Design and Construction of a High-Speed Human-Powered Boat

by

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ABSTRACT

The goal of this project was to design and construct a high-speed human-powered boat. I considered several design schemes before settling on a fiberglass displacement hull catamaran. The rider is supported in a semi-recumbent position between the hulls by a frame of welded thin-wall steel tubing. Drag data from an eight-man racing shell was scaled to my design displacement to predict a total drag of about 71.2 N (16 lbf) at a speed of 3.8 m/s (12.6 ft/s). To optimize the single propeller, I wrote a program for my calculator to perform the design iterations based on E. Larrabee's algorithms for a minimum-induced-loss propeller. The resultant predicted propeller efficiency was 90%, where propeller efficiency is defined as the ratio of forward speed times thrust of the propeller to the power input at the propeller in corresponding units ($\eta \equiv TV/P$). Neglecting losses in the rest of the drive mechanism, this would require a rider input at design speed of 306 watts (0.41 hp) to achieve the 3.8 m/s (12.6 ft/s) speed. I devised a method to lay out and carve the wooden prototype propeller, then used it as a pattern to cast aluminum-filled polyester replicas.

The original design drawing was adhered to fairly closely during construction, although I made several changes to take advantage of available resources or overcome construction difficulties.

Some design modifications in the drive system remain before the full-speed potential of the completed boat can be realized. Slipping of the twisted timing belt prevented any significant rider input at the pedals, but the boat moved easily and rather swiftly with almost negligible rider input. Improvements are planned to increase the maneuverability of the boat. The rider position was very comfortable, promising application for not only high-speed racing, but also liesurely pedalling.

I was impressed with the interest shown in the boat by bystanders at each launching. With a modification to the drive system to increase

performance and reliability, plus cosmetic changes to the hulls and frame, I believe the boat will not only be capable of significant speed, but also has a potential market in public recreation.

Thesis Supervisor: Dr. David Gordon Wilson

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SYMBOLS AND NOTATIONS

A	$\equiv$	cross-sectional area
C_F	$\equiv$	frictional resistance coefficient, $R_F/1/2\rho V^2S$
C_R	$\equiv$	residual resistance coefficient, $R_R/1/2\rho V^2S$
C_T	$\equiv$	total resistance coefficient, $R_T/1/2\rho V^2S$
D	$\equiv$	diameter
E	$\equiv$	modulus of elasticity
	$=$	2.0×10^5 MPa (29×10^6 psi) for steel
F_i	$\equiv$	force in frame member i due to real load
f_i	$\equiv$	force in frame member i due to virtual, or imaginary, force
F_n	$\equiv$	Froude Number, $V/\sqrt{gL}$
g	$\equiv$	acceleration due to gravity
	$=$	9.8 m/s^2 (32.2 ft/s^2)
L	$\equiv$	length
L.W.L.	$\equiv$	length of hull measured at the water line
P	$\equiv$	power
R_F	$\equiv$	frictional resistance of hull
R_n	$\equiv$	Reynolds' Number, VL/ν
R_R	$\equiv$	residual resistance of hull
	$=$	$R_T - R_F$
R_T	$\equiv$	total resistance of hull
S	$\equiv$	wetted surface area of hull
V	$\equiv$	speed, velocity

Δ	$\equiv$	displacement of hull (units of force)
δ	$\equiv$	linear deflection
η	$\equiv$	efficiency
λ	$\equiv$	scale ratio
ν	$\equiv$	kinematic viscosity
	$=$	1.122×10^{-6} m^2/s (1.208×10^{-5} ft^2/s) for fresh water at 16°C (60°F)
ρ	$\equiv$	density of fluid
	$=$	998.8 kg/m^3 (1.938 $lbf\text{-}s^2/ft^4$) for fresh water at 16°C (60°F)
σ	$\equiv$	stress in frame member
σ_u	$\equiv$	ultimate stress of material
σ_y	$\equiv$	yield stress of material

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1. INTRODUCTION

My intention for this project was to design and build a lightweight and high-speed human-powered boat. There have been several attempts at high-speed human propulsion on water, including several here at M.I.T. [1,2,3] but to my knowledge no one has, as yet, built a working prototype capable of beating a single-man rowing shell.

I hoped that successful completion of my prototype boat, first in a planned series of design iterations, would generate enthusiasm for alternative means of human propulsion on water, particularly at a time when most people are both energy conscious and seeking alternative forms of recreation.

As in any project in engineering design, I was faced with numerous alternatives with the potential to fulfill the design goal. This thesis documents the approach I have taken, and, while I have tried to list the reasons for choosing the approach I did, the reader should recognize that many of the seemingly arbitrary choices were based on my preference and judgment as the designer. I would highly recommend the pursuit of other attempts at a high-speed human-powered boat.

2. OVERALL DESIGN

There are four main areas to the design:

1. the seating position of the rider to allow for comfort and maximum power output;
2. a low-drag method of rider support;
3. an efficient method of propulsion; and
4. a lightweight framework to tie the boat together.

A fifth and at times overriding aspect of the total design was my limited time and resources to design and build a working prototype.

2.1 Seating Position

I decided to seat the rider in a semi-recumbent position. The position has been used successfully on several human-powered land vehicles, including the 80 km/hr (50 mph) plus Vector [4]. Although the advantage of safety in a front-end collision does not really apply in my application, most of the other advantages still hold. The wind drag of a recumbent rider is approximately 20% less than that of an upright seated rider [5, p. 96]. The position allows for a much lower center of gravity, increasing the boat's stability. The rider is constrained by a firm backrest against which he can push, allowing him to exert a force greater than his weight. I originally intended to make use of the seating position recommended by D.G. Wilson [6], but later modified the geometry to make use of an available aluminum-and-canvas low camp seat.

2.2 Rider Support

There are several alternatives for low drag rider support:

1. hydrofoil;
2. totally submerged hull;
3. planing hull; and
4. displacement hull.

Several people have investigated the feasibility of a human-powered hydrofoil [1,2]. To reach speeds at which the foils alone would provide the total lift requires either an external (to the boat) method of rider support, an unattractive proposition to me, or an alternative low drag method of support such as the other alternatives listed.

The totally submerged hull is an appealing alternative for it would limit any wave drag to the drag of the connecting structure which supports the rider and pierces the free surface of the water. According to F.H. Todd [7, p. 356], "Volume to volume, the submarine has a greater wetted surface than the ordinary ship, and so starts off with the handicap of greater frictional resistance. The absence of wave-making resistance therefore does not make itself felt until fairly high speeds are reached - perhaps 25 or 30 knots." Todd is referring to large surface ships, which according to Mandel [8, p.4.19] are rarely drivers at Froude numbers, F_n , exceeding 0.5 to 0.6. A 6.7 m (22 ft) boat operating at 3.8 m/s (12.6 ft/s) would have a Froude number of 0.48. Dynamic scaling of the 25 to 30 knot speed mentioned by Todd puts us around 2.7 or 3.0 m/s (9 or 10 ft/s) for the 6.7 m

(22 ft) boat. Unfortunately, a totally submerged hull has no means of compensating for changes in rider weight unless secondary displacement hulls are added, which negate some of the advantages of the submerged hull. Also, submerging the cylindrical hull two or three diameters below the free surface results in a boat which would be awkward to mount as it dropped to the secondary displacement hull(s) as we climbed on.

2.3 Hull Design

I decided to use displacement hull(s) for rider support. John Gillardi's boat [3] was a single displacement hull stabilized by two smaller hulls on outriggers. For several reasons I decided to use two identical hulls with the rider supported between. The catamaran has the advantage of inherent stability, and I thought construction would be easier having only one hull shape. The clearance between the hulls would allow room for a single propeller shaft and drive mechanism without having to make any watertight passageways through the hull.

To design the propulsion system, I needed to estimate the drag of the boat. The two predominant forms of drag for a surface ship are skin friction and wave resistance [8, p. 4.3]. A typical curve for a surface ship of the non-dimensionalized total resistance coefficient, C_T , versus Reynolds' number, R_n , is shown in Fig. 1 [7, p. 313]. Figure 2 [8, p. 4.21] shows the typical form of the non-dimensionalized residual resistance coefficient, C_R , versus Froude number, F_n , for a surface ship. The residual resistance is the difference between total resistance and frictional resistance and is due mainly to wave

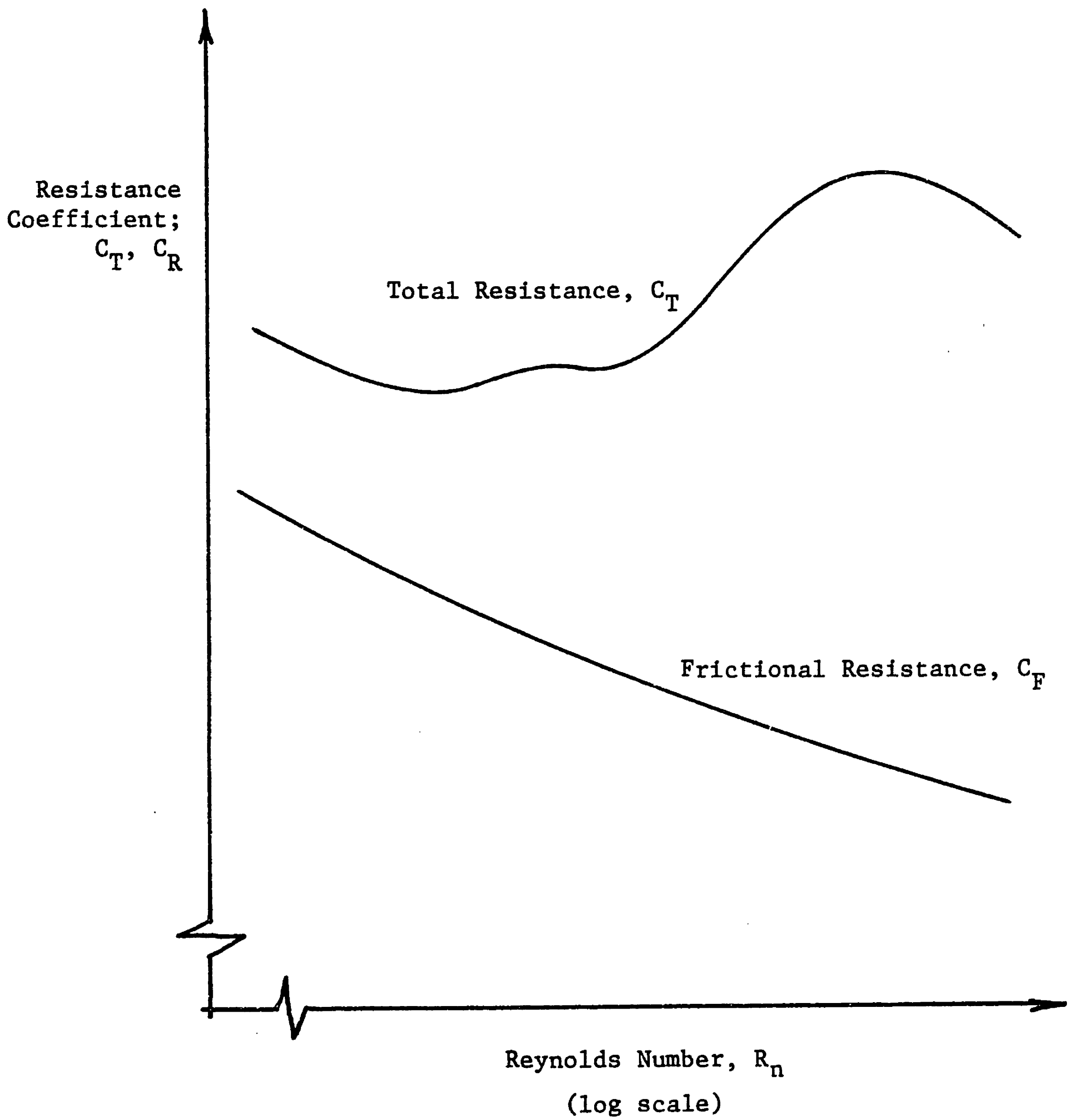

Fig. 1: Resistance coefficient versus Reynolds number for a typical surface ship.

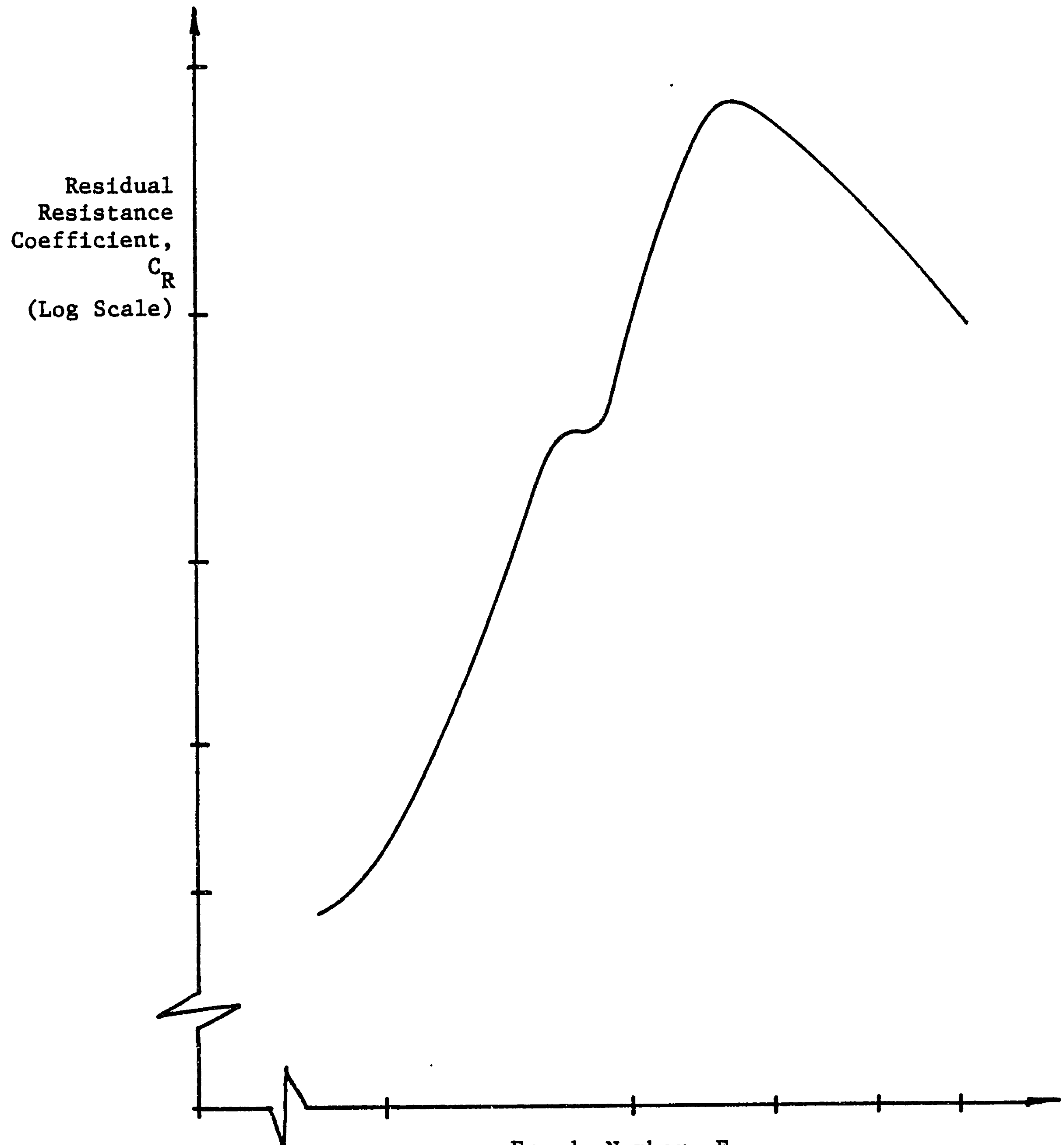

Fig. 2: Typical variation of residual resistance coefficient versus Froude Number for a surface ship.

resistance. For a certain displacement and speed, making a hull longer tends to reduce wave drag as the Froude number is reduced (Fig. 2), while reducing the hull length reduces the wetted surface area and so reduces the friction drag. It would seem that an optimal length could be determined to minimize the total drag for a given speed and displacement. Unfortunately, information such as that shown in Figs. 1 and 2 is empirical in nature. I found little literature available on the drag of small displacement hulls in this size and shape and speed range, so I looked to the configuration of racing shells as guidance for my hull design, assuming their shape had been optimized by trial through the years. The only report on drag experiments done on racing shells I could uncover was the work done by J.F. Wellicome [9], so I decided to dynamically scale his experimental data to determine the size and drag of my hulls. The form of my hull was dictated by my decision to use an existing hull of D.G. Wilson, similar in shape to Wellicome's hulls, as the form from which to mold my own hulls. My approach to the drag scaling follows the method used by naval architects to predict the drag of large surface ships from scaled model experiments. Several references on the technique [7, Chap. VII; and 8, Chap. IV and V] are in the list of references. I estimated the total weight of my boat and rider to be about 1,112 N (250 lb_f), a 556 N (125 lb_f) displacement per hull. Geometric scaling of the displacement of my hull with Wellicome's indicated an overall length of my hull of 6.8 m (22 ft). The drag calculations are contained in Appendix I and indicated an estimated total drag for both hulls of about 71.2 N (16 lb_f) at 3.8 m/s (12.6 ft/s).

If I could achieve an overall propulsion efficiency of 80%, defining propulsion efficiency as the ratio of forward speed times total thrust to power input by the rider in corresponding units, this would require a rider input of 343 watts (0.46 hp).

2.4 Propulsion

Impressed by the apparent success of propellers designed according to E.E. Larrabee's series of algorithms for a minimum induced loss propeller [10, p. 144] - a technique to minimize the energy loss in the slipstream of the propeller - I wrote a program for my calculator incorporating his computational procedures. The inputs which need to be specified by the designer in using this procedure are:

1. desired thrust of the propeller;
2. forward velocity of the propeller;
3. the rotational speed of the propeller;
4. blade element data for each of nine equally spaced radial stations in terms of non-dimensional lift and drag coefficients;
5. the number of blades chosen for the propeller;
6. the density of the fluid; and
7. the radius of the propeller.

The results of the computational procedure are:

1. propeller chord to radius ratio at each radial station;
2. blade angle at each radial station; and

3. the propeller efficiency, η , defined as previously as the ratio of the propeller thrust times forward velocity to the power input (torque times rotational speed) at the propeller.

The desired thrust of my propeller and the forward velocity were 71.2 N (16 lbf) and 3.8 m/s (12.6 ft/s) respectively - the results of dynamic scaling of the hulls to existing experimental data. The remaining inputs, except for fluid density which is dictated by application, are chosen at the discretion of the designer. The rotational speed of my propeller I varied between 400 and 600 rpm. I intended to mount a central propeller shaft between the hulls and I wished to keep the rotational speed low to minimize shaft whipping. Keeping the speed increase between an estimated 80 rpm pedalling speed and the propeller speed on the order of seven or eight to one would simplify the drive mechanism and allow the use of off-the-shelf components. I chose a blade element profile similar to the NACA 4415 profile [11, p. 490 and 491]. The 15% thickness to chord ratio was to increase the strength of the blade over a thinner section. The flat bottom chord would ease fabrication of the propeller. For a choosen propeller profile, the section's lift and drag coefficients are a function of design angle of attack which is at the discretion of the designer. Limiting the propeller to two blades was again to ease construction.

To estimate the diameter of the propeller I used Glauert's expression for ideal propeller efficiency as a function of propeller diameter [12, p. 204], derived from the early momentum theory of propellers where the propeller is idealized as a simple actuator disc with the thrust uniformly distributed over the area. According to Glauert [12, p. 203], the actual propeller efficiency is about 85% of the ideal efficiency.

Larrabee states that [2, p. 143], "The vortex theory of propellers is hence entirely consistent with the early momentum theories...," and also that his computational procedure, "... is consistent with the Betz-Prandtl form of propeller vortex theory." I felt my approach valid for the purpose of estimating propeller diameter.

The calculations are in Appendix II and the results are plotted in Fig. 3. I decided on a propeller diameter of 356 mm (14 in) since little in efficiency is gained for larger diameters, while a deeper draft would limit where the boat could be used. Again I should stress that once the decision is made to use E. Larrabee's computational procedure, the choice of inputs is at the discretion of the designer, and I have simply given my approach in determining design inputs. I performed a series of iterations before deciding on the final propeller geometry. The calcualted propeller efficiency was 90%, requiring a rider input at design speed of 306 watts (0.41 hp).

Joshua Lindsey [2] has also built a propeller to power his attempt at a high-speed human-powered boat using E. Larrabee's computational procedures. Although his propeller was available for my use, I decided to build a propeller of my own design. The diameter of his propeller was about 180 mm (7 in). By increasing the diameter to the 356 mm

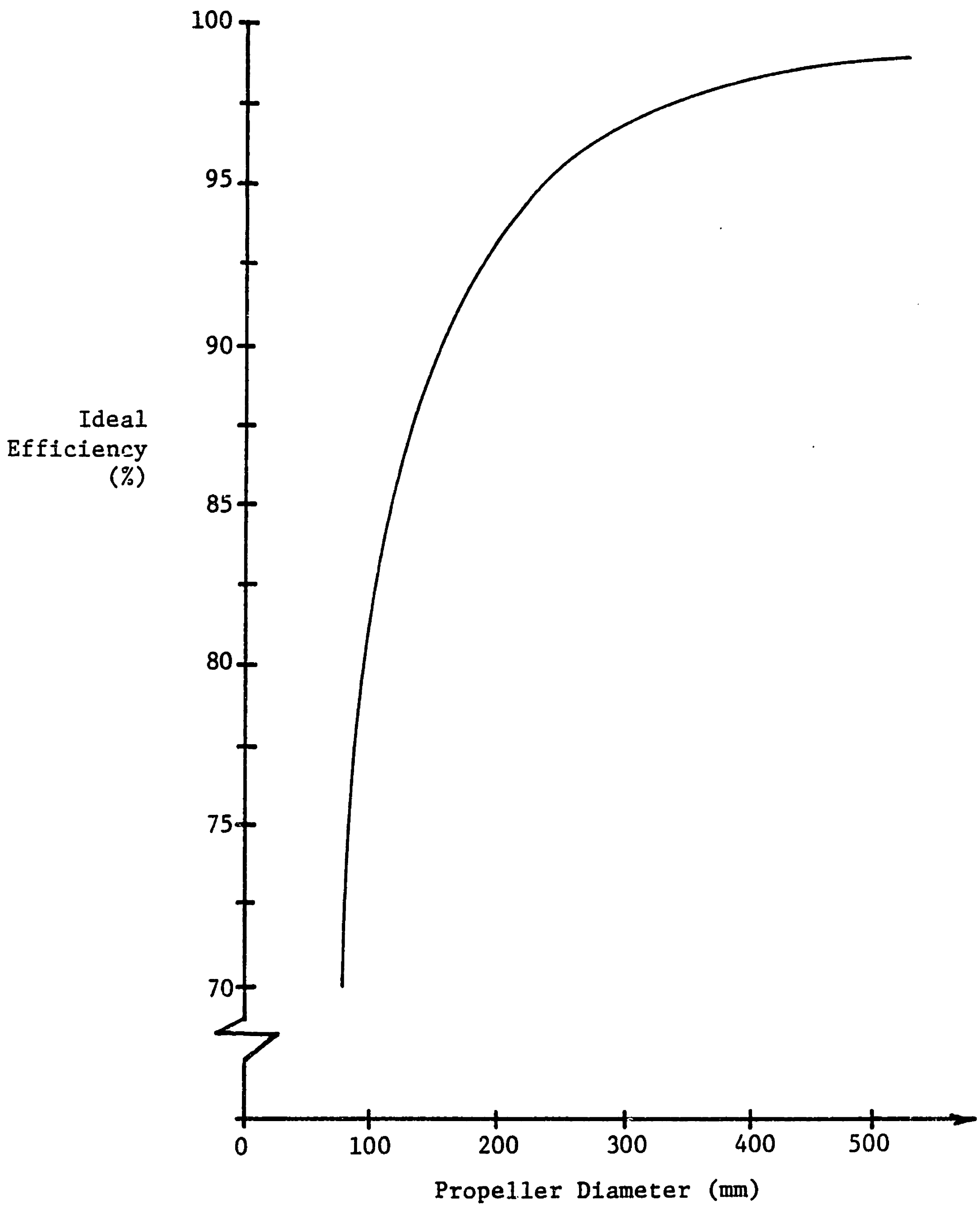

Fig. 3: Efficiency of an ideal propeller versus diameter.

(14 in) diameter I choose, I felt the efficiency of the propeller could be increased. Larrabee's procedure contains approximations pertaining to a "lightly loaded propeller" [13, p. 287]. The loading referred to is the thrust divided by the swept area of the propeller, or disc area. The light loading approximations implies that the contraction of the slipstream behind the propeller (region of increased fluid velocity) can be neglected and the trailing vortex from each element along the radius of the propeller (a concept introduced to explain why the lift of a finite airfoil goes to zero at the tips) can be assumed to lie on a cylindrical surface [14, p. 255]. According to E. Larrabee, a rough rule-of-thumb for determining if a propeller is lightly loaded is when the total thrust per disc area is less than one half the dynamic fluid pressure of $1/2\ \rho V^2$, where the velocity, V, is the forward speed of the propeller, and ρ is the fluid density. I felt that increasing the diameter of my propeller to 356 mm (14 in) would be more in keeping with the lightly loaded approximations, resulting in a more valid prediction of the propeller performance. Joshua Lindsey also intended to drive his propeller at 2,000 rpm. I felt this would cause excessive shaft whipping in my application.

I should also note that Joshua Lindsey's propeller was manufactured of aluminum on a milling machine specially designed to construct propellers. My desire was to develop a construction technique which would allow others without access to this specialized equipment to manufacture a propeller. The calculated efficiency of Lindsey's propeller was 87%. I think it would be interesting to compare the actual performance of his and my prototype propellers to give some insight into the computational and construction procedures.

2.5 Framework

The final step in the design was the framework to tie the boat together. I wanted to keep the frame simple for the purposes of analysis and construction. An interesting approach would be to use a mathematical technique [15,16] to design a structure of theoretically minimum weight. Unfortunately, to use such a method requires a precise identification of the loading to be imposed on the structure. In a boat like this, the loading is hard to identify and quantify - the designer must try to determine the effect of wave loads, collision with objects, pulling the boat up on the beach by one hull, mounting the boat in a way not recommended, and other static and dynamic loads the boat may see. My approach was to incorporate a number of truss-like sections for the framework, using my experience as guidance in determining frame member size. I decided on a maximum rider weight of 1,112 N (250 lb_f), then performed a simple stress analysis (Appendix III) with this static load on the main truss to satisfy myself that the members were stressed well below their yield stress. I assumed that unknown loads on the structure would be handled by the generous factor of safety (ratio between yield stress of material and stress due to applied static load of 1,112 N (250 lb_f)) in the framework.

Because of cost and ease of fabrication, I decided to use thin-walled cold-drawn steel tubing. To allow the boat to be easily transportable on land, the framework was fastened to the hulls with slip joints and the long braces were simply bolted to tabs on the frame. I allowed as much adjustment in the prototype frame as possible - the bearing support for the propeller shaft forward and aft were fitted in clamped vertical slip joints, the seat position could slide forward and aft along the frame, and the pedal bracket was bolted to the frame through slotted holes. In addition, the forward propeller-shaft bearing and the housing I made for the rear-propeller-shaft non-metallic bushing were both self-aligning to accommodate changes in the angle of the propeller shaft. My intention was to strengthen many of the areas of adjustment by making permanent and rigid connections, after I had satisfied myself of final dimensions from the initial boat trials. A sketch of the final design is shown in Fig. 4 showing the frame attached to the hull bulkheads. Some braces have been left out of the sketch for clarity.

2.6 Transmission

I originally intended to turn the propeller with a 3.0 m (120 in) length of a flexible chain produced by the Winfred Berg Company. The chain is similar to a bicycle roller chain, but has two stainless-steel cables joined by molded pins. I had hoped the chain would be flexible enough to make the 90° twist required between the axis of propeller shaft rotation and bicycle crank rotation. Aligning the chain turned out to be too difficult, so I decided to try an alternative drive. I mounted a countershaft on the frame about 0.6 m (2 ft) from both the propeller shaft and pedal crank. A 3/32 in wide bicycle chain connected the pedal crank and countershaft with a 49:18 speed increase.

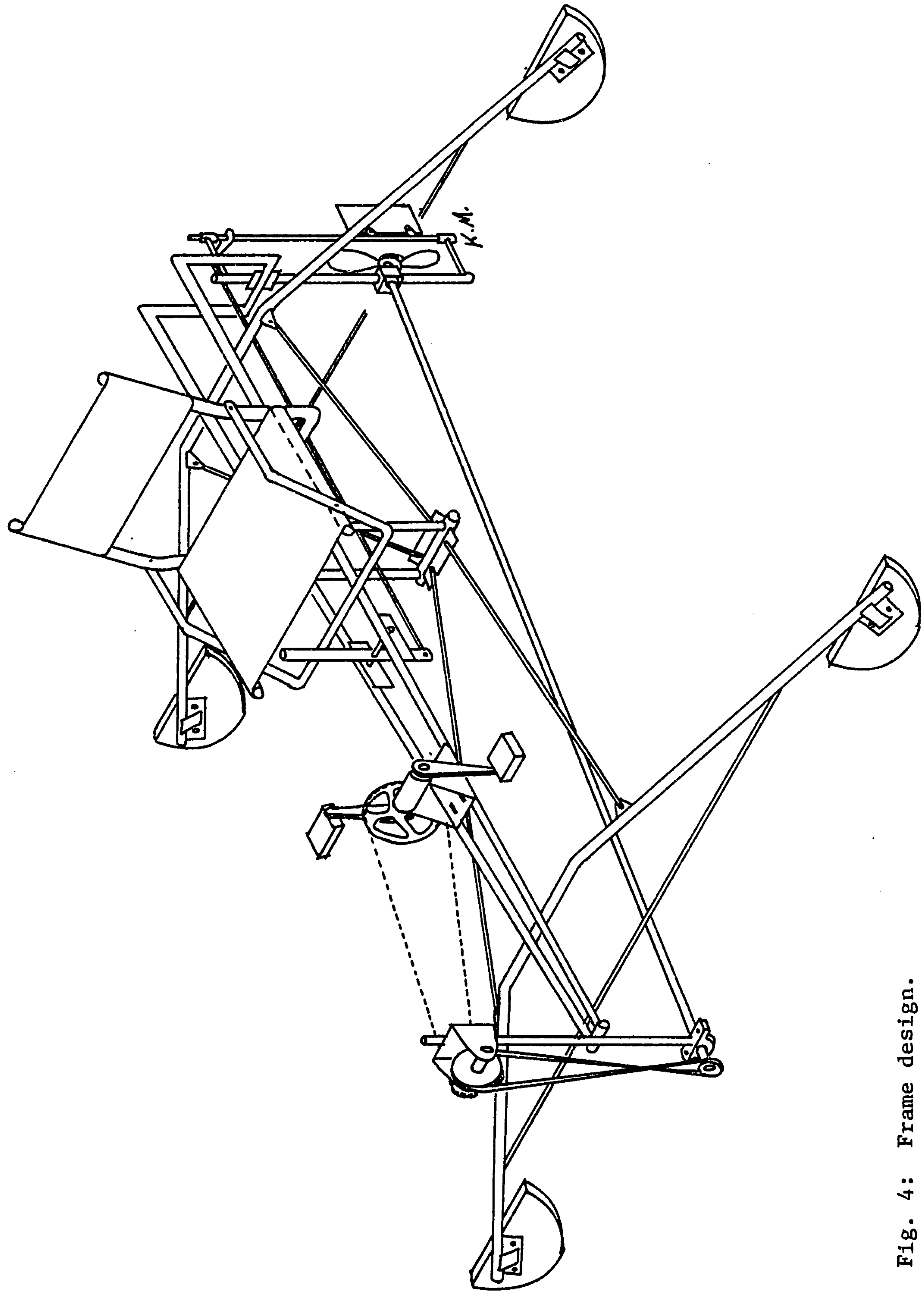

Fig. 4: Frame design.

I then used a single timing belt with a 2.3:1 speed increase to make the 90° twist required.

3. TRIALS

Under no-load condition the drive scheme worked very well. The first time the boat was launched, there was a problem with the steel bicycle chain alignment caused by too much flexibility in the counter-shaft mounting. Because of that, it was only possible to pedal very slowly, nowhere close to the 80 rpm designed pedal speed. In spite of the problems the boat seemed to move fairly well against a very strong headwind and current. Unfortunately, the first trial was brief, for while I was standing toward the bow of one hull realigning the chain, the waves were lapping over the gunwale of the hulls. Enough water lapped into the hull to lower the gunwale to water level and the rest of the hull filled very quickly. The hull had no additional displace-ment and the boat quickly sank from sight. Fortunately, the water was fairly shallow and, with the help of some amused bystanders, we pulled the boat back to shore.

For the next launching I relocated and added braces to stiffen the countershaft mounting. I also fastened nine one-gallon milk jugs in each hull to prevent another sinking. This time the weather conditions were much better, and we picked a launching spot slightly protected from the river's current. I was unable to obtain any indication of speed because the timing belt would skip whenever any significant effort was applied to the pedals. The seating position was very comfortable, ap-propriate for both liesurely pedalling and a strenuous output by the rider.

4. DISCUSSION AND RECOMMENDATIONS

I was pleased with the basic design of the boat, although there are several modifications I plan on prior to my next launch. The steel truss-like framework is light-weight and was easy to fabricate. The countershaft mounting is too flexible and the drive system needs to be redesigned. There is available on the market a 3-D chain that I believe would be appropriate for making the 90° twist required. The chain consists of a single strain member with a series of pins at right angles to each other and has been designed for the purpose of making a right-angle change in rotation. I have also located a cast reinforced nylon bevel-gear set that would be appropriate for my application. My intention is to use one of these two components, probably the bevel-gear set, in addition to redesigning the front of the frame to make the drive assembly much more compact and rigid. It was only problems in the drive which prevented full exploration of the propeller's potential. The system of casting the propeller allows me to have spares readily available.

I located the rudder directly behind the propeller for several reasons. This position allows steerage at low speeds, and also allows the entire frame to be removed from the hulls by four simple slip joints without rigging up steering between hull-mounted rudders and the rider's position. The rudder position was located about two-thirds aft from the bow. The rounded cross-section hulls had little resistance to lateral motion. These two features combined to give poor maneuverability; for turning the rudder tended to push the entire boat sideways. A crosswind also produced the same result. I believe the cure will be in mounting a full-length narrow keel along the bottom of the hulls, or possibly a single keel toward the bow of each hull. I also plan on enlarging the size of the rudder slightly.

The bottom bearing for the rudder post is below the propeller. This increases the draft of the boat slightly, but it gives necessary protection to the propeller. The original rudder post on my frame was adjustable to allow for varying the operating depth of the propeller during initial trials. A permanent and more rigid design will prevent damage to the rudder post itself.

Construction of the fiberglass hulls was the most time consuming part of the project. After my first sinking, I am convinced totally enclosed hulls are necessary, with internal foam bulkheads to prevent buckling due to shear flow, and also to provide the necessary buoyancy in the event the watertight integrity of the hull is violated.

APPENDIX I: DRAG CALCULATIONS

"m" subscript denotes my intended hull.

"s" subscript denotes eight man shell number II in Wellicome's report.

Comparing the displacements, Δ , of the two hulls gives the scale ratio, λ .

$$\Delta_s = 8{,}509 \text{ N } (1{,}913 \text{ lb}_f)$$

$$\Delta_m = 556 \text{ N } (125 \text{ lb}_f) \text{ (assuming a total weight of boat and rider of } 1{,}119 \text{ N } (250 \text{ lb}_f))$$

$$\lambda = \left(\frac{\Delta_s}{\Delta_m}\right)^{1/3} = \left(\frac{8{,}509 \text{ N}}{556 \text{ N}}\right)^{1/3} = 2.48 \text{ (assuming both hulls operating in water of the same density)}$$

Assuming geometric similarity between hulls, the scale ratio is used to find the length of hull "m."

$$(\text{L.W.L.})_m = \frac{(\text{L.W.L.})s}{\lambda} = \frac{16.8 \text{ m}}{2.48} = 6.8 \text{ m } (22.2 \text{ ft})$$

Froude number similarity requires:

$$\left(\frac{V}{\sqrt{gL}}\right)_m = \left(\frac{V}{\sqrt{gL}}\right)_s$$

A speed of 6.05 m/s (19.87 ft/s) for the "s" hull scales to a speed for the "m" hull of:

$$V_m = V_s\sqrt{\frac{L_m}{L_s}} = V_s\sqrt{\frac{1}{\lambda}} = (6.05 \text{ m/s})\sqrt{\frac{1}{2.48}} = 3.8 \text{ m/s } (12.6 \text{ ft/s})$$

At 6.05 m/s, $(R_T)_s$ = 441 N (99.17 lb_f). The total resistance is the sum of frictional resistance and residual resistance,

$$R_T = R_F + R_e$$

and the non-dimensional total resistance coefficient is the sum of the frictional and residual resistance coefficients,

$$C_T = \frac{R_T}{1/2\rho SV^2} = C_F + C_R$$

where

$$C_F = \frac{x}{\left[\log_{10}\left(\frac{R_n}{100}\right)\right]^2} = \frac{R_F}{1/2\rho SV^2}$$

and the variable x is found so that a plot of the skin-friction line is tangential to the low-speed end of the experimentally determined resistance curve; $x = 0.0816$ according to experimental data of Wellicome [9, p. 10].

The Reynold's number of "s" is:

$$(R_n)_s \equiv \frac{V_s L_s}{\nu_s} = \frac{(6.05 \text{ m/s})(16.8 \text{ m})}{(1.122 \times 10^{-6} \text{ m}^2/\text{s})} = 9.04 \times 10^7$$

from which the frictional resistance coefficient is found:

$$(C_F)_s = \frac{0.0816}{\left[\log_{10}\left(\frac{9.04 \times 10^7}{100}\right)\right]^2} = 0.00230$$

The frictional resistance can now be found as

$$(R_F)_s = (C_F)_s 1/2\ \rho_s S_s V_s^{\ 2}$$

$$= (0.00230)(998.8\ kg/m^3)(9.54\ m^2)(6.05\ m/s)^2(1/2)$$

$$= 402\ N\ (90.4\ lb_f)$$

The residual resistance of "s" is the difference

$$(R_R)_s = (R_T)_s - (R_F)_s$$

$$= 441\ N - 402\ N$$

$$= 39\ N\ (8.80\ lb_f)$$

Assuming my intended hull "m" also is operating in 16°C (60°F) fresh water and dynamically scaling the residual resistance:

$$(R_e)_m = \frac{(R_e)_s}{\lambda^3} = \frac{30\ N}{(2.48)^3} = 2.56\ N\ (0.58\ lb_f)$$

We now need to find the frictional component of the total resistance for the "m" hull:

$$(R_F)_m = (C_F)_m\ 1/2\ \rho_m S_m V_m^{\ 2}$$

S_m is found by assuming geometric similarity with "s":

$$S_m = \frac{S_s}{\lambda^2} = \frac{9.54\ m^2}{(2.48)^2} = 1.55\ m^2\ (16.70\ ft^2)$$

$$(Rn)_m = \frac{V_m L_m}{\nu_m} = \frac{(3.8 \text{ m/s})(6.8 \text{ m})}{(1.122 \times 10^{-6} \text{ m}^2/\text{S})}$$

$$= 2.32 \times 10^7$$

$$(C_F)_m = \frac{0.0816}{\left[\log_{10}\left(\frac{(R_n)_m}{100}\right)\right]^2} = \frac{0.0816}{\left[\log_{10}\left(\frac{2.32 \times 10^7}{100}\right)\right]^2}$$

$$= 0.00283$$

Substituting values we find:

$$(R_F)_m = (C_F)_m \, 1/2 \rho_m S_m V_m^{\,2}$$

$$= (0.00283)(1/2)(998.8 \text{ kg/m}^3)(1.55 \text{ m}^2)(3.8 \text{ m/s})^2$$

$$= 32.5 \text{ N } (7.30 \text{ lb}_f)$$

The total resistance of "m" is the sum of the frictional and residual resistance components.

$$(R_T)_m = (R_F)_m + (R_R)_m = 32.5 \text{ N} + 2.56 \text{ N}$$

$$= 35.1 \text{ N } (7.88 \text{ lb}_f)$$

Again, I should emphasize that these calculations were performed to estimate the drag of my intended hulls as an input into propeller calculations. I assumed that the drag of the total boat was due to the sum of the individual drag of each hull with no interaction effects.

It is interesting to note that, according to these calculations, the frictional drag accounts for greater than 90% of the total drag of each hull. This points to the advantage of using a round cylindrical hull to minimize the wetted surface area of the hull. Keeping the wetted surface of the hull as smooth as possible will also reduce the frictional drag.

The small fraction that residual resistance (2.56 N) comprises of the total resistance (35.1 N) for the scaled hull would perhaps indicate that the length of the hull could be shortened. Although increasing wave drag, the frictional drag would be reduced because of the decrease of wetted surface area of the hull, with perhaps the overall effect of reducing the total drag. An experimental investigation would be worthwhile for hulls of this general form and speed, for there is a great deal of interest in the area of human-powered water vehicles. Model testing in a calibrated towing rank would be appropriate.

APPENDIX II: IDEAL EFFICIENCY OF A PROPELLER

According to Glauert [12, p. 204], an expression for the ideal efficiency, η , of a propeller when it is treated as a uniformly loaded actuator disc is:

$$\frac{1-\eta}{\eta^3} = \frac{2}{\pi}\,\frac{P}{\rho V^3 D^2}$$

The propeller efficiency is defined as

$$\eta \equiv \frac{TV}{P}^{*}$$ (*This is the definition of propeller efficiency adopted by E. Larrabee and H. Glauest.)

Substituting for P in the top equation results in the following expression for propeller efficiency in terms of propeller diameter.

$$\frac{1-\eta}{\eta^2} = \frac{2}{\pi}\,\frac{T}{\rho V^2 D^2}$$

Given our previously determined design values:

$V = 3.8$ m/s (12.62 ft/s)

$P = 998.8$ kg/m^3 (1.938 lb$_f$ - s^2/ft^4)

$T = 71.2$ N (16 lb$_f$)

we can obtain the values in Table 1, shown graphically in Fig. 3.

TABLE 1

IDEAL EFFICIENCY OF A PROPELLER
AS A FUNCTION OF DIAMETER

Efficiency, η	Diameter, D [mm (in)]
0.990	554. (21.58)
0.980	388. (15.11)
0.975	345. (13.44)
0.950	238. (9.26)
0.925	189. (7.36)
0.900	159. (6.20)
0.875	139. (5.40)
0.850	123. (4.78)
0.825	110. (4.30)
0.800	100. (3.90)
0.775	91. (3.56)
0.750	84. (3.27)
0.725	77. (3.01)

APPENDIX III: FRAME ANALYSIS

In the small sizes of thin-wall steel tubing I wanted to use for the frame, the tubing available was drawn-over-mandrel 1020 steel with a minimum yield strength of 410 MPa (60,000 psi) and an ultimate strength of 480 MPa (70,000 psi). I planned to use 19.0 mm (0.74 in) outside diameter by 0.89 mm (0.035 in) wall tubing for the main members and 9.65 mm (0.38 in) outside diameter by 0.56 mm (0.022 in) wall tubing for the secondary braces. My approach to the strength analysis was to model the main part of the frame as a pair of pin jointed trusses, each carrying half of the total static design load of 1,112 N (250 lb_f). This simplified analysis would indicate if any of the members were stressed close to their yeild point. If so, I would redesign the frame or perform a more exact analysis. The joints in the actual frame were bolted or welded - the assumption of pin jointed connections allow a simple analysis and is valid for a frame such as this as long as all deflections are small. I first calculated stresses in each member assuming pin joints, and then checked the assumption of small deflections using the principle of virtual work. Additional reading on the principle of virtual work can be found in the list of references [16, 17,18].

The frame model is shown in Fig. 5 with half the design load of 1,112 N (250 lb_f), 556 N (125 lb_f), acting as a force W on joint b. Each pin joint is designated by a letter and the frame members by a number. The numbers in brackets indicate the length of the member.

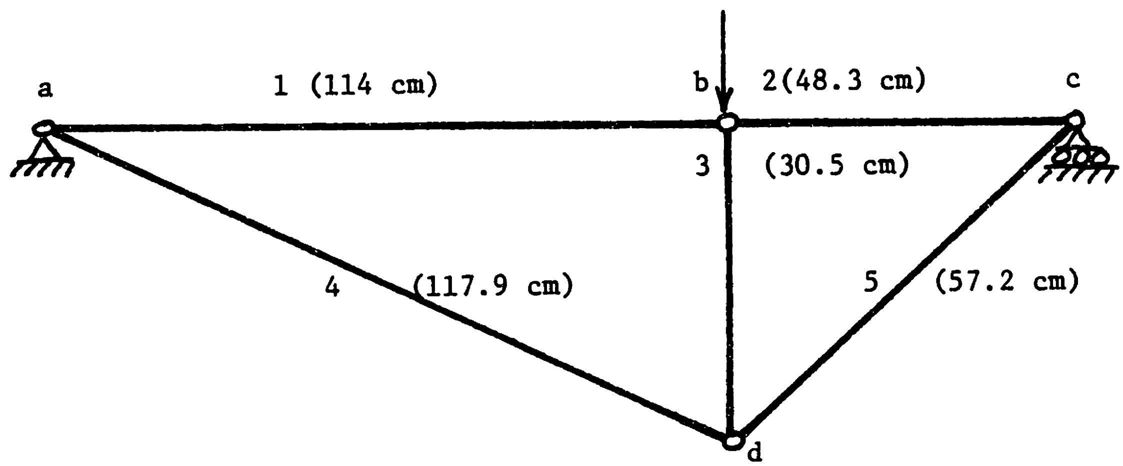

Fig. 5: Model of central frame truss.

By isolating each pin joint separately the forces on the individual frame members, F_i, were found.

To find the vertical deflections of point b under the static load W, I first apply an imaginary, or virtual, one Newton vertical load at b to the unstressed structure, assuming the load is small enough to cause no deflections to the structure. I then apply the real load, W. The external virtual work done as the one Newton load moves through the real deflection, δ, is equal to the total internal virtual work done as the virtual loads in the individual frame members, f_i, due to the virtual one Newton load, move through the real deflection of the individual members caused by the real load W :

$$(1\ \mathrm{N})(\delta) = \sum_{i=1}^{5} \frac{f_i F_i L_i}{A_i E_i}$$

This is the principle of virtual work.

Table 2 gives the results of the calculations, with compressive forces and stresses negative and tensile forces and stresses positive. Note the stresses in all members are well below the yield point of the material. In my judgment the frame is sufficiently strong to accommodate any reasonable additional unknown loads which may be applied to it.

The vertical deflection under point b due to the applied static load, W, of 556 N is:

$$\delta = \frac{\sum_{i=1}^{5} \frac{F_i f_i L_i}{A_i}}{(1\ \mathrm{N})(E)}$$

$$= \frac{(31,303\ N^2/mm)}{(2.0 \times 10^5\ MP_a)(1\ N)} = 0.16\ mm\ (0.006\ in)$$

With such a small deflection the initial assumption of negligible bending stresses remains valid.

TABLE 2

FRAME ANALYSIS RESULTS

Member, i	A_i [mm^2]	L_i [mm]	F_i [N]	f_i [N]	σ_i [MP_a (lbf/in^2)]	$\frac{F_i f_i L_i}{A_i}$ [N^2/mm]
1	50.7	1,143	-307	-0.55	-6.06(-878)	3,810
2	50.7	483	-307	-0.55	-6.06(-878)	1,614
3	50.7	305	-556	-1.00	-11.00(-1,590)	3,345
4	15.7	1,179	320	0.58	20.30(1,902)	13,805
5	15.7	572	365	0.66	23.19(3,361)	8,728

$$\sum_{i=1}^{5} \frac{F_i f_i L_i}{A_i} = 31,303 \ N^2/mm$$

APPENDIX IV: HULL CONSTRUCTION

Construction started with fabrication of the hulls, which was by far the most time-consuming part of the project. My intention was to make a mold from D.G. Wilson's hull. I would then fabricate four identical hulls and join them in pairs to form the two totally enclosed hulls. I ripped up two 1.2 m by 2.4 m (4 ft by 8 ft) sheets of plywood and laminated a table 0.6 m (2 ft) by 7.3 m (24 ft). I cut the top off the Wilson hull and then removed a section from the middle to achieve my design length. The hull was turned upside down and fastened to the plywood table. After a great deal of smoothing and fairing with automotive body putty, this would be the plug to fabricate the mold from which I could produce the four hulls. Unfortunately, removing the fiberglass mold from the plug was accompanied by tearing out large areas of gelcoat on the inside of the mold, in spite of four coats of mold-release wax. Again, a great deal of preparation was necessary to smooth the inside of the mold before laying up the first hull. This time I sprayed several coats of automotive sanding sealer on the inside of the mold to fill all voids and followed with four coats of mold-release wax. This helped, but did not eliminate the problem, for removing the first hull layup was again accompanied by tearing out portions of the interior of the mold. To save time I decided to use the mold and the first hull layout as the two boat hulls, leaving the top open for initial trials and covering them later when time allowed. The mold had three layers of ten-ounce fiberglass cloth while the first hull layup had only two, but the weight difference was not significant. The hulls were completed by fiberglassing in some plywood bulkheads and bolting some crossbraces in to prevent the thin hull from buckling under the shear flow, and bolting some steel brackets to the bulkhead to receive the ends of the frame slipj ints.

APPENDIX V: PROPELLER CONSTRUCTION

I decided to construct the propeller out of wood, since it would be the easiest material to shape. Of the species of wood common to early aircraft propeller constructions: walnut, birch, oak, and Honduras mahogany [19, p. 213], I decided to use birch since it is the strongest and is easily available. The rough cut block of birch I obtained was resawn and planed into 6.3 mm (0.25 in) thick strips. I then laminated the strips together with resorcinol resin to obtain a block 356 mm by 51 mm by 25 mm (14 in by 2 in by 1 in). Of the two appropriate glues for the propeller block - epoxy and resorcinol resin - the resorcinol has a longer history of similar use. Epoxy has good gap-filling properties and should be used on glue joints with 0.07 mm to 0.13 mm (0.003 in to 0.005 in) clearance. The strips I was laminating had very smooth planed surfaces negating the need for any gap filling, and I intended to apply a high clamping pressure to produce a very thin glue line - consistent with requirements for resorcinol bonding.

After contemplating several schemes to carve the propeller, the one I finally devised was simple and effective. The laminated block was trued to outside dimensions of 335.6 mm x 50.8 mm x 25.4 mm (14 in x 2 in x 1 in), which were the outside boundaries of the finished propeller. A radial centerline was laid off and divided into nine equally spaced radial stations, marked on the block as shown in Fig. 6a. Next the trailing edge was drawn on one face of the block and the point on each radial station was marked where the projected

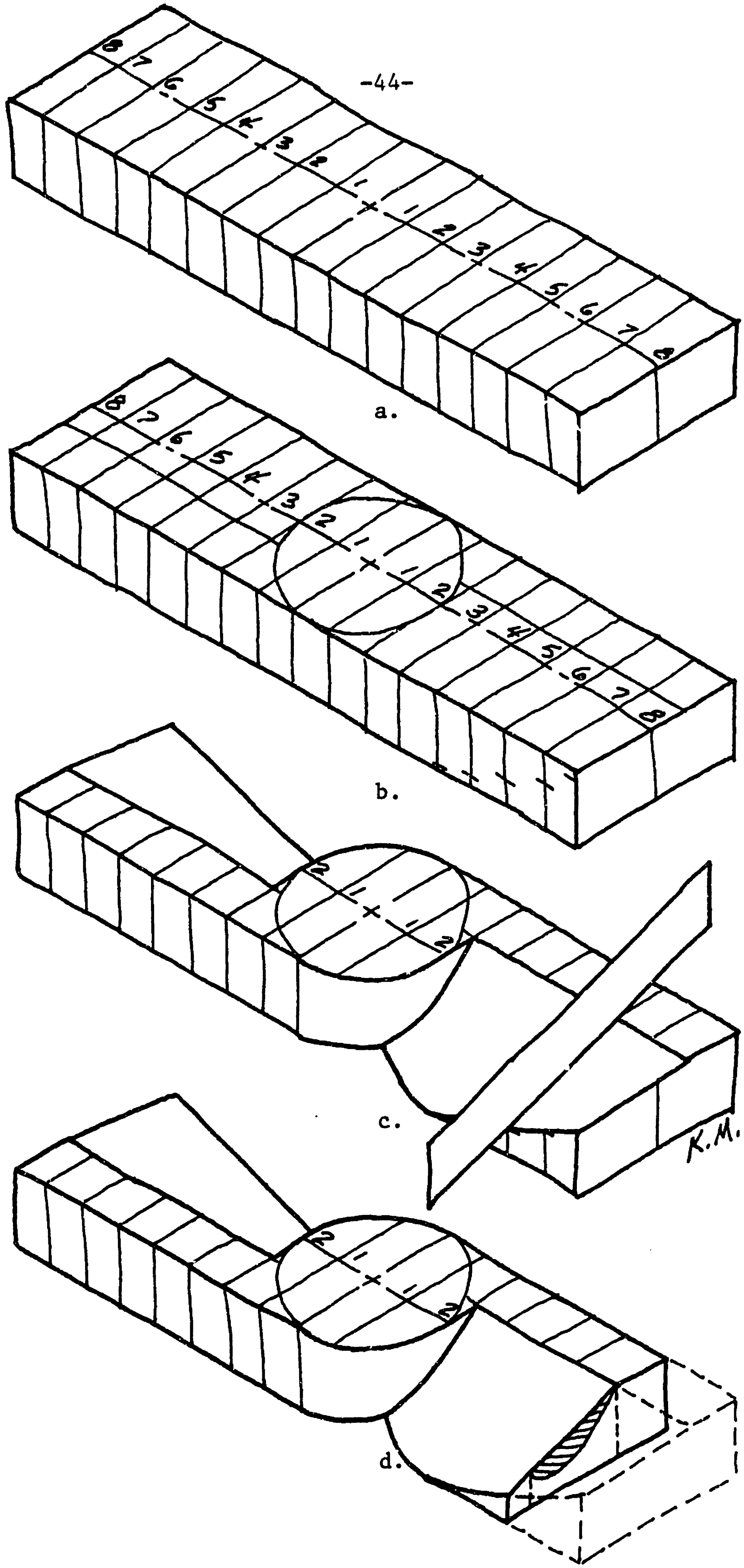

Fig. 6: Block layout steps for propeller construction.

bottom chord intersected the outside of the block (Fig. 6b). The hub diameter was also marked on the block at this time. A wood rasp and file were then used to fair the wood until a straight edge could be laid along the radial line between the two points (Fig. 6c). Next I marked points on the block opposite the face where the trailing edge is located where a perpendicular line from the face would intersect the leading and trailing edge of the propeller. Figure 6d shows these two points located at one radial cross-section. The points are then connected showing the projected outline of the propeller. I then band-sawed the propeller outline, keeping the saw cut perpendicular to the face of the block. It is a good idea to saw cut leaving a slight excess and then use a sanding dise to achieve the final outline. The final shaping of the propeller was done by hand using a file and a wood rasp.

To accurately guide the profile of the propeller I originally intended to lay out aluminum templates scaling my chosen airfoil profile to the corresponding chord length at each radial station. In traditional wooden propeller construction, the form at various radial stations is guided in this manner by templates, and the remaining propeller block is hand faired to a smooth curve. With a ten foot wooden propeller, Parck [20, p. 233] states that seven templates "... are sufficient to give a uniform contour within very close limits ..."

In constructing the propeller I decided to dispense with the templates and form the profile guided by a single layout of the profile. It was my feeling that the profile inaccuracies introduced would not noticeably impair the performance of the propeller. Obviously, using a template would allow greater accuracy of form at the expense of time and effort. I leave it to others to investigate the effect of form accuracy in a propeller for this application.

For my propeller design, the maximum thickness at each radial station occurred at about one-third of the chord as measured from the leading edge. Using some calipers set at the appropriate measurement, I formed the block to achieve that thickness at each radial station. The final profile of the propeller is sandpapered to shape.

Once the correct angle of the blade at each radial station, the correct chord length at each radial station, and the correct maximum thickness at the proper point along the chord have been accurately established by the method I have outlined, what I consider to be the three other important form features are:

1. a well-rounded leading edge;
2. a smooth contour from the maximum thickness to a sharp trailing edge; and
3. a flat bottom chord, particularly near the trailing edge.

My original intention was to use the wooden prototype propeller to power the boat, and then to investigate casting reproductions of the propeller in the future. I decided it would be a little risky to use this wooden prototype on the boat, for fear of damaging my only propeller, so I decided to cast some replicas and save the original as a pattern. To achieve a smooth finish on the propeller I sprayed it with several coats of lacquer and finished with two coats of wax. I made a box for the propeller with inside dimensions slightly larger than the outside of the propeller. After placing the propeller in the box, the remaining cavity was filled with a self-vulcanizing silicon molding compound (Castomold SR-B from the Castolite Company). After the mold had set overnight, I easily removed the propeller from a slit made along the trailing edge. A reproduction of the propeller was then cast in the mold using a 50/50 mixture by weight of aluminum powder and polyester resin (Castoglas SLR from the Castolite Company). The casting compound was recommended for structural applications in a catalog of the Castolite Company (see Appendix VI). Copper powder was also available as a filler, but at twice the cost and no obvious advantage that I could see. The compound seemed to work very well - it poured easily into the mold minimizing any porosity and surface cavities in the casting, and the aluminum filling allowed the hub to be easily machined to mount on the propeller shaft.

APPENDIX VI: LIST OF SUPPLIERS

This list should not be taken as any sort of recommendation by me, but only as a possible source of the supplies mentioned.

Boatex Fiberglass Company, Inc.
Box 156
Natick, MA 01760
(617) 655-2000 (fiberglass supplies)

Dixon Bros. Woodworking
72 Northampton Street
Boston, MA 02118
(617) 445-9884 (birch for propeller)

The Castolite Company
P.O. Box 391
Woodstock, Illinois 60098
(815) 338-4670 (silicon for propeller mold, casting resin and aluminum powder)

Chandler & Farquhar Co.
900 Commonwealth Avenue
Boston, MA 02215
(617) 566-7800 (distributors of Boston Gear bearing products)

Atlantic Tracy, Inc.
1 Powers Street
Lawrence, MA 01843
(617) 685-8333 (distributors for Dodge timing belt and pulleys)

Edgcomb Steel of New England, Inc.
West Hollis Street
Nashua, NH 03060 (steel tubing for frame)

LIST OF REFERENCES

[1] Brewster, Brad, "The Design and Development of a Man-Powered Hydrofoil," B.S.M.E. Thesis, M.I.T., Cambridge, Massachusetts, May 1979.

[2] Lindsey, Joshua, "Design and Development of a Human-Powered Hydrofoil," B.S.M.E. Thesis, M.I.T., Cambridge, Massachusetts, May 1979.

[3] Gillardi, John, "Design and Construction of a Humam-Powered Water Vehicle," B.S.M.E. Thesis, M.I.T., Cambridge, Massachusetts, February 1982.

[4] Cohen, Barney, "Future Bikes," The New York Times Magazine, August 19, 1980.

[5] Whitt, Frank Rowland and Wilson, David Gordon, Bicycling Science: Ergonomics and Mechanics, Cambridge, Massachusetts: M.I.T. Press, 1974.

[6] Wilson, David Gordon, "Optimum Pedalling Position," private communication.

[7] Comstock, John P., ed., Principles of Naval Architecture, New York: Society of Naval Architects and Marine Engineers, 1976.

[8] Mandel, Philip, Water, Air and Interface Vehicles, Cambridge, Massachusetts: M.I.T. Press, 1969.

[9] Wellicome, J.F., "Report on Resistance Experiments Carried Out on Three Racing Shells," National Physical Laboratory, U.K. Ship Division. Report Ship T.M. 184, July 1967.

[10] Larrabee, E., "The Screw Propeller," Scientific America, July 1980, Vol. 243, No. 1.

[11] Abbott, Ira H. and Von Doenhoff, Albert E., Theory of Wing Sections, New York: Dover Publications, 1959.

[12] Glauert, H., The Elements of Aerofoil and Airscrew Theory, New York: Macmillan Company, 1943.

[13] Larrabee, E., "Design of Propeller for Motorsoarers," Science and Technology of Low-Speed and Motorless Flight, NASA Conference Publications 2085, Part I, 1979.

[14] Glauert, Herman, "Airplane Propeller," Aerodynamic Theory, Vol. IV, William Fredrick Durand, ed., Berlin: Julius Springer, 1935.

[15] Cox, H.L., The Design of Structures of Least Weight, New York: Pergamon Press, 1965.

[16] Parker, E.W., Braced Framework, Oxford: Pergamon Press, 1974.

[17] Heyman, Jacques, Beams and Framed Structures, New York: Macmillan Company, 1964.

[18] Kinney, Sterling J., Indeterminate Structural Analysis, Reading, Massachusetts: Addison-Wesley Publishing, 1957.

[19] Weick, Fred E., Aircraft Propeller Design, New York: McGraw-Hill Book Comapny, 1930.

[20] Park Whyrill E., A Treatise on Airscrews, New York: E.P. Dutton, 1921.

BIBLIOGRAPHY

Aronson, Robert B., "Muscle-Powered "Cars" Set New Speed Records," Machine Design, August 21, 1980.

Brosins, Jack and Dave Leroy, Building and Repariing Canoes and Kayaks, Chicago: Contemporary Books, 1978.

Byde, Alan, Canoe Design and Construction, New York: John de Graff, 1973.

French, Thomas E. and Charles J. Vierck, Engineering Drawing and Graphic Technology, New York: McGraw-Hill, 1972.

Geary, Don, The Wilder's Bible, Blue Ridge Summit, Pennsylvania: TAB Books, 1981.

Giachino, J.W. et al, Welding Skills and Practices, Chicago: American Technical Society, 1971.

Larrabee, E., and Susan E. French, "A Computationally Simple Lifting Line Analysis of Propellers and Windmills for Design and Performance Predictions," personal communication, 1980.

Lord, Lindsay, Naval Architecture of Planing Hulls, New York: Cornell Maritime Press, 1946.

Reissner, Hans, Theory of Propellers, Providence: Brown University, 1942.

Scott, Robert J., Fiberglass Boat Design and Construction, Tuckahoe, New York: John de Graff, 1973.

Sharp, Archibald, Bicycles and Tricycles, Cambridge, Massachusetts: M.I.T. Press, 1977.