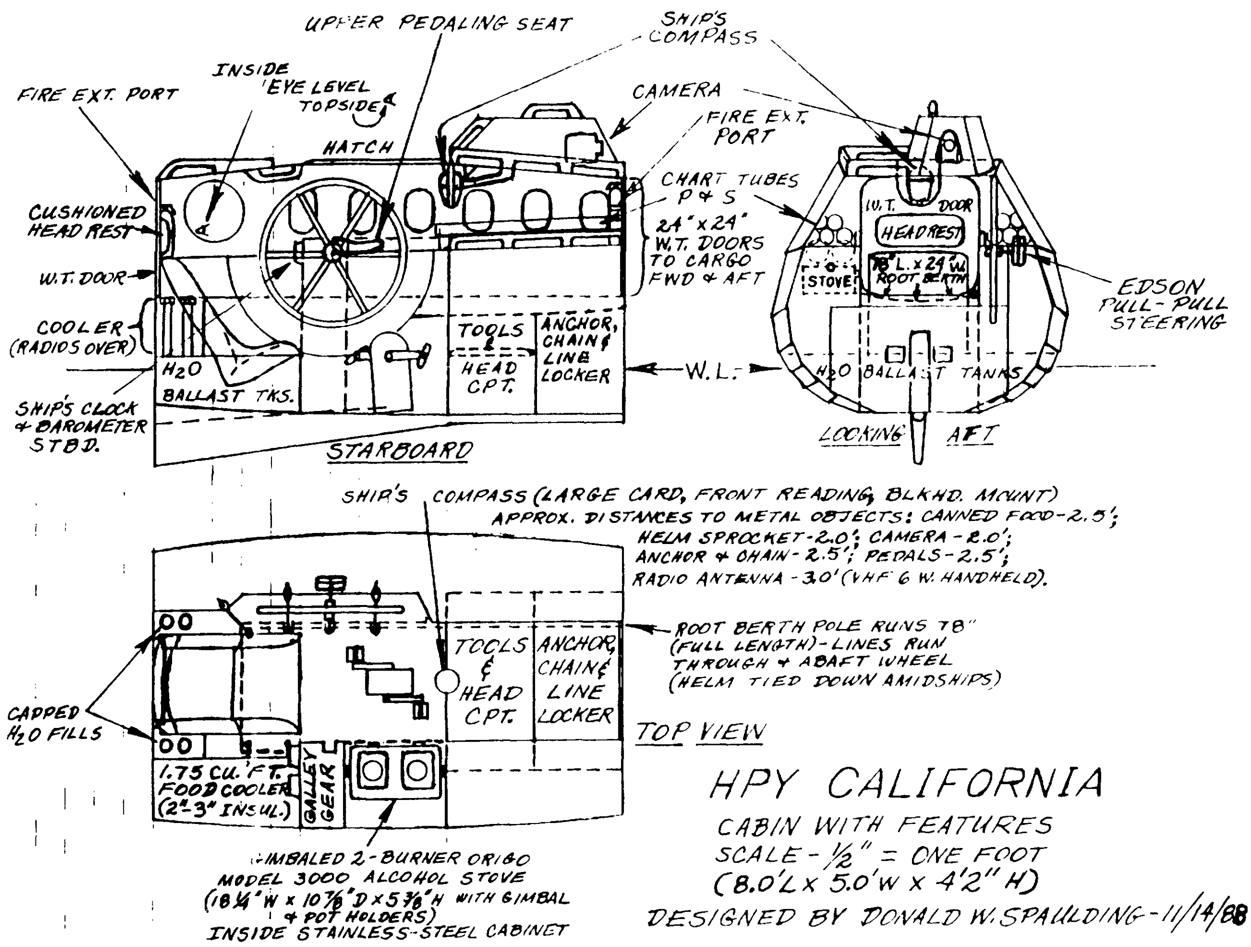
Because she will be entirely self-contained, carrying food for a non-stop 8,000-mile Pacific crossing from California to Australia–a voyage few conventional yachts attempt–the HPY *California* will have to be heavier than most of the boats that brave people have rowed lesser distances between landfalls(1). And since her weight will keep her slow, she must be built strongly enough to survive any storm and resume her leisurely pace afterward. She is capsize-resistant and rapidly self-righting, due to positioning heavier items beneath the middle deck in 24 watertight compartments. And she incorporates 150% positive buoyancy in her design.

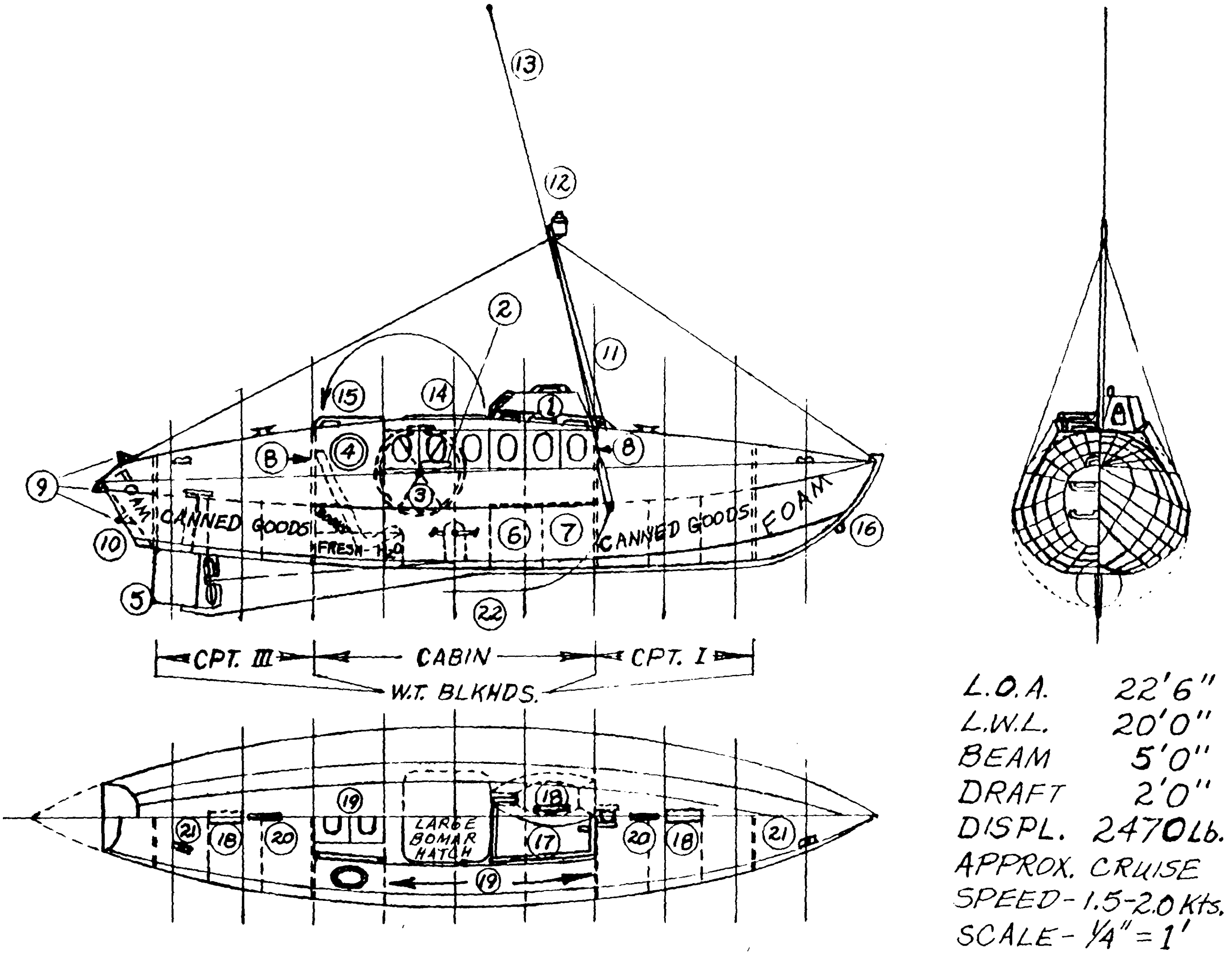
To keep her crew (her designer) snug and safe, *California*'s midship cabin is small, just 8 ft. (2.4 m) long by 54 in. (1.4 m) high, and only 27 in. (0.7 m) wide to the shelf below her G.E. windows. This limits how far one can fall or be thrown. During rollovers and pitchpoles a racing harness will hold the crew securely in his fiberglass bucket seat, where most necessities are within reach.

Cross-flow ventilation in heavy weather, even during repeated rollovers, is provided via "Hurricane Boxes" built into the enlarged full-width camera housing. With no moving parts the boxes contain a unique system of baffles, stacked in four self-draining levels, and draw air through the boat because of the pressure differential between the opposite-facing fore-and-aft clamshell ventilators.

California's revised dimensions are 24 ft. (7.3 m) by 5 ft., 6 in. (1.7 m) with a draft of 30 in. (0.76 m) and a displacement of about 2600 lb. (118 kg). California's bottom and transom are 1 in. (2.5 cm) epoxy-laminated marine ply, and the garboard and side planking are 1/2-in. (1.27-cm) plywood. The entire boat will be sheathed with epoxy-saturated glass and Kevlar. The Harken sprocket-chain and bevel-gear drive system is operated from either the recumbent bucket or the bicycle seat. depending on the weather, and drives a 16-in.- (406 mm-) diameter three-bladed propeller. An epoxy-laminated spruce skeg protects the propeller and bronze rudder. Heavy-duty Edson steering components are used, from the port bulkhead-mounted wheel to the manganese-bronze cruiser rudder.

Now under construction in Arthur Mulvey's Oceanside (California) boat shop, HPY California is due for launching about March 20, and after a 500-mile (805-km) shakedown cruise, is scheduled to leave on April 28 from Newport Beach for a 7-8-month voyage to Sydney, Australia. After a rest and re-fit California will be off through the roaring Forties to round Capes Horn and Good Hope, completing the first human-powered Antarctic circumnavigation at Wellington, New Zealand.





The human-powered yacht *California*

Donald W. Spaulding
P.O. Box 9854
Anaheim, CA 92812

Don Spaulding, 50, a retired Orange County bus driver, served in submarines with the U.S. Navy. His previous long-distance endurance adventures include a 1974 record of 57-1/2 hours coast-to-coast U.S. trips on a Honda 750 motorcycle and live-aboard sidecar he designed, co-driven with his former wife Leanne.

Notes

1. See, for example, Ned Gillette, "Rowing Antarctica's 'Most Mad Seas'", National Geographic, Jan. 1989, for an account of four men who rowed a 28-foot (8.5 m) dory across the 600-mile wide (966 km) Drake Passage.

Numbered features

1. Offset camera housing (insures rapid righting)
2. Bicycle seat for harbor and fair-weather propulsion
3. 30" destroyer-type helm wheel on port bulkhead (for use at either pedal station)
4. Vent. ports P & S
5. Manganese-bronze cruiser rudder
6. Head on C.L.–stove fuel P & S
7. 18 lb bruce, 12 lb. danforth high tensile anchors and chain
8. 24" x 24" w.t. doors to cpts. I and III, with overhead hinges 2" gaskets, 6 dogs.
9. Teak boarding ladder
10. Countersunk transom to assist boarding and help lever bow up in a following sea
11. 6-ft. laminated wooden mast
12. self-contained 6 v. guest nav. lt.
13. 8-ft. stainless radio antenna, grounded through mast stays for lightning protection
14. Large bomar hatch-hinges aft for flow-through ventilation w/open ports
15. Teak handrails/skylight guard
16. Towing and anchoring eye
17. Anchor and promenade deck
18. Prisms atop cpts #I and #III and camera housing
19. 15 Lexan windows, 18' x 12" x 1/2";
   6 each P & S
   2 cabin skylights and
   camera housing waveshield
20. 10" anchor/mooring cleats fwd. and aft of cabin, on centerline

❑