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We are indebted to the authors and Marti Daily whose dedicated help made this issue possible. Pat Poole

Letters to the editor

Exotic Wheel Design

I was really "grabbed" by Allan Klumpp's article on designing exotic driving wheels for bicycles. I thought "This all makes total sense!" I immediately rushed to my PC and designed a variation on his Figure 6 wheel for my existing rear rim and old Phil Wood high-low hub. After obtaining the necessary spokes, I immediately took my wheels apart, drilled the rear hub, and built the old pieces into a new pair. (While I was at it, I decided to build a radial front wheel, since I could discern no justification other than inertia of thought processes for its four-cross spoking.)

Some comments:

1. While Allan's design techniques are impressive, I submit that the whole process is a lot faster and clearer with CAD. For those who have PCs, but no CAD, I recommend EasyCAD2,' by Evolution Computing in Tempe, Arizona. I purchased mine from a mail-order house for $109.00, and am mightily impressed by its speed and power on my old 80286 AT clone with math co-processor. The drawings it puts out on a 9-pin dot matrix printer are of professional quality, though they do print slowly if complicated.

2. When I printed out my Figure 6 wheel design, I thought "That's ugly!" By placing the radial left-flange spoke between the parallel right-flange spokes, the design became 12 sets of three parallel spokes, which I think looks great (see drawing below).

3. It may either be a typo or my imperfect understanding of Allan's design techniques, but I figure the wrap angle of the conventional 36 spoke cross 4 wheel to be 90º rather than the 80º stated in the article, since the two opposite spokes are in diametrically opposite holes. I find it easier to think in terms of the angle of the spoke to a radius drawn to the spoke head. It might be easier still to think in terms of angular deviation from tangent.

4. I noticed on CAD that I had to be careful to place the "space" between adjacent inside- and outside-flange spokes crossing at 22-1/2º, to prevent one spoke from crossing directly over the head of the other. This is easy to find and correct with CAD, but a real pain when encountered at the bench.

5. I had always put the "driving" spokes on the inside of the right flange, reasoning that if Joe or Jane Klutz dumped the chain between the sprocket and the hub, the "trailing" spokes would sustain the damage. Not having done this recently (thanks more to Suntour then to my shifting expertise), I yielded to Allan's logic and put the drivers on the outside of the new wheel. I hope I don't find it too hard to ride with my fingers crossed!

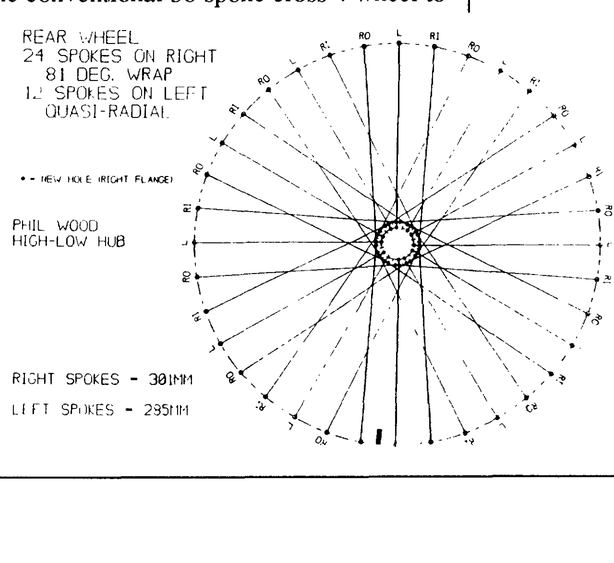
6. The wheel was easy to build and to true, as advertised.

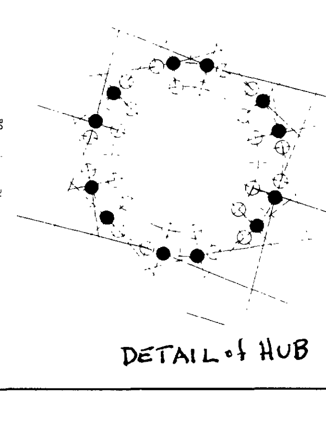
7. One caveat. It wasn't until the wheels were all built and installed that I recalled that my custom frame is <u>not</u> symmetrical, but that the rear stays are offset to the right to permit reduced wheel dish. I'll ride the wheel first, then consider re-dishing it.

Thanks to *Human Power* and to Allan Klumpp for a really exciting, inexpensive, practical project. I can hardly wait till morning to see how it rides!

Don Retierman
2474 Thata Way, Hemet, CA 92544

(continued on page 8)





DETAIL of HUB

HUMAN POWER

The technical journal of the IHPVA

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TENNESSEE TECH'S "*TORPEDO II*" - OVERALL WINNER 3rd International Submarine Races

Design and Fabrication of the Torpedo II Human Powered Submarine

by

Leo Benetti-Longhini , Byron A. Pardue, Scott Turner

INTRODUCTION

The design and fabrication of a human powered submersible present a number of interesting challenges. The vehicle must operate in the demanding ocean environment and still make maximum use of the amount of power available from a human. A human powered submarine must be built for both maximum reliability and efficiency. These design concepts were the goals of the team from Tennessee Technological University. The first priority of the project was the submarine be designed and built by students at the school. The second goal was to keep the vehicle as simple as possible. The last priority was to ensure that the safety of the occupants came before everything else. The team was rewarded for meeting its design goals by receiving awards for the best use of composite materials by an academic institution and the overall grand prize. This paper will outline the methods used by the students to design and build the Tennessee Tech Torpedo II.

SUBMARINE HULL

The shape of the Torpedo II is a body of revolution (ie. circular cross-section) dictated by basic arc or line profiles. The clear nose is a 380mm (15 in) radius removable half-sphere of polycarbonate, the center section is a cylinder of 762mm (30 in) diameter, and the tapering tail section including the propeller spinner is formed from an arc of 3.2m (127.5 in) radius (see Figure 1). Overall length is 3.3m (11 ft). The non-laminar body shape is based primarily on Simplified Methods for Estimating Torpedo Drag by John D. Brooks and Thomas G. Lang of the Naval Undersea Warfare Center, Pasadena, California and Underwater Drag Reduction Through Optimum Shape by Bruce H. Carmichael of North American Rockwell Corp. , Anaheim, California.

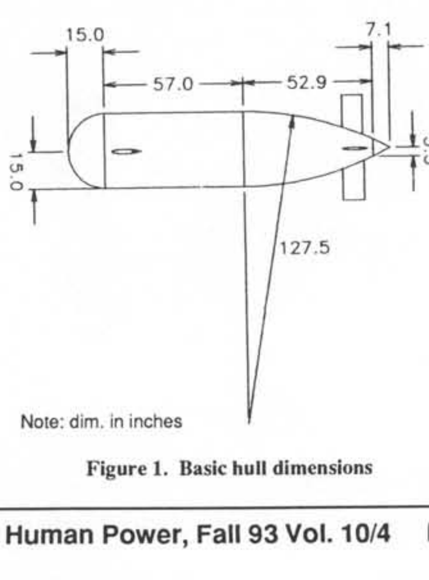

Figure 1. Basic hull dimensions

The structure of the hull is a foam-core sandwich of carbon fiber and fiberglass fabric in epoxy resin formed from a two part fiberglass mold. Vessel construction began with the building of an accurate plug or pattern incorporating not only the basic surface but all the hatch profiles, control-surface root fairings, and joint edges necessary for attachment of various sub-assemblies (such as the clear nose for example). A wooden frame is built to support a steel pipe of 63mm (2.5 in) diameter and 3.6m (12 ft) length on two pairs of roller bearings. This forms the basic structure of a crude but highly functional lathe. Wooden discs representing various cross-sections of the hull are attached concentrically to the pipe at locations dictated by the shape of the submarine. Arcs are used for the tapering tail section. The wooden structure is then given "volume" with two-part insulation spray foam (Insta-Foam). Plastic sheeting is used to wrap the wooden structure to keep the foam in place until cured. The foam, after rough cutting, is sanded with a belt sander held vertically in a fixture and guided on a track parallel to the pipe axis with the whole plug rotated by hand to provide the feed for the sander. The foam is then covered with a thin layer of auto body repair putty. The same sanding method is used on the body putty to provide circular passes of the sanding belt width. Control-surface root fairings are formed from wood profiles blended into the body putty surface. Hatch profiles and joint edges are cut by a router bit chucked in a supported hand drill. Cutting passes are made by either rotating the entire plug against the fixed router bit (concentric cuts) or by sliding the drill fixture against a fixed plug (linear cuts). All of the above methods are supplemented by significant amounts of elbow grease and hand sanding. Final surface finish is with auto body primer (see Figure 2). The overall method is labor intensive, converts about two thirds of the body putty to dust (filter masks are a necessity), and is only suitable for bodies of revolution. It is a surprisingly accurate and suitable method when labor is readily available.

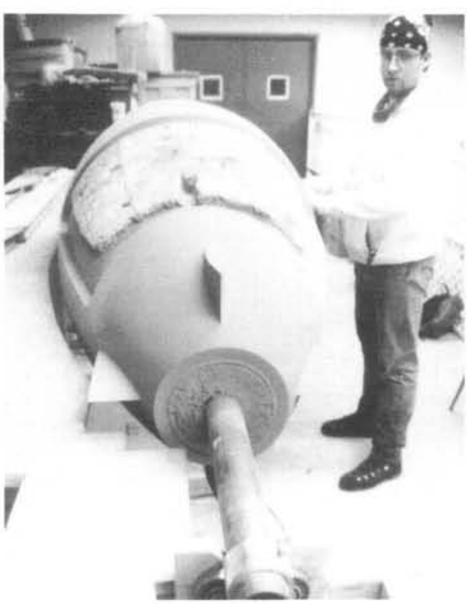

Figure 2. Finished plug viewed from rear

Circularity is checked with a dial indicator held perpendicularly to the surface while the plug is slowly rotated and surface waviness is checked by using the same dial indicator mounted perpendicularly to a small plate with nylon "feet" and then slid (feet touching) over the surface of the plug. The arc profiles are checked using templates. Each of these methods allows for the location of high and low spots which are sanded or built up, respectively.

The plug is prepared for mold making by first creating a parting wall at the mid-plane around the periphery of the plug. This defines the top and bottom halves of the mold (a necessity in removing the plug from the mold after creation of the mold). The wall is constructed of light plywood and the gap between the plug and the wall sealed with oil-based modeling clay. The entire plug surface is waxed several times with mold release wax and then sprayed with light coats of PVA (poly-vinyl-alcohol) which forms a surface that will not allow the epoxy resin, used in the mold, to adhere permanently to the plug. The first half of the mold is "laid-up" beginning with fine 3 oz. per square yard fiberglass fabric with successive layers (in our case, 10 layers) of heavier weight such as 6 and 8 oz. fabric. Once the first half is laid-up the entire plug is turned upside down and the wooden parting wall removed. The bare side of plug is waxed as is the fiberglass "wall" of the first mold half. PVA is sprayed onto these surfaces and the fiberglass lay-up process is repeated for the second half mold. Note that the wall of the first half mold defines the wall of the second half mold. Once the second half has cured the walls are drilled with alignment holes so that the two halves can be bolted back together accurately. The mold halves are removed from the plug with a bit of coaxing from wedges and rubber mallets and then washed to remove the water-soluble PVA film that has released from the wax on the plug. This is a brief description of the process used and is not detailed enough for actual use. It is recommended that sources such as Alexander Aeroplane, Wicks Aircraft, and Aircraft Spruce & Specialty be contacted for their catalogs which contain, besides product pricing, much useful information. These companies also list several books on composite construction which, although aimed primarily at composite aircraft, are applicable to virtually any vehicle.

The lay-up of the actual hull begins with preparation of the two-part "female" mold. After the mold is accurately bolted together its internal surface is waxed several times and coated with PVA. A layer of fine fiberglass cloth (4 oz. plain weave) is laid on the entire surface and wetted-out with epoxy resin. West System epoxy from Gougeon Brothers was chosen due to its high MEE (moisture exclusion effectiveness). An additional layer of fiberglass fabric (6 oz. crowfoot weave) is laid down as the second layer. The third layer is laid from 8 oz. carbon fiber plain weave fabric. After curing the thin vessel shell is removed from the mold with some coaxing mallets and wedges. It is significantly easier to remove a thin part from a mold than a thick one. The shell is then repositioned back into the mold to keep its shape correct and strips of curved closed cell foam (Dow blue foam) are bonded with a slurry of epoxy and micro-balloons (hollow glass spheres) into the hull. One-inch thick foam strips are used in the upper half of the hull and 6mm (1/4 in) strips are used in the lower half. The surface of the strips are sanded smooth and covered with a layer of 8 oz. plain weave carbon fiber fabric. A layer of plain weave 4 oz. fiberglass cloth is laid-up simultaneously with the coarser carbon fiber fabric to give a smooth finish to the interior or the hull. This sandwich of foam and composite fabric forms a very stiff structure and at the same time provides for the necessary buoyancy required for the stability of the vessel in water as discussed in the control system section.

HATCHES

The Torpedo II has three hatches: a main hatch, for entrance and egress from the submarine for both navigator and propulsor, a rear maintenance hatch to provide on-shore access to the drivetrain and emergency buoy system, and the transparent nose bubble which serves to facilitate the recovery of the submarine, by allowing the water inside the submarine to escape, and to provide a emergency exit from the submarine.

The main hatch provides a large polycarbonate window so the propulsor can see out and the support divers can see inside. It is hinged on both sides with retracting pins that use the dead and limit positions of a four-bar mechanism to provide a positive latch feel. Both sides of the hatch use the same latch mechanism. This allows one side to serve as a hinge while the opposing side is opened or vice versa. Alternatively both latches can be opened for removal of the hatch. The rear hatch incorporates the emergency buoy system in its' center. It is held in position with two aluminum tabs at the front end and two plastic, spring loaded finger latches at the rear. The front bubble is held in place by four plastic draw latches, and provides a nearly panoramic view for the navigator.

Both the main and rear hatch are constructed using the original female mold. The main and rear hatch are formed in two stages. First a form is made from the hatch sill of the hull mold using fiberglass cloth, epoxy, and colloidal silica. These forms are then placed in the bottom of the female hull mold and aligned to match the curvature of the top. This using of the hull mold to form the hatches eliminates the need for an entirely new hatch mold (see Figure 3). Construction of the actual hatches is virtually identical to the methods used in layup of the submarine hull, including the use of Dow blue foam as a sandwich core. The hatch construction differs slightly with the addition of carbon fiber tow (roving) to reinforce the inner perimeter of the hatches and the addition of the polycarbonate window to the main hatch. The polycarbonate is curved in a large cardboard tube heated by a blower. The hatch is placed in the hull mold for shape with the polycarbonate positioned over the window cutout. A layer of carbon fiber cloth and fiberglass are laid on top of the polycarbonate on the inside of the hatch. A vacuum bag is used to hold the window in place while the epoxy cures. Latches and associated hardware are then installed and the hatch is sanded to fit the hull properly. Nylon blocks are used as guides for the pins that hold the main hatch in place and stainless steel rods connect those pins to a handle at the center bottom edge of the hatch.

Figure 3. Lay-up of rear hatch

The front bubble is free molded from polycarbonate by a skylight company. Its' diameter approximately matches the diameter of the hull and the shape approximates a hemisphere. It is necessary to create a mounting ring on the inside of the bubble to provide alignment between the bubble and hull, to provide a mounting area for latch fittings, and to reduce flexing, aiding the removal and installation operations needed for recovery of the submarine. The rim of the bubble where the mounting ring is positioned is roughened by sanding (to aid bonding of the epoxy to the polycarbonate). Arcs of Dow blue foam are cut to match the hull thickness, beveled at the front edge, and bonded to the inside rim of the bubble using a slurry of epoxy and microballoons. The front of the hull was then waxed and sprayed with PVA and the bubble is carefully affixed, using tape, in the desired position. The intent is to use the hull front perimeter as a pattern to create the rim mounting profile. Low density filler is then applied into the gap between the foam and the front of the hull until all the voids are filled. A layer of carbon fiber and fiberglass cloth are laid over the foam and allowed to cure before the bubble is removed from the hull. Holes for the latch hardware screws are filled with epoxy mixed with colloidal silica and the latches are installed.

CONTROL SYSTEM

The submarine is steered by four control surfaces; two horizontal dive-planes in the nose and two vertical rudders in the tail (see Figure 4). The profiles of the control surfaces are symmetrical NACA 66-015 of 203mm (8 in) chord length and 305mm (12 in) span for the dive-planes and 188mm (7 in) chord length and 356mm (14 in) height for the rudders. Additionally, there are two horizontal stabilizers in the tail that are completely fixed. These are symmetrical NACA 0010-35 sections of 188mm (7 in) chord length and 356mm (14 in) span. The horizontal stabilizers, along with the rudders, also protect the rotating propeller blades from possible contact with obstacles such as ropes defining the race-course.

All four control surfaces are comprised of a fixed main wing and moveable trailing edge of one-quarter chord length that pivots to provide directional control. The moveable trailing edges are permitted a maximum deflection of 60° from center (120° total). Due to problems during the race the rudders were limited to approximately 40° deflection each way. This proved to be barely sufficient to keep the submarine in its track

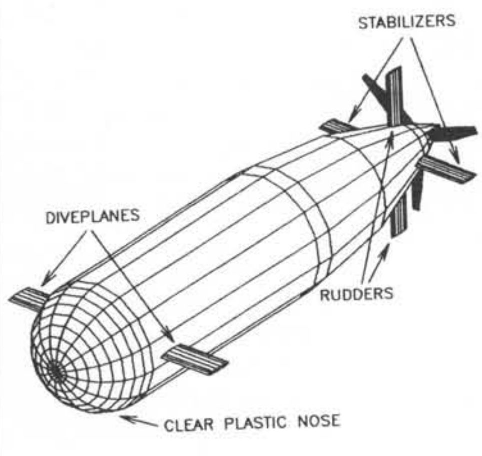

Figure 4. Location of control surfaces

during the turns of the race course. It is the opinion of the race pilot/navigator that full deflection of the rudders (60°) should be more than sufficient for the turn radii of the course. Additionally, the amount of dive/climb ability of the submarine appeared greater than necessary since a properly balanced (neutrally buoyant) submarine requires only the slightest amount of trim to keep it level during the race. In other words, the designing of the dive-planes to provide control forces similar to the forces of the rudders is not a requirement of the race course since there is virtually no diving and climbing as compared to turning. It is important to note that a neutrally buoyant vessel is critical if dive-plane size is to be reduced to a minimum.

The control system provides for pitch changes through the use of dive-planes and for yaw changes through the use of rudders. Roll control is not a necessity since the submarine is keeled with ballast weight. This, besides providing for buoyancy adjustment, lowers the center of gravity and provides for the stability necessary to counter-act the torque of the rotating propeller. Since the downward pull of gravity is opposed solely by buoyant forces (not wing lift as in an airplane) using ailerons for pure turning is impossible. Ailerons can, however, be used for roll control. The drawbacks to using ailerons for roll control are, firstly, that the vessel gains additional complexity and, secondly, that the ailerons are ineffective at slow speeds. The only part of the race where a vessel is "slow" is immediately out of the starting gate, precisely where the counter-torque from the accelerating propeller is likely to roll the submarine over. Since the speed is slow the ailerons cannot correct this roll. This unwanted roll will invariably allow for air bubbles to collect inside the vessel altering its buoyancy significantly and provide for a quick trip to the ocean surface.

A total of eight stainless-steel 51mm (2 in) stroke double acting cylinders are used in the control system of the Torpedo II. The system is comprised of four cylinders (masters) receiving input motions from the submarine pilot and four cylinders (slaves) producing output motion for the control surfaces (dive planes and rudders). Each slave is connected with flexible hosing containing a fluid back to its corresponding master cylinder. The flexible hosing forms a closed circuit for the fluid such that when a master cylinder is moved back and forth fluid is pumped to and from the slave cylinder causing the slave to move just like the master. The installation in the submarine is such that the four master cylinders are built into a joy-stick mechanism (see Figure 5) located in the nose of the vessel and operated by the submarine pilot. Each slave cylinder is connected to one of the four control surfaces. The movement of the joystick handle by the pilot thus activates the master cylinders which in turn drive the corresponding slave cylinders which activate the control surfaces, ultimately providing for the steering of the submarine.

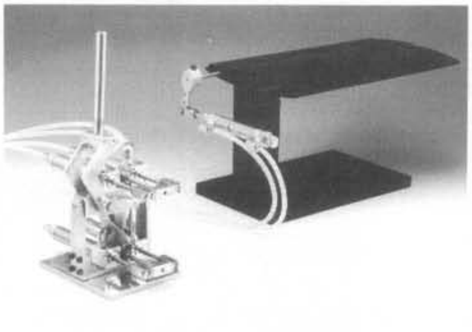

Figure 5. Joystick mechanism with dive plane

The hydraulic system works reasonably well. It must be remembered that the system was designed to be independent of vessel shape and to be easily installed in any type of submarine having only dive planes and rudders. It has fulfilled this objective, but it is not without a few flaws. Firstly, it is costly. Each cylinder costs twelve to fifteen dollars and a spare for each is needed. The spares are needed to change out cylinders with leaky seals, the second flaw. A small leak will cause a slave cylinder to drift out of phase from its master, limiting the maximum steering deflection. Changing out a single cylinder requires bleeding air bubbles from the nylon hoses, the third flaw. Although this becomes easier with practice it is not a job that anyone looks forward to doing. It is the opinion of those who worked intimately with the control system that a direct linkage system be designed specifically for a particular vessel shape.

One might imagine that a control system comprised of hydraulic cylinders would be very "dead" to the touch. In other words, very overdamped and providing no feedback to the manipulator of the controls. This is true if one considers "feel" through the pilot's hand alone. In fact, if the joystick is pushed to any maximum deflected position and released it will not return to center or neutral. There is, of course, always the visual reference to the course markers, but this does not provide any muscular feedback. There is, however, significant feel through the body of the pilot/navigator. This is primarily due to the extended position (laid out) of the pilot. A sharp turn is easily sensed by the body since the feet are significantly closer to the rudders than is the head (in this case, two meters). This could be taken as an added benefit of being in an extended position inside the vessel and not curled up in a fetal-like position.

EMERGENCY BUOY SYSTEM

The emergency buoy system, also called "dead-man" system, uses one single-acting stainless cylinder to hold a neon colored cylindrical buoy in a small "well" in the upper rear surface of the submarine. The function of the buoy is to immediately float to the surface on a tethered string to warn chase-boat based rescue divers when something is amiss.

The stainless cylinder is part of a charged circuit consisting of a small pressurized SCUBA tank, a flow regulator, a pressure regulator, and two hand-operated bleeder valves that remain open unless held in the closed position by the two occupants of the submarine. The flow and pressure regulators are connected directly to the SCUBA tank. Flexible nylon hose connects the bleeder valves and air cylinder to the two regulators. The system is charged with air under pressure which overcomes the force of the cylinder's internal return spring and extends the piston rod of the cylinder, thus keeping the buoy pinned in the "well" (see Figure 6). If a bleeder valve is not held closed (which would happen if one of the occupants became incapacitated) air leaks from the valve causing a drop in circuit pressure and allows the spring in the cylinder to push back the piston rod thus releasing the buoy.

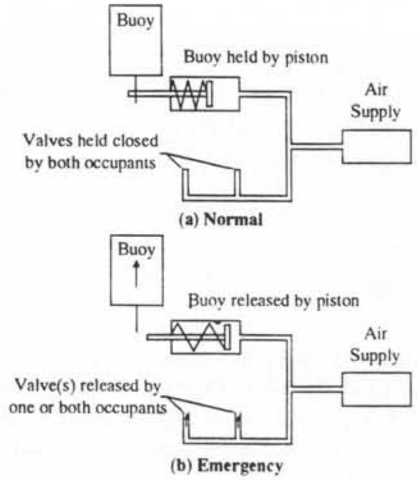

Figure 6. Circuit of emergency buoy system

The pressure regulator and flow regulator each fulfill distinct requirements in the setup of the system. The pressure regulator's only function is to create a pressure that opposes the spring force of the stainless cylinder. The pressure adjustment must be such that the spring is fully compressed and no higher. The flow regulator's function is to vary the recharge rate. The flow regulator must be adjusted such that the recharge rate is less than the bleed rate when one of the bleeder valves is open. If the recharge rate is higher than the bleed rate the pressure will not drop when a bleeder valve is open and the buoy will not release. It is obviously not important to have a high recharge flow rate since the system is inactive (does not need recharging) under normal use. Additionally, it is important to select a spring loaded cylinder with a high enough spring constant to overcome the friction of the buoy's "upward' pull on the side of the piston-rod. A small cylinder (small spring force) will not be able to retract its piston rod when the circuit pressure drops due to this friction. The system, which remains dormant until one of the occupants becomes incapacitated or needs assistance, has been totally reliable.

DRIVETRAIN AND PROPELLER

The drivetrain is designed to provide the most efficient and durable transmission of power to the propeller blades. Component choice is based on experience from previous submarine efforts and competitions. The Torpedo I used a very simple arrangement that proved to be nearly bulletproof during the 1991 International Submarine Race and the design was modified only slightly for the 1993 ISR by changing the drive ratio. By keeping the number of elements in the system to a minimum and reducing frictional losses both efficiency and reliability could be improved. The new design consists of a gearbox manufactured by Adantex originally manufactured as a reducer providing a two to one reduction ratio from a single input shaft to two output shafts at a right angle to the input. The Andantex gear box is used in reverse from the manufacturer's set-up (the output shafts were used as the input in the system to double the rotation of the drive shaft). The pedaler's rpm ranged from 70-90 once in motion resulting in a propeller rpm of 140 to 180. Crank arms machined from aluminum and threaded to accept bicycling type clipless pedals were attached to the input shaft of the gearbox. To reduce the width of the gearbox, the flanges are machined and the seals and bearings are moved inward. Additionally, the seals are reversed to prevent ocean water from entering the gearbox. Second stage oil, normally used in the lower stage of an outboard motor, is used inside the gearbox. The stainless steel drive shaft, is connected to the gearbox through a misalignment coupler. A plate bearing centered in the extreme stern of the submarine supports the opposite end of the shaft. The propeller mounting plate assembly, which allows for adjustment of the blade pitch angle, slips inside the hollow stainless shaft and is held in place with a shear bolt. The performance of the drivetrain has been flawless and its only shortcoming is the lack of pitch control while in motion (as opposed to on-shore adjustability).

The propeller blade profiles are based primarily on A Propeller Design Process for Human-Powered Submersibles by Patrick K. Poole of the Naval Systems Engineering Dept. of the U.S. Naval Academy in Annapolis, Maryland. The section used throughout is the NACA 63-412. Modified blade element theory is used to predict blade angle at various locations along the length of the blade. A computer program is utilized to optimize the profile selection from a preselected group. Blade construction utilizes full scale plots of sequential cross sections of a CAD generated blade model. The cross-sections, along with respective reference lines and location points, are transferred to thin plastic board and cut accurately on a scroll saw. These plastic cross-sections are positioned sequentially on two stainless steel rods to form a spine-like structure. The spaces between the blades are filled with auto body filler and sanded to match the plastic cross sections (see Figure 7). This forms a male pattern for the creation of a two part female mold with a non-linear parting line along the leading and trailing edges of the blade. Blades are fabricated in the mold from carbon fiber fabric and tow with an epoxy and silica based filler material for the core. A stainless steel rod is bonded to the core of the blade such that it protrudes from the blade root providing a shaft for the mounting plate assembly to clamp down on.

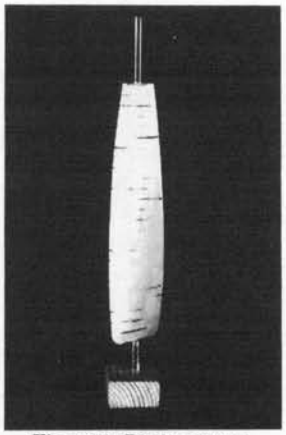

Figure 7. Blade pattern

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ed - Leo, Byron and Scott are students at Tennessee Technological University in Cookesville, TN. Their design, construction, and performance made their sub the OVERALL WINNER at the 3rd International Submarine Races

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Letters to the Editor *(from page 2)*

In the spring/summer HP, Allan Klumpp had an interesting article, "Designing Exotic Bicycle Wheels for Superior Strength". There are a couple of other ways spoke breakage could be reduced without much difficulty or adding weight.

I am wondering, why not bend both chainstays so that both rear dropouts are moved about 1cm. (3/8") to the right, then build the back wheel without dish? Bending the chainstays compensates for the lack of dish, so the tire stays on the bicycle centerline. Jim Leis of Santana says this would make a "strange-looking frame that would appear bent:, but this is less of a problem for people who spend more time riding their bikes than looking at them. The real disadvantage I see with this plan is that back wheels would no longer interchange with other bikes.

Which brings us to plan #2: Why not simply use stronger spokes on the right side of the wheel, and save the weight on the left side spokes? I've started building my back wheels this way with good results. Since the left-side spokes see about half the stress of right-side spokes, they would apparently need only half the sectional area; or in other words, assuming that all spokes are using the same strength materials, the spokes on the left side would appear to need only be 70% the diameter and half the weight of the right-side spokes.

The ultimate in aerodynamics and weight for a back wheel should be a tensioned disk wheel. The sides, made of a very thin material, made thinner toward the rim, would be glued precisely in position to the rim and hub, then put in tension by some way of spreading the hub flanges apart. It should be possible to make the rim and wheel sides/spokes lighter with the same strength.

In the same issue, Bruce Henry's "Letter to the Editor" calls the environmental problems of automobiles "over-emphasized". I disagree. In urban areas, where people live, up to 85% of the air pollution is due to motor vehicles. A lot of restrictions have been placed on smoking in the past couple of years, because the EPA announced it estimates about 3,000 americans a year die from the effects of second hand smoke. The same agency estimates that about 30,000 americans a year die from air pollution. Motorists have no right to poison my air.

In the article on the "Cheetah", it is claimed that the bike's Reynold's numbers are in the 4 million range, and that drag would be kept to a minimum by keeping the thickness-to-chord ratio under 15%. In the earlier days of the IHPVA, the Reynold's number of a record-setting HPV was figured to be around 10 million, and the fineness ratio for lowest air drag for a given internal volume was believed to be about 3-1/2 - 4:1, for a thickness/chord ratio of 25-29%. Would someone smarter than me please explain the discrepancy?

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PS–David Gordon Wilson has pointed out that building a dishless back wheel by moving the hub and freewheel to the right could cause chainline problems. I build rear-drive semi-supine recumbents, where the distance between cranks and rear axle is so great that this effect on chain angle is negligible. However, this is of more concern to people with old-fashioned upright bikes. Perhaps use a bottom bracket spindle for a triple with a double crankset? Use spacers? I have solved chainline problems on my own bikes by bending spider arms in or out with a big pair of vice-grips (not recommended) . . .

Charles Brown

Rim Temperatures

David Wilson's interesting study (Vol. 10, No. 3) to calculate rim temperatures due to braking leads him to warn Moulton users to be "especially careful" (on long descents).

When I presumed to launch back in the 1960's on the cycling world the radical departure from the conventional "Starley", I was fearful of all aspects of my use of small wheels for adult cycling for the first time in large volume production.

I was conscious of the interaction between heat input and dissipation as with speed, but was put off by the complexity of the exposed "fan effect" to attempt to calculate. In any case, I prefer the reality of structured experimental approach in innovation, especially in something as "transparent" as a bicycle. We used temperature sensitive paints on the rims of a Moulton and conventional 27" wheel lightweight, and did side-by-side braked descents on local hills, (Winsley and Widcombe in Bath). The conclusion was that while the small wheel heated up quicker – as it restores friction quicker on wet braking – the steady-state temperatures were of the same order within the limitations of our local hills.

One thing we did find was that tubular tyres were specially liable to bursting for whatever reason; and I never specified them since.

I remember asking early test riders touring in the Alps to take particular note of any troubles from braking. I do not recall any report of problems from that day to this on the millions of rider-miles experienced on Moultons.

But having read Dave's article I asked John Talbot a hard riding "Moultoneer" for 30 years, what his experience has been. As he frequently rides on his unfaired AM.7 with a friend who persists due to "image consciousness" in using a "conventional" lightweight road bike: the comparison is relevant. John says that he always out-drops his companion, and often observes 50 mph in normally seated crouch. Using the not-to-be recommended full crouch with his stomach on the saddle he has seen a terminal velocity of 56 mph on a 1 in 5 descent approximate in North Wales. Raising the head and trunk act as a powerful air brake at high speeds, available only on the normal riding position bikes when unfaired. Neither riders have had tire problems from overheating, and have not thought fit to compare rim temperatures.

My own view on normal riding position bikes either conventional or Moulton (unfaired) is that the issue is not a pressing one. Certainly we do use substantial rim tapes and one normally carries a spare tube, it being so small, rather than patching. But it is a thought to pursue fairing the rim section and using bladed spokes to lessen separation and to improve heat rejection.

It is on machines with high ballistic coefficients such as tandems and recumbents that the issue is real. I would certainly tend towards separating the braking from the rim, and what about an air brake or chute!

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I hope my ideas of a rim that will be cooler after long periods of hard braking will be of some uses to the bicycle industry.

I am proposing rims with twice the height on the sides. This moves the brake pad inside the tire bead and the metal has an area on both sides of the pad to dissipate the heat. The only thing that seems to limit the height of the flange area inside the bead is the need for access to the spoke nuts.

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A HUMAN-POWERED SUBMARINE DESIGN PROCESS

by

Bradley DeRoos, Foster Stulen, Tom Ramsey, Dave Carey, and Michael Neal

Abstract

The International Human Powered Submarine Races are held biennially to foster the advancement of underwater vehicle and subsystem technologies. The basic rules of the race are that the vehicle must contain two people (one pilot and one propulsor), the vehicles must be free-flooding, and all vehicles must adhere to a stringent set of safety rules both prior to and during the race. Many of the design considerations for human powered submarines also apply to other types of underwater vehicles. A remotely operated vehicle (ROV) and an autonomous undersea vehicle (AUV) can each benefit from the research and development efforts that occur as a result of these races. Optimization of the power transfer efficiency from the power source to the surrounding fluid is a key design consideration for underwater vehicles. For AUVs, which carry their power source on board in the form of batteries or fuel cells, increased efficiency can result in longer traverse distances, longer mission duration, or decreased vehicle size due to the ability to incorporate smaller power sources. For maximizing a human powered submarine's speed, the optimization of human power generation and transmission is vital. There are many factors that affect power transfer optimization including human factors, propeller design, and drive train design. The methods used in evaluating, integrating, and testing these factors are presented in this report.

Introduction

Battelle designed and built the Human Powered Submarine (HPS) Spirit of Columbus for the 1991 Human Powered Submarine Race competition. The Spirit of Columbus was propelled by an innovative high drag/low drag propulsion system which mimicked the swimming techniques of the duck or frog which present a high drag profile during the power portion of a stroke, and low drag profile during the return portion of a stroke.

The Battelle team decided to convert from this non-conventional propulsion system to a more conventional propeller drive in order to be more speed competitive. The composite hull of the Spirit of Columbus which had relatively small frontal and wetted surface areas and a hydrodynamically shaped profile was used for the 1993 effort. Other systems were to be revamped as the system design progressed. The vessel was rechristened as the Subjugator. The system design specifications are listed in Table 1.

Length	10.5 ft (3.20 meters)
Outer Diameter	28 inches (0.71 meters)
Wetted Surface Area	72 ft^2 (6.69 m^2)
Air Capacity	180 scf (5.09 m^3)
Hull	Form 58

Table 1. System Specifications

Technical Approach

The following tasks were laid out and subsequently performed as part of the design process:

- Drag testing was performed to obtain an experimental drag coefficient for the vehicle. Although theoretical values of the drag coefficient are available for this hull form (Form 58) based on either the frontal area or wetted surface area, design features such as control surfaces, tie-down points, a tow-line for the surface buoy (required for safety), and other small appendages degrade the accuracy of these values. Both the drag coefficient and shaft horsepower (power delivered to the propeller) were required for comparison to the propeller design code.

- Ergometer testing was performed to measure the propulsor's power generation ability as a function of time. The races consist of 100 meter time trials followed by 400 meter eliminations. The final elimination race to determine the overall winner is 800 meters in length. The values of sustainable power levels by the propulsor for the expected duration of the races were required for propeller design and optimization.

- Propeller design was performed utilizing a computer code based on propeller lifting line theory. This code was developed by Dr. Lee of The Ohio State University and requires boat velocity and propeller rpm as inputs.

- Dynamic analysis of the propulsion system was performed using the CADSI's DADS™ dynamic modeling program. Review of raw video footage from the 1991 race showed that instantaneous angular propeller velocities for the two submarines in the finals fluctuated by over 30 percent from their average. Fluctuations in angular velocity can significantly affect a propeller's overall efficiency, therefore one prime objective of this development effort was to explore means to smooth the power transfer between the propulsor and the propeller.

- Performance testing was conducted to validate the output of the computer model and ready the submarine for the Third International Race event.

Figure 1 illustrates the basic steps performed during development effort. The ergometer test results were used to refine the operating configuration up to one week before the time trials occurred.

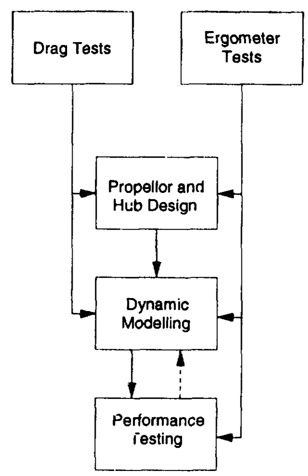


Figure 1. Development Flowpath

Drag Testing

Drag testing of the Form 58 hull was performed in a 25-yard (22.86 m) long swimming pool having a maximum depth of 10 feet (3.05 m). For these tests, an underwater puller was built. The puller was fabricated using a variable speed 2 horsepower (1492 watts) air-driven motor attached to an 18-inch (0.46 m) diameter drum (both located above the waterline). The towline was wrapped around the drum and passed down through a sheave located 5 feet (1.52 m) below the waterline. The towline was attached through a fairlead in the bow to a spring scale inside the boat. The spring scale was used to measure the drag force acting on the hull at various tow speeds. This test configuration allowed a rider located inside of the boat to read the spring scale while the boat was pulled through the water. Placing a person inside the submarine allowed the effects of air passing out the vent holes located on top of the submarine to be accounted for in the drag coefficient determination.

Tests were performed at tow speeds ranging from 2.9 ft/sec (0.88 m/sec) to 5.7 ft/sec (1.74 m/sec). Drag force was measured with the submarine travelling at terminal velocity in the deep end of the pool. Of the 18 test runs made, the lowest C_D obtained (on wetted surface area basis) was 0.009. The lowest value was accepted as the most accurate based on the assumption that the boat was "most" true for that run. This value was calculated based on a measured drag force of 17 lbs (75.6 Newtons), at a velocity of 5.12 ft/sec (1.56 m/sec). The wetted surface area for the hull (including control surface area) is 72 ft^2 (6.69 m^2). The drag coefficient was calculated using the following formula:

$$Drag = \frac{1}{2}\rho C_D S V^2 \qquad (1)$$

where Drag = Hull drag in lbs (Newtons)
ρ = Density in slugs/ft^3 (Kg/m^3)
C_D = Coefficient of Drag
V = Velocity in ft/sec (m/sec)
S = Wetted surface area in ft^2 (m^2)

The theoretical C_D for this hull form is 0.0036 assuming laminar flow conditions. A theoretical C_D based on turbulent flow assumptions was also calculated. It is reasonable to assume that turbulent flow exists given the operating environment at the race site (i.e., currents, wave action). The C_D calculation was made given a hull thickness-to-length ratio (D/L) of 0.22 where D is the maximum diameter of the body of revolution and L is the length of the boat. Using this method, the C_D was calculated to be 0.0063. A summary of the theoretical and measured C_D values for the HPS Subjugator are found in Table 2. This table shows that, as expected, actual C_D values are higher than those predicted by theory due to appendages, interference effects, etc.

Form 58 Theoretical, Laminar Flow	0.0036
Form 58 Theoretical, Turbulent Flow	0.0063
HPS Subjugator (Measured)	0.009

Table 2. Theoretical & Measured C_D values

Ergometer Testing

The amount of power that can be generated by a human is highly variable and depends on many factors. Factors include the size of the individual, their sex, physical condition, specific task training, heredity, nutrition and hydration, emotional and psychological state and duration of the task. All these factors impact the amount of power and total energy available from a given individual on a given day performing a given task.

The amount of available power affects the optimal propeller design and limits the speed of a human powered submarine. Therefore, it is critical to accurately measure, and improve through training, the amount of power available from a propulsor.

Battelle's approach to propulsion system design was to first measure the average amount of power that could be sustained by candidate propulsors for periods of time required to complete the 100 meter sprint race. This power was then used in a computer-aided design of the propeller which will be described in a following section. To measure power, an ergometer was needed to test the propulsors underwater in a position that approximated their position in the submarine. The ergometer was also to

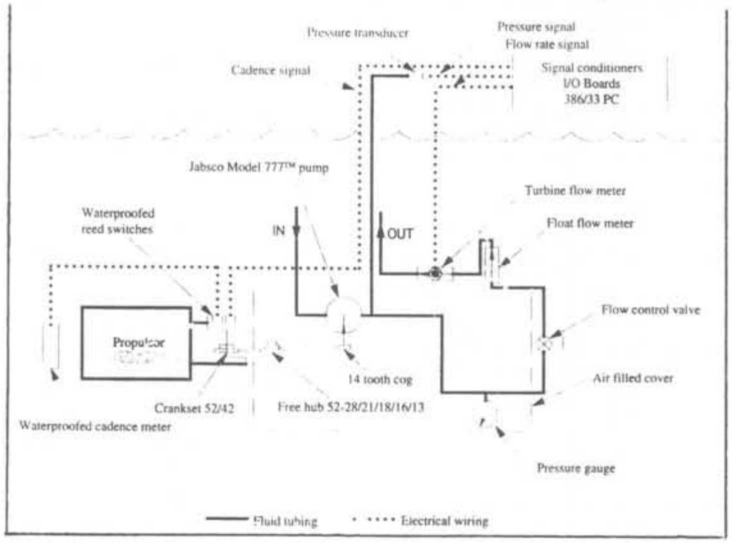


Figure 2a. Ergometer Layout

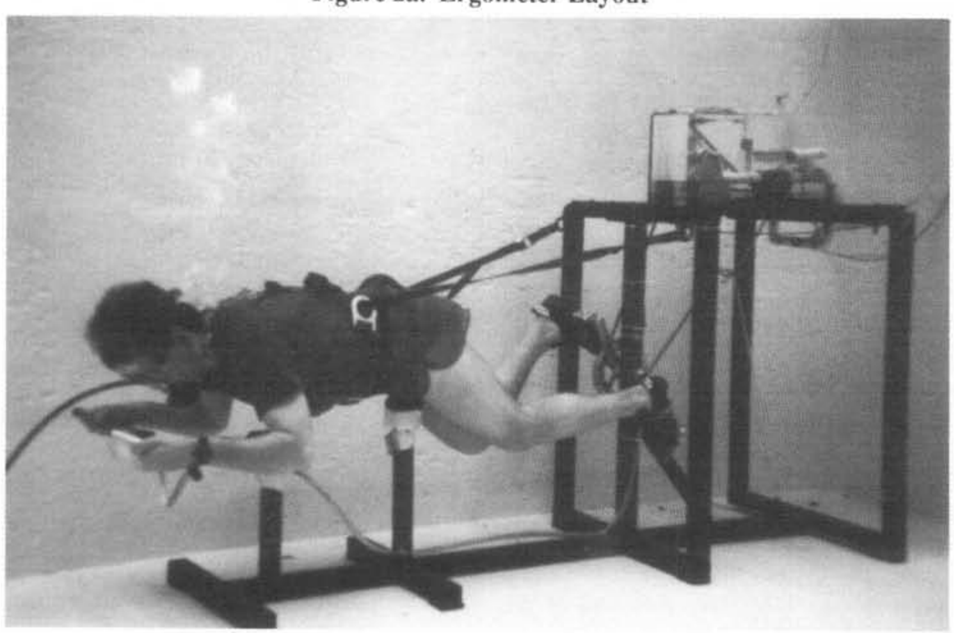


Figure 2b. Underwater Ergometer

serve as a training device to improve propulsor performance (in addition to conventional training exercises such as jogging, bicycling, climbing stairs, etc.) and to develop a sustainable horsepower versus duration curve. The principal of operation for the ergometer is that a propulsor works against a pump with the output power controlled by changing pump discharge pressure through the use of a valve. A schematic of the ergometer is shown in Figure 2a, and a picture of it on the bottom of Battelle's Research Pool is shown in Figure 2b.

The pump used at the "heart" of this system is a Jabsco Model 777™ pump which operates with known pressure and flow characteristics over the desired human power input range of approximately 0.13 to 0.85 horsepower (97 to 634 watts). The pump is driven through a two-stage speed-increaser that makes use of standard bicycle components. This configuration allows the propulsor to work at normal cadences for a cyclist and to drive the pump in its operating range. A cover is placed over most of the pump drive assembly. This cover is filled with air after the ergometer is submerged to decrease the energy wasted by the chain and fast spinning gears. This is especially important in this design, because the second chain is traveling considerably faster than the first chain that is driven by the cranks.

The force generated by the propulsor reacts against the frame of the ergometer. This is accomplished by two means. First, mountain bike handle bars are attached to the frame, which the propulsor uses during the test. Second, he wears a safety harness fitted with adjustable nylon straps that hook to the ergometer frame. The length of the strap is adjusted so that the propulsor is in a comfortable cycling position. Both means are also implemented in the submarine.

The optimal performance of a propeller system occurs at one advance ratio as will be discussed in following sections. This condition requires that the propulsor maintain a constant cadence. Cadence monitoring is accomplished by including a standard bicycle computer, which was waterproofed, on the ergometer. A second magnetic reed switch is installed for automated data collection. The cable for this reed switch is led up to poolside. A polyethylene tube connected between the pump and valve is led up to poolside where it is connected to a pressure gauge. A turbine flow meter is used to measure flow with its output sent poolside as well. The outputs of the pressure gauge, flow meter, and reed switch are connected to signal conditioners, which are in turn connected to an input/output card of a computer. The computer is used to continuously display pressure, flow, and cadence during the run and to record the data for subsequent analysis. A mechanical pressure gauge is installed between the pump and the valve, and a mechanical flow meter is installed downstream of the valve. Both are easily read by the support diver during the test, and serve as backups to the electronic instrumentation.

Typical test results are shown in Figure 3. The start and ending of a test for a "seasoned" cyclist and propulsor are shown in Figure 3a. The start and ending of a test of a fit male who is not a cyclist and had no experience on the ergometer are shown in Figures 3b. Both starts are characterized by an initial transient while the propulsor seeks the desired cadence. In addition to the fact that the trained propulsor can maintain the cadence longer than the untrained propulsor, there is a second difference. At the end of the tests the trained propulsor maintains a much smoother output than the untrained, which can be seen by comparing the curves in Figures 3a and 3b.

As discussed in the section Computer Modeling, it is important to maintain smooth cadence. A smooth cadence produces a greater velocity than a variable cadence even though the average horsepowers may be equal. This is because the smooth cadence will result in a higher average propeller efficiency, thus producing a higher average speed. It was found that higher cadences resulted in less fluctuation in output horsepower. This finding drove the design cadence to 80 rpm.

One objective of the ergometer effort was to develop a curve of sustainable power versus duration, which is shown in Figure 4. For comparison the same curve for a "first-class" athlete on a bicycle ergometer is shown. The curve in Figure 4 is a key result that is used to set the pitch of the propeller blades for the different lengths of the race. As an example, consider the 100 meter sprint. This race should last approximately one minute. Based on the curve in Figure 4, the propulsor can maintain 0.5 horsepower (373 watts) for that period. This is then used to calculate the optimal propeller blade pitch. A simulation is then run to predict the duration of the race from start to finish.

This new time can be used to refine the estimate of sustainable horsepower, propeller pitch, speeds, and time. The same procedure is used for the 400 meter and 800 meter distances. This allows the propeller to be "tuned" for the particular race using the unique adjustable-pitch propeller design described in the following section.

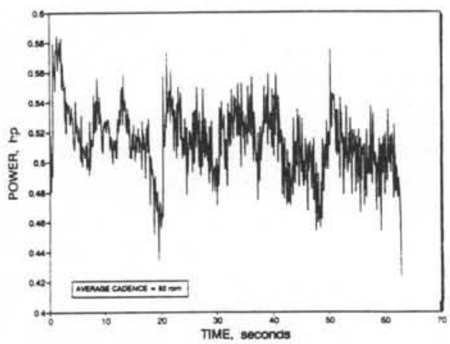


Figure 3a. Trained, Experienced Propulsor Horsepower Output

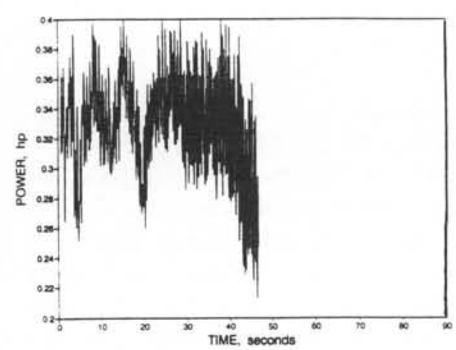


Figure 3b. Untrained, Inexperienced Propulsor Horsepower Output

Propeller Design and Fabrication

As previously stated, the entire drive train was redesigned in order to maximize power transfer efficiency which in turn maximizes speed. This makes efficient design of the propeller extremely critical.

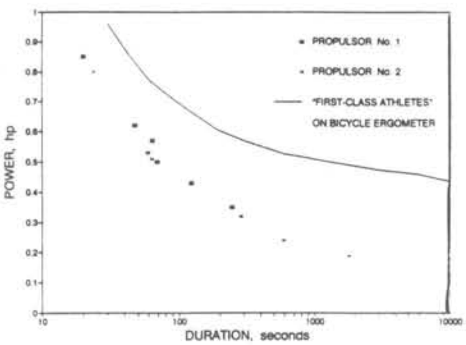


Figure 4. Sustainable Horsepower as a Function of Event Duration for Battelle's Propulsors

The analysis of propeller design was done with a computer code written by Dr. John Lee from The Ohio State University. This code was originally used to design the aircraft propellers for the Avanti Piaggio P-180 Turboprop Business Jet, and was converted to allow design in water, an incompressible fluid. All of the blades designed for this particular application were derived from the Clark-Y16 airfoil section.

An "aircraft" propeller was chosen due to inherently high efficiencies and non-cavitating tendencies. The obvious choice for the optimal performance is a variable pitch propeller, but the completely variable pitch concept was rejected due to mechanical complexities that are not justifiable given the predictable event durations and human power capabilities. To allow for design optimization, a ground-adjustable propeller was chosen. This allows the blades to be set at an angle before the race which will give maximum performance for the power level delivered over the duration of the race.

The main inputs into the propeller design code were submarine velocity and propeller rpm. The design objective was to maximize thrust while maintaining constant power. This was accomplished through varying blade twist and the addition of camber to some of the blade sections. In general, the hub and tip sections are symmetrical and most of the thrust is produced by the outer half of the blade due to the higher relative fluid velocities. The hub sections are symmetric for streamlining and little thrust is produced due to the low fluid velocity near hub. The tip sections are kept symmetric in order to reduce tip vortices. Special attention was paid to the blade section lift, drag, and moment coefficients in order to eliminate or minimize cavitation. One of the primary drivers for the blade design was the cadence of the propulsor. Eighty rpm was selected as an optimum propulsor cadence based on the ergometer testing performed prior to the propeller design. A propeller rpm of 120 (a 1.5:1 gearing ratio) was chosen to increase inertial effects while maintaining high efficiency.

The most cost-effective method for blade fabrication was determined to be the lost wax process. The coordinate file of the designed blade was imported into CATIA™, a solid modeling and CNC milling package. Two halves of the mold were built up as solid models and CNC milled from aluminum. The aluminum mold was used to make wax blades which were replicated in stainless steel. Figure 5 shows the completed propeller assembly. This method of fabrication yielded highly uniform blades at a cost of $80 per blade (casting cost only).

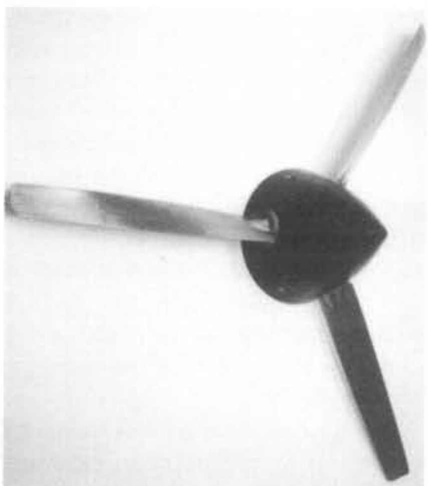


Figure 5. Completed Propeller Assembly

Computer Modeling

A dynamic simulation of the submarine was conducted using a commercial multi-body dynamics software package (CADSI's DADS™) in order to predict the performance of the submarine under various conditions and to determine the effect of unsteady propulsor pedaling. The purpose of the simulation was to study straight-line running performance. Because of this, the dynamics associated with the control surfaces was not modeled. The separate rigid bodies of the simulation model are the hull, which is free to translate and roll on a cylindrical joint that is connected to ground, the crank, which is connected to the hull by a revolute joint;

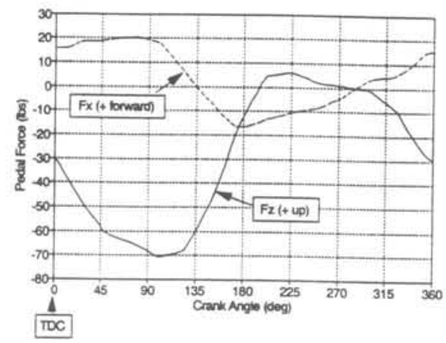


Figure 6 Typical Pedaling Forces versus Crank Angle Curves

an upper transmission shaft, which is connected to the hull by a revolute joint and driven by the crank; and the propeller, which is also connected to the hull by a revolute joint and driven by the upper transmission shaft through a rotary spring. Rolling of the hull (about its longitudinal axis) is allowed so that the response of the submarine to off-center buoyancy forces that are used to counteract the propeller torque can be simulated. The rotary spring between the upper transmission shaft and the propeller was used to simulate a "loose" connection in efforts to reduce the effect of unsteady propulsor pedaling.

The forces, which drive the model, consist of propulsor pedaling torque, the longitudinal force on the hull, and the drag torque and thrust generated by the propeller. Losses in the transmission were neglected since the input torques used were derived from the ergometer tests which measured horsepower at a point "downstream" of the transmission system. The pedaling torque was computed from measured pedaling forces for a typical bicycle rider on rollers. The curves used for this forcing function are shown in Figure 6. The curve of pedaling torque versus crank angle was normalized to one and used in an algorithm that allows the horsepower exerted by the propulsor to be varied.

The drag force on the hull was computed according to Equation (1). The thrust generated by the propeller and the associated drag torque can be calculated using thrust coefficient (K_t) and torque coefficient (K_q) curves for a particular propeller blade angle as a function of the advance ratio. For this analysis, the curves for various blade angles ranging between 25° and 45° were obtained from the propeller design code. The relationships between efficiency, thrust, drag torque, hull speed, and propeller speed are defined by the equation given below:

$$\textit{Propeller Torque} = K_q(J)\rho n^2 d^5 \qquad (2)$$

$$\textit{Thrust} = K_t(J)\rho n^2 d^4 \qquad (3)$$

$$\eta_p = \frac{J}{2\pi}\frac{K_t(J)}{K_q(J)} \qquad (4)$$

$$\textit{Advance ratio, } J = \frac{V_s}{nd} \qquad (5)$$

where:
V_s = submarine speed
ρ = density of water
n = propeller speed
d = propeller diameter
ηp = propeller efficiency

Figure 7 is a sample plot of K_t, K_q, and propeller efficiency versus advance ratio at a blade angle of 32° (measured at a point 10.5 inches (0.27 m) from the axis of rotation). This was the design point of the propeller at 0.5 hp (373 watts) and 120 rpm. Of particular importance to note on this graph are the ± 22 percent efficiencies marked. The ± 22 percent points relate to the maximum fluctuation in the advance ratio (J) predicted by the computer model as a result of time-varying input torque. The efficiencies associated with minimum and maximum advance ratios at ± 22 percent rpm fluctuation are 81 percent and 82 percent, respectively (with an average efficiency of approximately 85 percent over that range of advance ratios). If the rpm fluctuation varies by ± 33 percent as seen in previous race tape videos, the average efficiency drops to less than 80 percent.

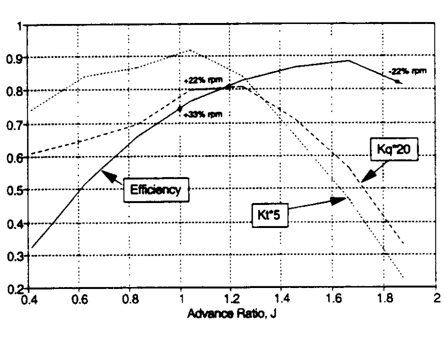


Figure 7 Propeller Characteristic Curves

The computer simulation described above was used to evaluate several concepts to reduce the effects of unsteady pedaling. One concept was a "loose" rotary spring between the crank and the prop in order to deliver a constant propeller torque, using the spring to alternately release and store cyclic variations in the propulsor's pedaling. Although the spring does effectively reduce the torque variations, it was realized that in addition to providing a constant torque to the propeller, the spring also presents a constant load torque to the propulsor. Relative to the nominal design (drive shaft rigidly connected to the propeller) the propulsor would experience less resistance in parts of the stroke where he can deliver maximum torque, and more resistance at top and bottom dead center, where the propulsor's torque capability is at a minimum. The overall result would be a very unnatural pedaling feel and the possibility of being "backdriven" by spring wind-up at top and bottom dead center crank positions. Another concept was a flywheel attached to the crank. Although the simulation showed good results, this solution was deemed impractical because of the large size of the flywheel necessary (assuming special gearing was not used to increase the flywheel spin rate) and the limited space available inside the hull for the addition of flotation that would have been required to counteract the weight of the flywheel mechanism.

Dynamic simulation runs were made at 0.5 and 0.35 hp (373 and 261 watts) with both time varying torques and constant torques being used as inputs to the model. Figure 8 shows speed versus time for the 0.5 and 0.35 hp (373 and 261 watts) computer simulations. The 100 meter lines indicate the times that the submarine would cross the finish line for each case. The effect on speed of having a time-varying torque is a reduction in steady-state speed of approximately 0.15 knots (0.08 m/sec) for the 0.5 hp (373 watts) input case.

System Performance Testing

Performance testing was performed at The Ohio State University's 50 meter swimming pool. This facility was used for three test and evaluation periods prior to open water testing. During a pool test period, data was collected to check the validity of the computer model that was developed. Figure 9 is a speed versus time plot that shows the model prediction at 0.5 hp and the actual results from two of the speed runs. This figure shows that the submarine accelerates faster than the model predicts, with the terminal velocity being very close to model prediction. The higher acceleration is likely due to higher power levels being delivered by the propulsor during the start-up period. The propulsors were attempting to maintain 80 rpm during the whole run, which requires greater power when the submarine is at a standstill than when the submarine is travelling at terminal velocity. Subsequent testing in a 100 meter facility showed a maximum sustainable velocity of 3.8 knots (m/sec) for that duration event.

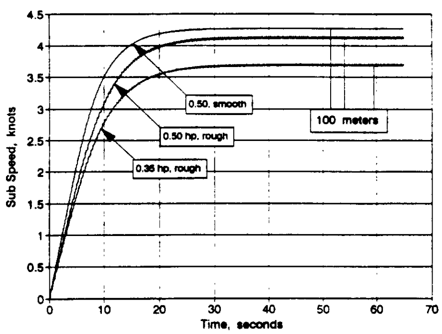


Figure 8. 0.5 hp and 0.35 hp Time versus Speed Computer Simulations

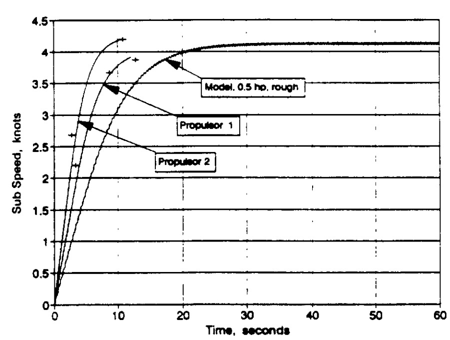


Figure 9. Time versus Speed Plots for Two Propulsors During Pool Testing

Figure 10 shows a plot of propeller angular velocity as a function of time for one of the test runs. Propeller rpm information was derived from an underwater video recording. The curve was generated by frame stepping through several complete propeller cycles (1/30 second between each frame), and calculating the angular velocity for each time step. The curves represent the best fit through the data points collected. This analysis showed that Battelle's design goal of minimizing propeller angular velocity fluctuation was met. The rpm fluctuation is +/- 18 percent as compared to the computer simulation prediction of +/- 22 percent. The instantaneous efficiency curve is also shown at the bottom of Figure 10.

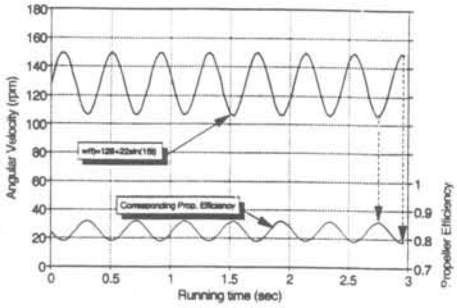


Figure 10. Instantaneous Propeller Angular Velocity and Efficiency as a Function of Time

Race Results

Battelle's submarine finished third in speed of the 43 participants in the Third International Submarine Race. The average speed over the 400 meter course was 2.9 knots (1.48 m/sec). This is approximately 0.7 knots (0.36 m/sec) slower than predicted by the model. This difference can be partially attributed to increased drag caused by control surface motion (which is not accounted for in the model), and partially due to the conservative "line" run by the pilot to prevent fouling on the race course buoys.

Conclusions

The design objective of optimizing human power vehicle performance in an aqueous environment was pursued rigorously. The submarine configuration for the races is shown in Figure 11. Tasks in this effort included: design of an underwater ergometer, propulsor testing on the ergometer, computer-aided propeller design, CNC machining of molds for propeller fabrication, computer simulation, and the design of an efficient gearbox for power transmission.

Predicting submarine performance by using a computer model that requires input information on vehicle drag, human performance, and propeller operating characteristics has yielded fairly accurate results. The computer model closely predicted terminal velocity and propeller rpm, and thus validated both its accuracy and the accuracy of the inputs including horsepower, drag coefficient and propeller performance. The computer model is now a tool that can be used to explore alternative designs and evaluate changes such as horsepower input or drag variations. This will be extremely beneficial for future races where improved methods of drag reduction, more efficient drive mechanisms and propeller designs, or increases in horsepower may become available.

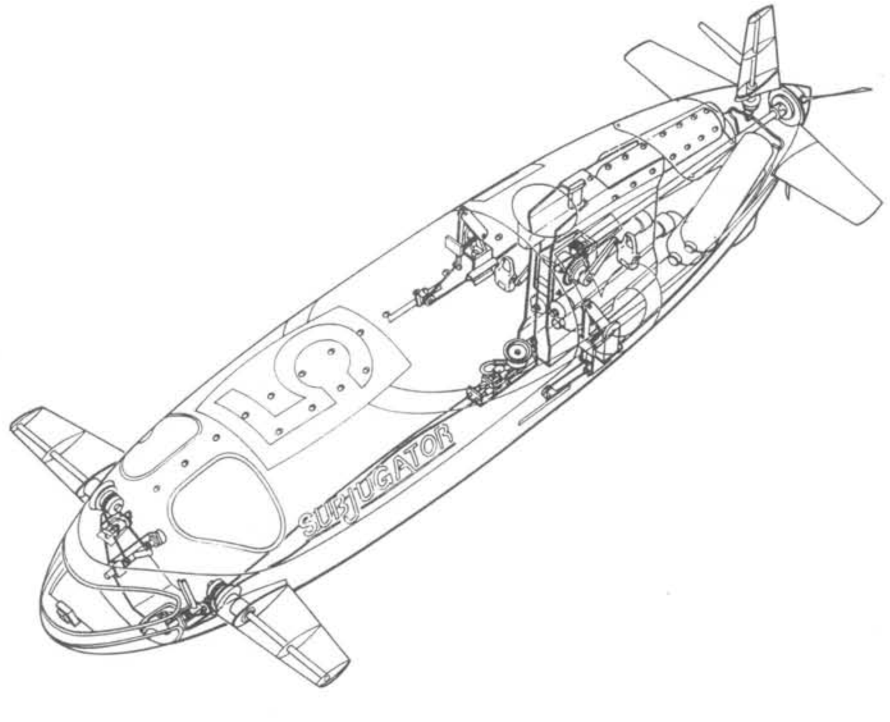
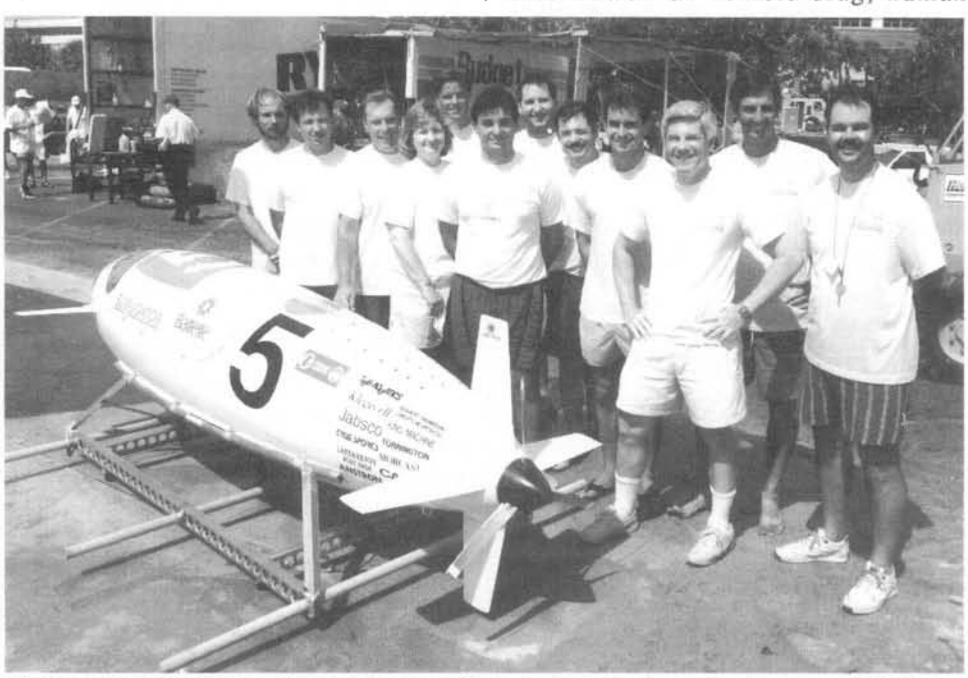


Figure 11. Human Powered Submarine SUBJUGATOR



"SUBJUGATOR" **and Battelle Team** (*Photo by Ray Graham, Photogroup of WPB*)

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The VICTORY Human Powered Submarine

by
Ray Scholl, Winston Churchill H.S.
Ed Leibolt , DTRC Submarine Club

The VICTORY human powered free-flooding submarine was built for as an entry in the 3rd International Submarine Races held at Ft. Lauderdale, Florida in June 1993 (See Figure 1). The VICTORY submarine was designed by the students of Winston Churchill High School of Potomac, Md. under the mentorship of the David Taylor Research Center Human Powered Submarine Club of Bethesda, Md. The design of this entry is based on the past experience with the TURTLE and TURTLE II submarines from the 1st and 2nd races in 1989 and 1991, respectively.

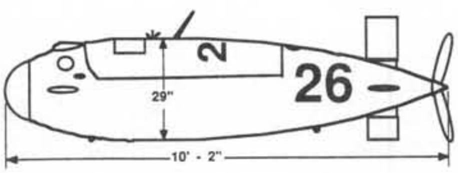

Figure 1. Side View of VICTORY

The design and construction of VICTORY took place over a two year time period. The student design team developed an overall design philosophy of emphasizing innovation, affordability, and style. Target design goals set by the design team included a 5 knot design speed, \$5000 maximum cost, 150# maximum weight, and a 4 person launch/recovery. The design of the hull and control surfaces were based on optimization studies of approximately 25 different hull forms and 15 different control surface geometries utilizing EXCEL© spreadsheet (developed by the team) and RUNSUB© submarine design (developed by M.I.T.) computer programs.

The hull design was developed by first setting the maximum hull diameter to enclosing two stacked crew members and two dive tanks. Next, the power (250 watts)(0.3hp), design speed(2.5 m/s)(5 kts) and maximum diameter were input in to the RUNSUB© program. Different hull lengths and shapes were tried until the optimized hull was found. The full scale hull offsets were then generated. A very innovative construction method was develop to construct the hull Closed cell hollow Styrofoam© rings were stacked to form the hull shape(See Figure 2). The advantage to this design is the elimination for the need of construction molds. The foam rings are glued together in quarter sections Epoxy resin/glass mat are laid over the outside of the sections. The sections were then pieced together and a second layer of epoxy/glass was added. The outside of the hull was then faired with automotive body putty. The hatch was cut and one layer of epoxy resin/glass mat was applied to the inner surface of the hull. This formed a foam core sandwich hull.

The propulsion system consisted of a linear drive mechanism connected to a two bladed propeller(See Figure 3). The advantage of this design was the use of linear leg motion to drive the submarine and there by reducing the amount of water movement around the propulsor's legs. Also the linear drive train reduces interior space requirements over a cyclic drive unit. The drive has two pedal operated trolleys on either side of an aluminum I-beam. The trolley wheels run on V-shaped rails that allow linear motion with minimal drag and restrain the trolley in the transverse direction. The trolley is connected to a bicycle chain that runs over fore and aft bicycle freewheels. The forward freewheel is free to spin in both

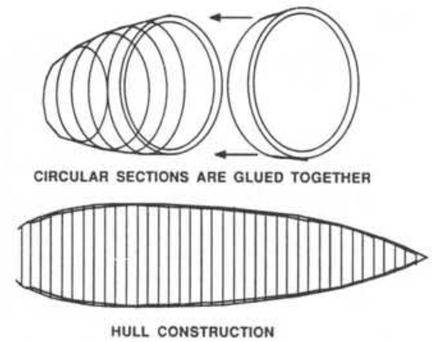

Figure 2. Foam Ring Mold/Hull

directions. The aft freewheel ratcheted in one direction and drives a bevel gear box in the other direction thereby transmitting power to the propeller shaft on the downward stroke. On the other side, the trolley moves forward and does not provide power to the bevel gear box. If the propulsor has enough training time, as ours did, then he/she can provide constant power to the propeller by starting the downward stroke on one side before the other side has completed the power stroke. This eliminates the "dead spot" typical of cyclic power units. The propeller design was developed from lifting line theory and was based on the TURTLE II hull shape. The team used the TURTLE II raked two-bladed propeller instead of their own design due to time, expertise and fiscal constraints.

The control surface designs were based on a NACA 0015 foil shape. The RUNSUB© program was used to optimize the size of the control surfaces. A one quarter scale radio controlled model of the

The Winston Churchill H.S "VICTORY" *(photo by John Chiffer)*

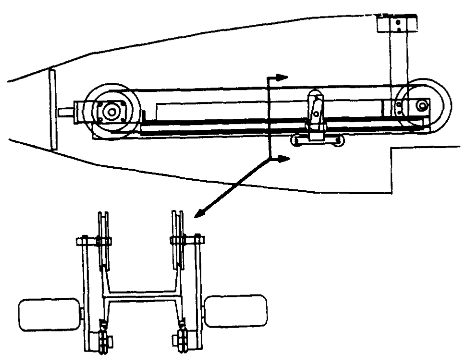

Figure 3. Linear Drive Unit

submarine was built and was used as a test platform for control system design. Various control surface sizes and placements were tested to optimize both the control of the submarine and drag reduction. The VICTORY control system consists of two mobile horizontal bow planes, two stationary stem planes and two vertical rudders. The bow planes main function is depth keeping. There was some vertical motion allowed, but only to counteract any errors in ballasting, The sub was trimmed and ballasted to be neutral before every run therefore not have the tendency to rise and sink. There was no active ballast system. The stationary planes were added after a stability problem was encountered during the testing stages. The top and bottom rudders serve to turn the sub in a turning radius necessary to navigate the race course. The pilot operated the sub via a C-section control yoke attached to the bow planes and a lever connected to the rudder by a flexible shaft.

The life support system relied on two

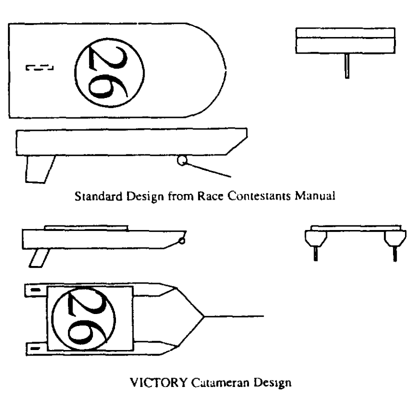

Figure 4. Catamaran Towed Buoy Design

80 cubic foot dive tanks manifolded together to provide the necessary air for the race. Past experience on TURTLE and TURTLE II showed that this was an adequate air supply provided the crew had enough training time at depth in the vehicle. Two SHERWOOD first stage regulators were attached to the manifold. Three second stage regulators were attached to the first stage regulators. Two second stage regulators(primary and emergency) were used for the pilot as a safety feature due to his extreme confinement. In addition to the life support system, each crew member had to wear a Spare Air™ pony bottle, Mae West style buoyancy compensator, and weight belt.

The safety devices for the sub include two bicycle brake lever deadman switches which were connected to a brake caliper. The brake caliper released a tethered safety buoy to the surface in an event an emergencies. The team designed a tethered catamaran towed buoy that tracks the submarine. This design significantly reduced the drag over the standard design used in previous races (See Figure 4). The hinged crew hatch latch consisted of a sliding bar that engaged the hatch in two locations. The bar could be operated internally and externally of the submarine

To allow the crew to navigate, an acrylic hemisphere was installed in the bow of the hull. This provide 180°of visibility. There were also two port holes on the sides to allow the propulsor to see out. The hatch view port was installed to allow the safety divers to see the crew.

The VICTORY's race performance was disappointing. Although it passed both the on-land and in-water safety inspections, it did not perform well on the actual race course. Given two speed trials to qualify, VICTORY failed to leave the starting gate. The first attempt resulted in a broken propeller shaft coupling which transferred power to the propeller. The second attempt was aborted because of a premature release of the safety buoy. The team did win the Judges Award for there efforts. Subsequent time trials were held at the Naval Surface Warfare Center, Carderock Division(formerly David Taylor Research Center) in a testing tank. A 60m (200 ft) straight line course was set up and VICTORY recorded a speed of 1.95 m/s (3.86 kts) for a running start. The sub was not equipped with the towed buoy for this trial and this time was not officially witnessed by any IHPVA officials. This proved to our team that VICTORY was a viable sub and definite future contender.

The team currently plans to enter the next race with a modified VICTORY submarine. These plans include modifying the power unit for more strength, modifying the deadman switches for more reliability, and fairing the hull for more drag reduction. A new propeller will be designed and built based on VICTORY's hull. There will also be an effort to find and train an alumni(at least eighteen years old) of the school to be the propulsor/pilot for the next race.

The team consisted of:

Students	DTRC Sub Club
Andrew Binstock	Ed Leibolt(propulsor)
Judd Borakove	Dan Dozier(pilot)
Mike Chang	Mary Leibolt
Jared Farber	Mike Dozier
Nathaniel Frink	John Ware
Peter Liu	Cheryl Ware
Anita Milman	Steve Mays
Raymond Scholl	Bruce Crock
Emery Shen	Kent Brady
Todd Sheridan	**Teachers**
David Weitzberg	Ed Dennis
Brian Wolfman	John McGoldrick

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ed- The "VICTORY" was one of a growing number of high school entries in the sub races.

The Spirit of Annapolis
Preparing for the Human-Powered Submarine Race

by
LCDR W. A. Davidson

As we began to plan for the Third International Human-Powered Submarine Competition we considered the Naval Academy's history with this event. Simplicity was the primary emphasis for the design and fabrication of SQUID, the Academy's first entry to the human-powered submarine competition in 1989. This philosophy paid off well as SQUID was named the overall winner. For the second competition innovation was key. The new submarine, SUBDUE, was driven by a paddle-wheel type propulsion system. Unfortunately the submarine had difficulty obtaining forward thrust and was disqualified. SQUID was also entered in the second competition, and after clocking the forth fastest time, was also disqualified because of problems with pulling the surface buoy underwater.

For the third competition we agreed to again embrace the "simple and reliable" philosophy. Our primary objective was simple; Don't get disqualified. Of course we also wanted to win again. Because this event is primarily a race, we believed the best way to win was to concentrate on speed. As each competition has shown, speed was best achieved with a bicycle or linear drive system turning a single propeller. This scheme also fit in with our "simple and reliable" philosophy so this is where we focused our attention.

A lack of funds and time required us to make use of one of our existing submarines, SQUID or SUBDUE. The continued success of SQUID made it a logical choice, however, it was often on display at various engineering events at the Naval Academy and elsewhere. Since it was not readily available for needed modifications and trial runs, it was decided to retire SQUID. This was not considered a set back however. The SUBDUE hull was smaller than SQUID's giving a 15% reduction in total wetted surface area and a 12% reduction in frontal area. Both reductions translate to increased speed as shown in Figure 1.

Because of the extensive modifications that would be made to the submarine, it was given a new name, THE SPIRIT OF ANNAPOLIS. A name the midshipmen felt would represent their school well.

While a new propulsion system was being designed for this hull, experiments were conducted to find out how much horsepower could be produced by a human underwater. Earlier tests conducted at the Naval Academy suggested that a prone position allowed a diver slightly more production capability than a sitting position. More important, a 20% savings in air consumption was obtained when a prone position was used. We began our experiments with the assumption that our propulsor would be in an inverted prone position, laying above the pilot. This was the configuration the hull was originally designed for. We found that the inverted position worked best because of the reduction in static lung loading.

To conduct the experiments an underwater ergometer was rigged to simulate pedaling conditions inside the hull, figure 2. Our first objective was to find an optimum cadence for the propulsor. On land, bicyclists will try to maintain about 100 rpm as this has been shown to be the most efficient cadence. The experiment was a simple one. Several midshipmen volunteers were placed underwater in the ergometer and told to pedal as fast as they could for 10 minutes while breathing off a scuba regulator. After several days of familiarization runs, data collection began. The horsepower production was determined from their sustained cadence for a given load. A wide spectrum of loads was applied resulting in a wide spectrum of pedaling speeds. Although the tests indicate a slight increase in horsepower production at 45-50 rpm, the results are inconclusive, see Figure 3. An interesting and unexpected result was that most of the divers pedaled at 45 or 50 rpm despite the loading. This may suggest a "comfortable" cadence zone

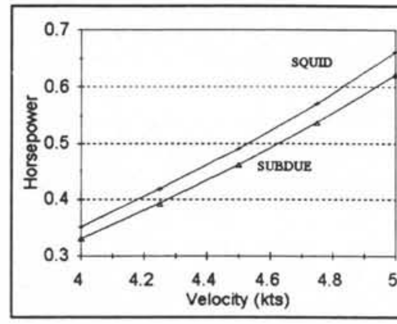


Figure 1. Shaft Horsepower vs. Speed for SQUID and SUBDUE Hulls. PC = 0.5

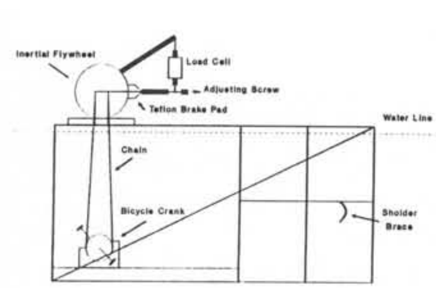
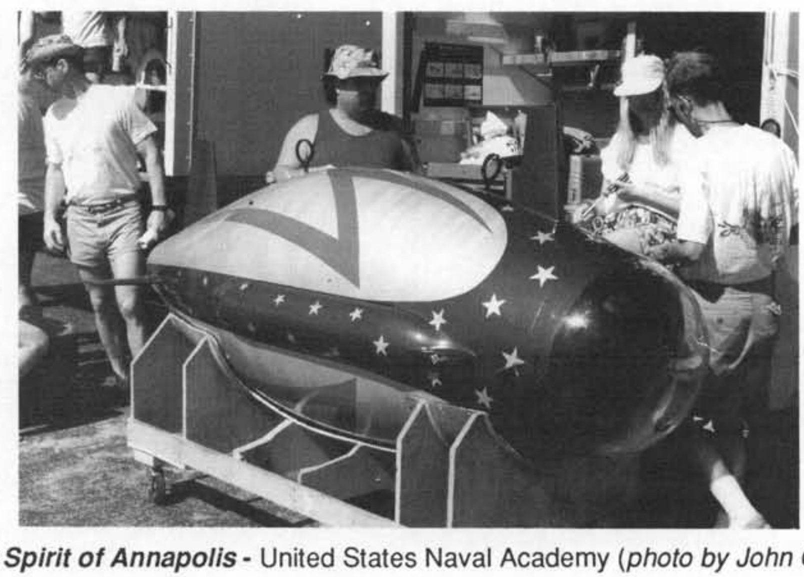


Figure 2. Underwater Ergometer



The Spirit of Annapolis - United States Naval Academy (*photo by John Chiffer*) and could be significant for future study on air consumption efficiency.

Our second objective was to figure out the maximum horsepower capability of our midshipmen divers. Early designs were based on the assumption that 375 watts (0.5 hp) could be obtained. Since well over one horsepower can be produced by humans on land it was a reasonable assumption. Further testing on the ergometer revealed that 375 watts (0.5 hp) was possible but only for very short durations. Figure 4 illustrates the power production capabilities of the midshipmen divers compared to bicyclists. It became

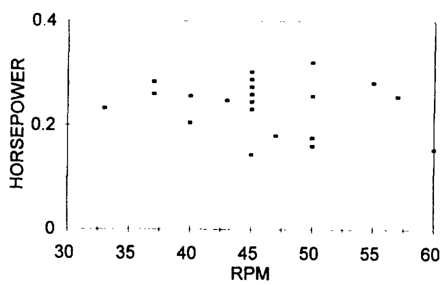


Figure 3. Power vs Cadence

obvious to us that we had extremely limited power capabilities, less than originally hoped.

To try to maximize the power capabilities, a great deal of effort was placed in the design of an optimum propeller. This was an area were we believed significant innovation was possible. As a guide to developing an efficient propeller we looked at human-powered aircraft designs. Here emphasis was placed on large, slow turning propellers. As our propeller design began to unfold, we too discovered that relatively large blades would be best. Specific propeller design took three distinct paths.

The first propeller, designed by midshipmen was a high efficiency variable pitch propeller, approximately 0.9 (3 ft) from tip to tip designed to spin at 120 rpm. Though this was perhaps the best design, and was specifically tailored for our hull,ä it proved to be too difficult to fabricate, requiring a three-axis NC milling machine. Secondly, a large, 1.2 m (4 ft) propeller was designed specifically for a 50-rpm cadence. The design effort here was greatly simplified. Although not as efficient a design, it had a major advantage, it could be built by hand at the Naval Academy. Thirdly, an off the shelf ultra-light aircraft propeller was purchased. These propellers had the general characteristic that we were looking for and most importantly could be obtained immediately. In the interest of time, we began in-water testing with the ultra-light propeller while the other propeller designs continued. Although only providing a 12 degree pitch, this propeller performed remarkably well, pushing our submarine to speeds of 1.26 m/s (2.5 kt). Faster speeds may have been obtainable, however the confined space of the tow tank, where we held our trial runs, prohibited this. Nevertheless, this propeller performed well enough to allow us to perfect our ballasting and maneuverability. Ultimately, the hand crafted propeller replaced the ultra-light propeller as it produced considerably more thrust.

The hub used to hold the two propeller blades was designed so that we could change blades at will. This was done so we could try out different blade designs with a minimum of effort. Also, initially we considered using two different sets of blades, one for the 100 meter sprint and one for the 400 meter course run. In the end we determined that one blade design would work well for both events. Most important, this hub would allow us to replace a broken propeller blade at the race sight if necessary. This conformed with our reliability objective.

Two types of drive train systems were considered, a linear system and a bicycle crank system. Linear drive systems have potential advantages in requiring less space, producing more thrust and requiring less leg motion by the propulsor. Several ideas were designed, but all of them were fairly complex and did not show a significance savings in space or an increase in thrust. These ideas were abandoned as not justifying their complexity and questionable reliability.

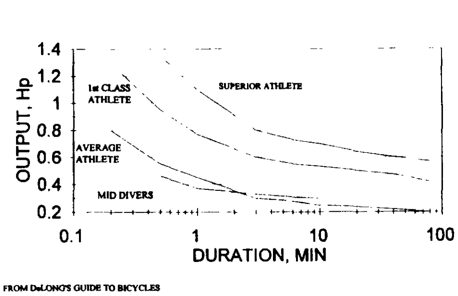


Figure 4. Power Output, Duration of Effort

A bicycle driven chain system was used for SQUID and proved very effective. Broken or derailed chains had caused problems for other competitors however and so a direct drive system was decided upon as the simplest and most reliable system. Since our propeller needed to spin at only 50 rpm, our direct drive system did not require any special gear reductions. Two miter gears connected the drive shaft to the bicycle crank. Using only two gears kept mechanical losses to a minimum. This system also had the appeal of simplicity and reliability. Trial runs proved the system to be effective. At the competition however, when the propulsor gave his maximum effort, the torque produced by our large blades caused our whole drive train assembly to twist. This destroyed the gear alignment causing the two miter gears to strip each other. A postmortem on the submarine revealed that the torque was so excessive that it even twisted our 12.3mm (0.5 in) stainless steel drive shaft about 10 degrees.

Because of its modular construction we could quickly remove and replace the entire drive train assembly. In this respect the simplicity of the system paid off. Because repairs were swift, we avoided being disqualified. Unfortunately the gear problems persisted, limiting our speed to two knots, which eventually got us eliminated from the race.

Although we didn't win the competition, we were pleased that most systems worked well, including the surface buoy and launch and recovery vehicle. Since an inadequate surface buoy eliminated SQUID from the previous competition, we were very interested in designing a buoy that would not be pulled underwater easily.

Several shapes were considered for the surface buoy in an effort to compromise between maximizing buoyancy and minimizing drag. Tow tank experiments indicated that our earlier surface buoy had a nose that was too blunt providing a large frontal area which waves could easily push underwater. Our improved design was shaped more like a ship's bow allowing the buoy to cut through the waves. This stream line shape added only four pounds of drag to the submarine when towed at speeds of five knots. At the race site the waves were very choppy but the buoy sliced through the water as designed.

Another system we were very pleased

with was our launch and recovery vehicle. This vehicle was an A-frame on wheels. It held the submarine above the water during transit. Once at the race site, the submarine was winched down into the water and then launched. Not only did it protect our submarine through the rough waters during the competition, it provided a stable work platform offshore.The next competition is scheduled for June 1995. Design efforts have continued and we should begin construction of the third hull to represent the Naval Academy shortly. This hull will provide another 20% reduction in frontal area, greatly reducing form drag. An efficient and reliable drive-train/propeller package will dominate our design efforts over the next two years. We are certain that tremendous advancements can be made in this area. Hulls cannot be made much smaller or more effectively shaped but slow speed propulsion system design is still in its infancy.

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Cost Effective Fabrication of a Graphite/Epoxy Submarine Hull

by
Glen H. Besterfield, Ph.D.
Wayne B. Paugh, B.S., B.A.

ABSTRACT

The fabrication of a cost effective graphite/epoxy submarine hull requires design consideration, prudent material selection, process preparation, and overall adherence to patience, teamwork, and dedication. The increasing presence and attributes of composites in marine applications has led to their inclusion in the design of the submarine hull. In unison with several constraints and parameters, the composite hull was designed to encapsulate two occupants, be free-flooding, and address significant safety issues. The methodology of construction was based upon a male plug / female mold / male part technique. Utilizing industry standards and processes, Hercules™ AS4/3502 carbon composite lamina was layered within a female fiberglass mold and cured at an acceptable temperature cycle. After yielding symmetric male part halves, the halves were joined using stainless steel rivets and strengthened through inserted circular aluminum stiffeners. The weight of the hull was significantly less when compared to fiberglass, while far exceeding its strength. At an overall project cost of $6,000, the finished submarine hull represents a cost effective and lightweight structure that has the integrity to endure marine conditions and the capacity for unprecedented superior performance.

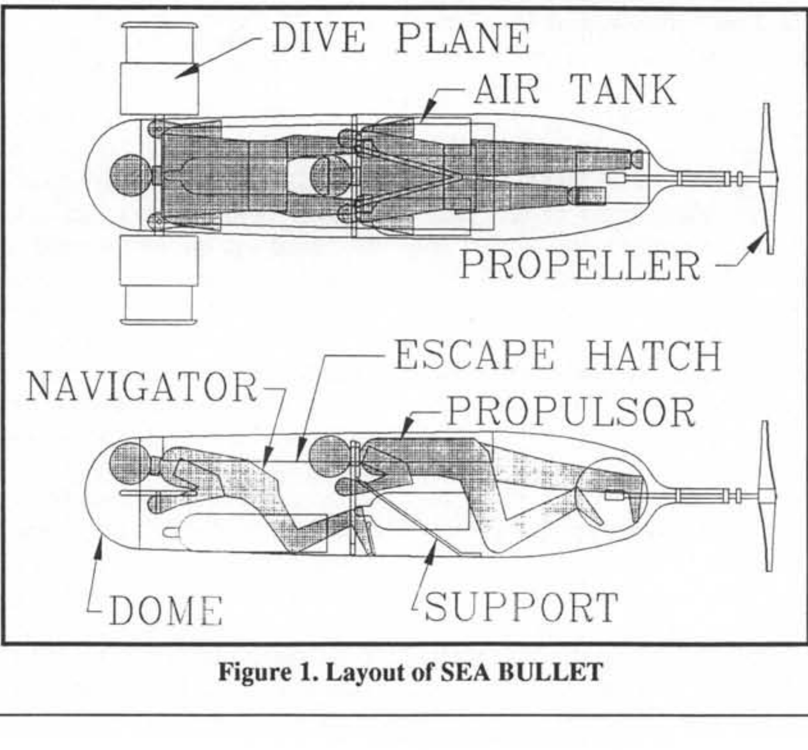


Figure 1. Layout of SEA BULLET

INTRODUCTION

Since the middle part of this century, composite materials have played an important role in the development of marine structures. In particular, the recreational boat building industry has invested considerable technical resources into the construction of fiberglass hulls. More recently, carbon masts and sailboat hulls have been utilized in the America's Cup competition. In a sport where speed is directly dependent upon the critical element of weight, wind-surfing boards and masts have been constructed with high-tech composites such as carbon and kevlar for almost a decade now. At the 3rd International Human-Powered Submarine Races, many of the submarines were comprised of advanced technological composites in some capacity. Considering this sequence of events, increasing technology, availability, and overall cost effectiveness, it seems quite certain that there will be numerous applications for advanced composites within the marine industry as the future inevitably unfolds.

The motivation for using high-tech composites in the marine industry is not only clearly evident, but multi-faceted. The most popular and inherent reason is the impressive strength-to-weight ratio of almost all composites. In fact, a typical graphite/epoxy composite has the ability to exhibit the strength of steel, and at the same time weigh five times less. Another significant advantage of composites is their notably high stiffness. A typical sailboat mast formed from a graphite/epoxy composite can be 3 times stiffer than an identical aluminum mast yet weigh half as much. Furthermore, there are numerous other advantages for using composites in a marine environment, for example,

- good corrosion resistance
- seamless construction ability
- high energy/impact absorption
- good dielectric properties
- absence of magnetic properties

There are varying types and combinations of composites which can be used in a marine environment, ranging from the traditional aspects of fiberglass to the innovative qualities present in high-tech composites. Examples of these composites are: glass fibers (S-glass or E-glass) / polyester matrix, glass fibers (S-glass or E-glass) / epoxy matrix, carbon fibers / epoxy matrix, kevlar fibers / epoxy matrix, and boron fibers / epoxy matrix. In addition to these separate and unique composites, a sandwich construction may also be employed, by mating with a core element. Finally, as was previously mentioned, there exists varied hybrid combinations that are equally effective and unique in both properties and performance.

HULL CONFIGURATION/ LAYOUT

Prior to actual hull fabrication, a considerable amount of time needs to be invested in the hull design. Design constraints and parameters limit and ultimately dictate the final hull shape. The primary design constraints that were imposed upon the design are as follows, with respect to brevity:

- two man enclosed hull
- free-flooding
- safety regulations

These constraints were set forth by the "Official Race Guidelines and Rules" issued by the submarine race organizational committee. Other constraints which could be employed, but were not forced upon the design, are: the minimization of surface area, frontal area, and internal volume, as well as simplicity, durability, and ease of maintenance. Of course, both surface and frontal area, and internal volume cannot all be minimized simultaneously. After taking into account the aforementioned constraints, the designer is free to design according to optionally self-imposed design parameters, based upon his or her own personal interests and innovations. While the design engineer has the flexibility of intuition, nature has the ability to admire a wide variety of human endeavors and unforeseen accomplishments. In unison, some of these design considerations might include, but are not limited to:

- dimensional geometry
- material options
- additional systems layout

The most important consideration in hull configuration and layout is the ergonomics and/or human factors. For example, a certain layout and geometry could be necessary depending on the chosen propulsion system. Other possible human factors which could influence the design are:

- occupant sizes and comfort
- underwater visibility
- reclined, supine, or prone positioning

In Figure 1, the final layout of the University of South Florida's "USF Sea Bullet" is shown. Note that the navigator and propulsor are both in the prone position and the primary emphasis is on ergonomics and human factors.

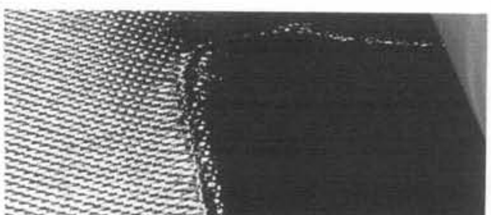


Figure 2. Hercules™ AS4/3502 Carbon Composite Woven-Roving Mat With Noticeable Small Weave Defects.

MATERIAL SELECTION

Hercules™ AS4/3502 Carbon pre-preg in 6" uni-directional tape and woven-roving mat was selected. Incidentally, this is the same material used in the General Dynamics™ F-16 Fighting Falcon. Hercules donated this particular carbon mat to the University of South Florida because it had small weave defects (See Figure 2), and therefore, was unable to comply with outstanding military specifications. In the material designation "Hercules™ AS4/3502", AS4 represents the type of graphite fiber and 3502 is the type of pre-impregnated epoxy resin. This combination results in an ~ 60% fiber volume fraction. Curing Hercules™ AS4/3502-T6 requires 8 hours in an autoclave at 350F with a specified ramp and 414 kPa (60 psig). These requirements dictate that the mold, and consequently, the fabrication methodology, maintain the ability to resist both high temperature and pressure. In

Figure 3. Initial Step in Male Plug Construction.

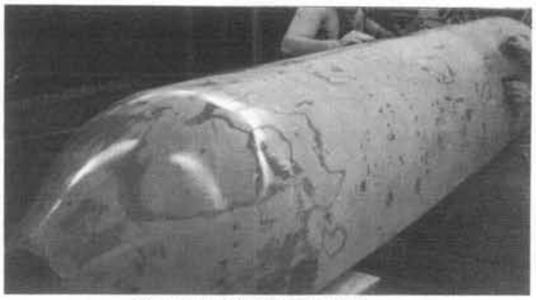


Figure 4. Final Male Plug

addition to the specific pressure accommodation, the laminate must also be vacuum bagged throughout the curing process, thus complicating the procedure even further.

MOLD FABRICATION

Once the hull material is chosen, the fabrication methodology may be determined and the subsequent construction phase is commenced. There are three prominent composite fabrication techniques for marine hulls (boats, submarines, etc.), that is,

- one-off construction
- female mold / male part
- male plug / female mold / male part

Each of the methods has advantages and disadvantages, and all three methods can be used for the fabrication of a submarine hull. A one-off construction technique is the easiest and quickest method, although a significant amount of time is needed in the final fairing and the additional use of an autoclave/oven contributes to the considerations of this procedure. The second two methods are very similar but the third method will typically produce a higher quality finish and was highly recommended by General Dynamics™. Consequently, the male plug / female mold / male part construction technique was implemented, based on advisory opinion and previous experience.

Construction of the male plug, which represents one half of the submarine

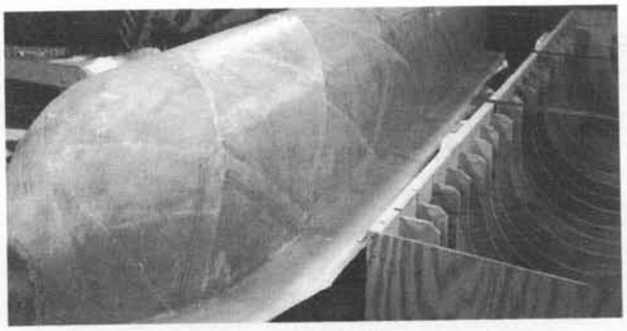


Figure 5. Final Female Mold with Support Structure

because of symmetry about the longitudinal axis, begins with 9.5mm (3/8 in) plywood templates of the hull cross-section. These templates of the actual submarine hull geometry are aligned and positioned every 152mm (6 in) along the longitudinal axis (See Figure 3). The templates are extended down an extra 127mm (5 in) in order to make the mold slightly larger. Once the plywood templates have been secured, paper mache' is applied over the templates, followed by a liquid polyurethane foam that is used to fill the voids. After sanding down the foam until the plywood contours are revealed, a thin layer (~ 6mm thickness) (1/4 in) of Bondo™ is applied and smoothed using sanding techniques to create a male plug that represents the final shape of the hull (See Figure 4). Note that it is of utmost importance to attain a Bondo™ finish that is precisely smooth and uniform because the final quality of the male plug dictates the ultimate quality and appearance of the actual submarine hull.

Once the male plug has reached completion, the next step is to lay-up fiberglass over the male plug to create a female mold. Prior to the lay-up, two compounds to induce mold release are applied to the male plug. Part-All™ Paste #2 and Polyvinyl Alcohol (PVA) were used, however, other similar chemicals are commercially available to ensure mold release as well. The female mold is constructed by laying fiberglass mat (a heavy weight mat such as 20 oz. should be used) and an epoxy resin over the male plug (See Figure 5). It is very important to lay the fiberglass mat in alternating directions to prevent mold warping or deformity after the mold is released. The authors suggest 6 lamina symmetrically oriented at 45/45/0/0/45/45 degree (See 45 degree orientation in Figure 5) will produce a female mold or laminate which is ~ 5mm (0.20 in) thick. An epoxy resin is used because the female mold will have to be heated to a sustained temperature of 350F. The authors suggest a 5 to 1 slow-hardening epoxy resin which will require several temperate days for a full cure. In our case, several intervals of outdoor daylight exposure greatly assisted in the timeliness and completeness of the mold cure cycle. After completing the lay-up and allowing the resin to harden, the female mold will easily release from the male plug, provided care was taken in all previous steps.

Note that the interior of the female mold is as smooth as the original male plug. Before executing the carbon lay-up, the female mold must be reinforced to withstand the rigors of vacuum and autoclave pressures. This is accomplished by constructing a wood frame and utilizing the previously discarded plywood from each of the original plywood contours (See the right side of Figure 5). The fiberglass mold is attached to the wood support structure with fiberglass strips and epoxy resin to prevent any deformation of the mold due to the vacuum and autoclave pressures. The final step in the female mold construction is to post-cure the epoxy resin. This is accomplished by placing the entire mold and support structure in an oven for 6 hours at 121C (250F).

CARBON LAY-UP AND VACUUM BAGGING SEQUENCE

The main emphasis of this paper is to document a low-cost construction method for fabricating a carbon composite hull. In the previous sections, the procedure for constructing a female mold is similar for any type of composite hull with the exception of using an high temperature epoxy resin

Before performing the carbon lay-up, the mold must be coated with a mold releasing interface which is similar to the Part All™ and PVA previously described. However, now a high

Figure 6. Carbon Lay-up Procedure.

temperature releasing agent must be incorporated. The authors suggest using Frekote™ 700NC. Three or four coats of the releasing agent should be applied at 30 minute intervals before the carbon lay-up is allowed to commence. It is important to precisely follow the manufacturer's instructions when applying any releasing agent. Ultimately, throughout the entire endeavor to completion, the authors found that preparation, patience and perspiration were significant keys to unlocking the pathway to success.

After removing the carbon material (1.2mm (48 in) wide roll of woven-roving mat) from an extended preserving freeze, all of the lamina should be cut and fitted prior to lay-up. The authors suggest 6 lamina (each lamina is ~ 3.8mm (0.15 in) thick) symmetrically oriented at 45/45/0/0/45/45 degrees to eliminate any isolated part deformations. This will produce a male part which is ~ 2.3mm (0.090 in) thick prior to curing and ~ 2.0mm (0.080 in) afterwards.

The first carbon lamina is laid into the hull and a heat gun or high-powered blow dryer is used to induce the carbon lamina to conform to the circular shape (See Figure 6). Furthermore, the use of heat is necessary to persuade the carbon to draw into any complex curvatures of the female mold. Before applying the second lamina,

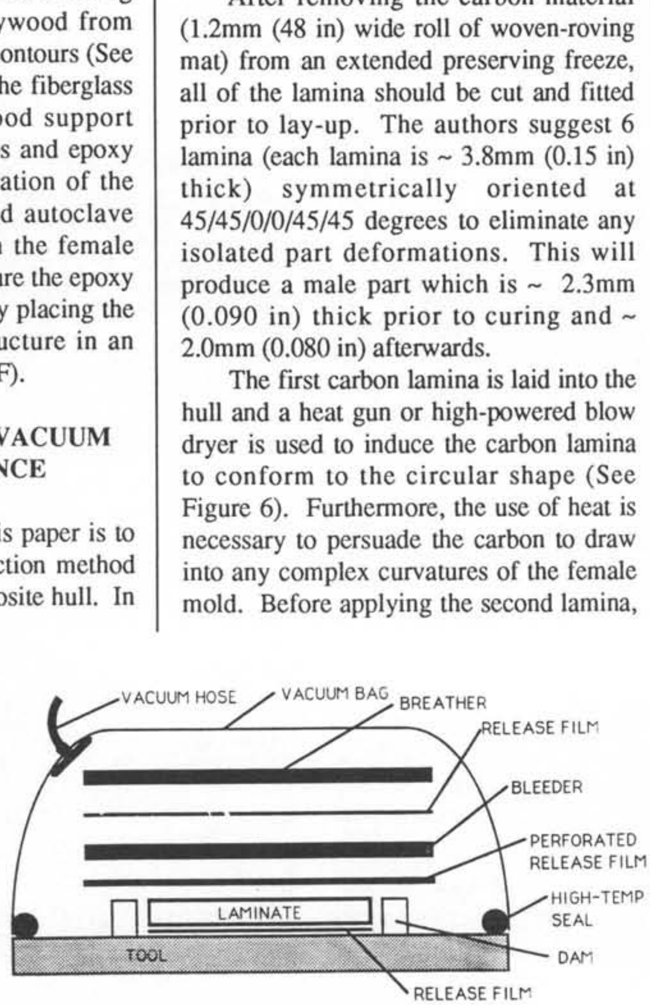


Figure 7. Typical Vacuum Bagging Sequence.

Figure 8. Final Lay-up After Vacuum Bagging.

a small amount of Acetone can be applied to the first lamina with a rag. This helps the second lamina adhere to the first lamina. In between each lamina, 5 layers of uni-directional tape has been laid in three places along the hull to act as ring or circumferential stiffeners (See Figure 6). Once all 6 lamina have been carefully positioned to produce the laminate, the mold is ready for vacuum bagging

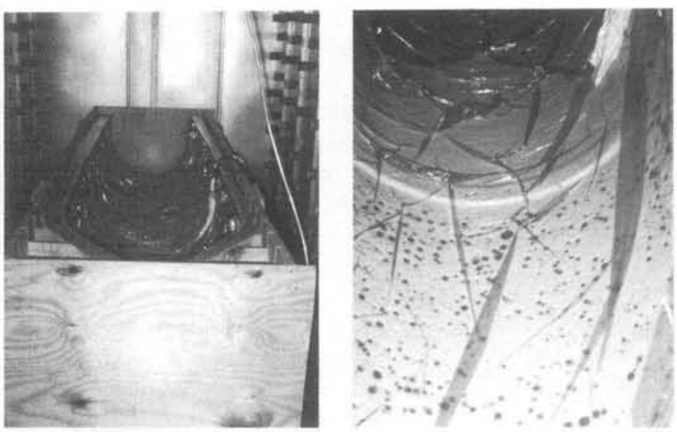
A typical vacuum bagging sequence, which depicts the maximum number of layers, is shown in Figure 7. Often, the minimum configuration can be used for almost all lay-ups except those applicable to military specifications. This minimum configuration incorporates only the release film, bleeder material (which also acts as the breather), and the vacuum bag. In place of the release film between the laminate and tool (i.e., mold), a high-temperature releasing agent has been applied to produce a smoother, more desirable finish. A perforated high-temperature release film (melts at 315C [600F]) is placed over the carbon and secured to the mold with a high-temperature flash tape. The perforated release film has several purposes, that is,

- allows excess resin to pass into the bleeder material
- keeps the bleeder material from sticking to the resin
- allows air voids in the lamina and layers to be removed with the vacuum as the resin temperature increases

The bleeder material is placed over the perforated release film and secured to the mold with flash tape. The purpose of the bleeder/breather, which is made of a spun polyester, is also multi-fold, that is,

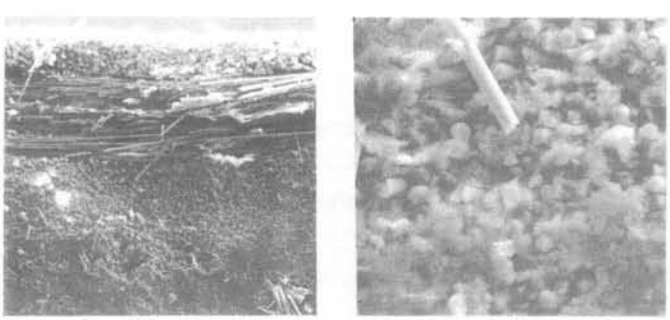
- absorbs the excess resin
- allows any gases created from the resin during curing to be removed by the vacuum
- creates a uniformly distributed vacuum

The final step in the process is securing the vacuum bag. Before laying the vacuum bag over the bleeder/breather material, a sealant (often called tacky tape and withstands an 204C [400F] cure) is attached and pressed around the entire perimeter of the mold. A nylon bagging film (withstands an 400F cure) is laid over the bleeder/breather material and carefully secured to the tacky tape, resulting in a air-tight seal. To create the vacuum, a copper tube is inserted between the mold and vacuum bag, and then sealed with additional tacky tape. The final mold and carbon lay-up after vacuum bagging is shown in Figure 8.



(a) (b)

Figure 9. (a) Final Lay-up in Oven and (b) After Curing.



(a) (b)

Figure 10. SEM Photographs (a) X 55 and (b) X 500

CURING PROCESS

Creating the required vacuum (~ 660mm (26 in) Hg) can be accomplished with a high-powered industrial vacuum cleaner or a vacuum pump. Although it is suggested that the carbon be properly cured in an autoclave at 414 KPa (60 psig), a large oven (not pressurized) will suffice if military specifications and ideal strengths are not mandatory. In fact, an autoclave at 414 KPa (60 psig) yields only an ~ 25% increase in final laminate strength with all other parameters equal. The actual advantage of using an autoclave is that a better compaction rate is achieved, which is conducive to higher material strength. However, a strong vacuum can produce similar results if superior strength is not imperative nor critical for a given application. In addition, the use of an autoclave complicates many of the procedures which have been previously described, that is,

- female mold needs to be thicker
- wood support structure possibly requires metal construction in order to withstand the increased autoclave pressure

Consequently, the authors suggest the use of an oven for overall simplicity and related economic reasons. In Figure 9a, the mold, support structure, and carbon lay-up is depicted in the oven. Note the copper tube extending from the support structure. The copper tube should be extended outside the oven in order to maintain critical vacuum levels throughout the entire curing cycle.

The curing process lasts for 8 hours at 149C (300F). Note that the required cure temperature for Hercules™ AS4/3502 is 177C (350F), however, this value can be relaxed and still give a complete cure provided the process still lasts 8 hours. In Figure 9b, the lay-up is shown after the curing process. Note that some of the resin has been absorbed by the bleeder material during curing. Once the curing is complete the carbon hull can easily be removed from the mold after cool down, provided release agents were applied thoroughly and properly.

After removing the half hull from the mold a small section should be analyzed in a Scanning Electron Microscope (SEM) for void content. SEM photographs of the graphite/epoxy composite are shown in Figures 10a and 10b at a magnification of

55 and 500, respectively. A bundle of graphite fibers/whiskers can be clearly seen in the upper portion of Figure 10a, whereas a single whisker appears in the upper portion of Figure 10b. The high density of particles in the epoxy matrix indicates a very good compaction and low void content.

HULL ASSEMBLY, STIFFENERS, AND FINISHING

The procedure which has been described in the previous sections of this paper is only for bottom half of the hull. By repeating the process, starting with the section entitled "Carbon Lay-up and Vacuum Bagging Sequence", the top hull halve can be fabricated. Before laying up the carbon, the cut-out for the hatches and a lap joint should be incorporated to simplify the assembly. Once both hull halves are complete, they are joined along the lap joint with epoxy glue and stainless steel rivets every 76mm (3 in). In order to strengthen the completed hull and provide hard points to mount internal components, three circular aluminum stiffeners were added (See Figure 11). The final step in the fabrication is to apply an epoxy paint to the internal and external surfaces. It is very important to do this before any water tests to prevent any water absorption by the carbon.

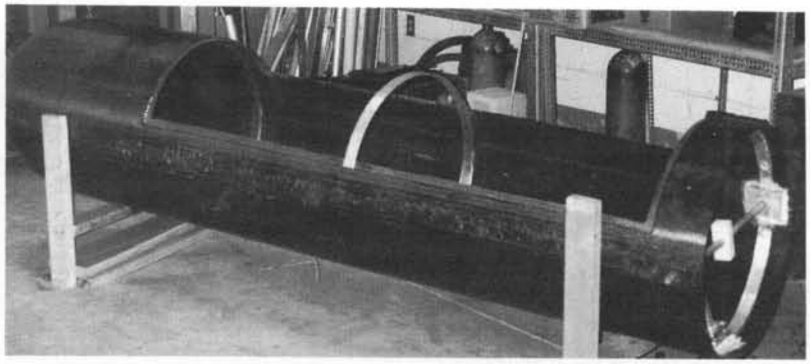


Figure 11. Assembling the Hull Halves.

Figure 12. "USF Sea Bullet" and Crew

SUMMARY

To summarize, the estimated overall cost of the entire project is presented below.

Hercules™AS4/3502	\$ 5000
Lay-up and Vacuum Bagging Material	\$ 500
Plug and Mold Related Materials	\$ 500
Total	\$ 6000

Although the expense associated with the Hercules™ AS4/3502 composite may seem excessive, its true market value is diminished due to the small weave defects present. Ultimately, the generous material and monetary donations that accompanied the project compensated for the expenses incurred, and in part, allowed the completion of a technologically impressive product, and more importantly, an invaluable educational experience. Below, in Figure 12, is a picture of the final submarine and its crew, as they appeared in Ft. Lauderdale competing in the 3rd International Submarine Races. Success was certainly evident in the overall performance of the submarine hull, the spirit of its team, and the unmistakable pride etched in their eyes.

Finally, addressing the strength and weight of the carbon hull, it easily outdistanced a similar hull made of fiberglass. Specifically, the 2mm (0.080 in) thick hull weighed 156N (35 lbs) and exhibited exceptional strength and stiffness. The use of a high-tech composite is also reinforced by the fact that it is inherently resistant to the potentially corrosive nature presented by an aquatic environment. However, as a direct result of the small weave defects in the woven-roving mat, and the non-standard curing procedure applied, values for the hull strength may be only estimated, with strengths at ~75% of the theoretical values

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H-P SUBMARINES: DESIGN PARAMETERS

by

P. K. Poole

It has been about five years since we first started our design efforts for the Naval Academy's entry (SQUID) in the First Human-Powered Submarine Race. Looking back on that experience, I recall not knowing specific values for a number of very fundamental parameters from which to begin the design process. Now, three races and nearly one-hundred entries later, I am amazed to hear contestants asking the same questions. Knowing, or at least accurately estimating your critical design points is essential to a successful design. Lack of such ability, in my opinion, is the primary reason for poor performance by a large number of contestants in the first three races. For this reason I offer the following views. The reader is cautioned that these are subjective comments based on my limited experience and personal observations.

SPEED: We might as well tackle this one right off since it is the probably the most often exaggerated. Every race I hear the same talk of predicted speeds in the 3.3, 3.8, and even 4.4 m/s (6,7 and 8 knots) range. For the life of me I cannot understand how these numbers were calculated. Given that the anticipated speed of the sub is the most significant parameter of interest, an accurate assessment of it is critical. There are numerous design values which contribute to the predicted speed, some having more influence than others. I hold the following to be the most significant:

(1) the POWER generated by the propulsor.
(2) the PHYSICAL SIZE of the sub.
(3) the EFFICIENCY of the propulsion device.

I'm sure all the hydrodynamicists out there are screaming to include the drag coefficient, Cd, which of course does have an effect on the speed, but I don't place it in the big three. My reasoning for this omission is that unless your design resembles a refrigerator box or you have gross hull-fluid separation, I can just about guarantee that your overall effective Cd (based on wetted surface area) is in the 0.010-0.012 range, putting everybody on the same plane. I've heard the arguments for "laminar flow" bodies and their incredibly low theoretical Cd, but I do not believe that one can be built and operated in the real environment and get the same results obtained in the lab. I don't dispute the theory or experimentally measured data. If a body can be made with no discontinuity effects in the viscous layer and operated in a laboratory like environment I may reassess my position. This of course would mean no heat distortion of the plastic nose dome, perfect interface between the dome and body, no hatches, no appendages, smooth, non-accelerating motions by the crew, and that ever popular well behaved ocean. Get real.

If, after you have sketched out a design which incorporates the expectation of a reasonably well designed propeller, the following will give you a conservative first estimate (for 3< L/D <8) of your predicted speed:

$$V = \left\{ \frac{K_1 \cdot P/D^2}{\xi^3 + \xi^2 - \xi - 1} \right\}^{\frac{1}{3}}$$

where

$$\xi = \left[1 + 0.1 \left\{ \frac{L}{D} \right\} \right]^{\frac{1}{2}}$$

V = Speed (m/s)	(kts)
L = Boat Length (m)	(ft)
D = Max Diameter (m)	(ft)
P = Shaft Power (watts)	(hp)
K_1= 0.01	(K_1= 600)

(Note: see Rule No. 1, below)

As an example, a sleek 3.4 meter (10 ft) long, 60 cm (2 ft) diameter hull with a 300 watt (0.4 hp) propulsor could get in the 2 m/s (3.6 kt) range. For the same boat to hit that magical speed of 2.73 m/s (5 kt) a propulsor power of roughly 750 watts (1 hp) is required. This assumes we know, or can make a reasonable estimate of, the propulsor power.

POWER: Making an accurate estimate of the power output of the propulsor can be very difficult. The human-engine is a peculiar entity, especially when operating underwater in an extremely small enclosure. The difficulty lies in the rather large number of models available and the lack of an accurate analytical model from which to work. Short of measuring the design power by testing the individual on an ergometer, in the water, in a similar position to that anticipated in the sub, your best bet is to rely on measurements taken from others or test in air and adjust the results to account for the underwater effects. Prior to the first race in 1989 everyone was using a number in or around 185-225 watts (0.25-0.30 hp) based on a study done by the Navy Experimental Diving Unit in Panama City, Florida[1]. The problem with this was that the Navy study identified the power level for endurance pedaling, not for the rather short time period anticipated in the Sub Races (approximately 2 minute sprint and 10 minutes head-to-head, MAX). Generally, the 200 watt (0.25 hp) level is well below what most physically fit males can achieve AFTER a short time practicing in the actual environment. I'd like to give some numbers on female propulsors but none were involved in my test program. From my experience and the measurements of midshipmen at the Naval Academy, power levels of 300 watts (0.4 hp) are readily achievable for most propulsors with practice, and 375-450 watts (0.5-0.6 hp) for the exceptional. The key parameter is the FREQUENCY at which the human-engine operates. The Navy study suggested a cadence rate at 40-45 RPM, which seems appropriate for the endurance effort at 225 watts (0.3 hp). I found this to be significantly low when trying to achieve higher power level.. This may be an individual trait but for the majority of those tested I found that by pushing the propulsor RPM to the 75-85 range THEN incrementally increasing the torque to the maximum possible while sustaining a ten minute test period, average power levels just below 375 watts (0.5 hp) could be routinely achieved with practice from nearly all those tested. Keep in mind my test duration requirement was ten minutes. For short (one minute or less) test periods, 450-550 watts (0.60-0.75 HP) are possible. If you are looking to do the 100 meters in record time your power design point would be significantly higher than that for the longer head-to-head racing.

The maximum pedaling cadence does appear to have a limit at about 95 RPM. Above this the power level begins to decrease (i.e. the maximum sustainable torque decreases faster than the RPM increases) and the air consumption rate increases rapidly. This later effect was evident even when the torque on the propulsor was reduced. I attribute this to the increased work required getting the air in and out of the lungs and the limitations of a demand regulator. The measured air consumption rate of the propulsor was in the 70-100 ALPM (Actual liters per minute) (2.5-3.5 Actual Cubic Feet per Minute) range. I have had great difficulty finding a simple correlation for the measured data from the 15 subjects (all midshipmen) tested over a two year period. The closest I have come are the below relationships, a human-engine Specific Speed, Ns, and Thermal Efficiency, Ne, based on metabolic energy conversion. All values refer to the propulsor.

$$\mathrm{Ns} = \frac{K_2 \cdot RPM \cdot \sqrt{BR}}{(\mathrm{T}/M_b)^{0.75}} \approx 0.05 - 0.06$$

$$\mathrm{Ne} = \frac{K_3 \cdot P}{BR} \approx 0.2 - 0.3$$

RPM	= Cadence	
BR	= Breathing Rate (l/min)	(ft^3/min)
T	= Torque (N-m)	(ft-lbf)
M_b	= Body Mass (Kg)	(lbm)
P	= Power (watts)	(hp)
K_2	$= 7X10^{-3}$	($K_2 = 1.6x10^{-4}$)
K_3	= 0.07	(K_3=1.9)

It would appear that increasing the crank arm would directly raise the power levels assuming the pedaling cadence can be maintained. I found that the crank arm could be increased to at least 20 cm (8 in) without causing a reduction in cadence for all subjects tested.

In discussions relating to the pedaling action in the underwater environment the effects of viscous drag and added mass due to leg motion always comes up. These effects are real and do effect performance but are generally difficult to minimize. The following is my estimate of the power lost to these effects:

$$P_{lost} = K_4 \cdot RPM^3$$

P_{lost}	= (watts)	(HP)
K_4	$= 1.32x10^{-4}$	($K_4 = 1.75x10^{-7}$)

At a pedaling cadence of 80 RPM an estimated 70 watts (0.09 HP) is lost. This effect is unavoidable and secondary to the effects of minimal inertia, regulator performance, and high leg-force demands. Figure 1 gives a set of curves used to estimate your propulsor's underwater power levels from measured values in air. The in-air power values are corrected for the losses due to leg motion and SCUBA demand regulator limitations. Leg force curves are cross-plotted to help optimize your individual propulsors power design point. For the uninitiated (anyone who hasn't had the pleasure of pedaling a sub) the experience is difficult to describe. Imagine pedaling a bike (with Kate Smith sitting on the handlebars) up a steep hill while breathing through a hose and your body in a zero gravity environment. Everything is in the legs, no body inertia, no body weight. All designers should experience this feeling to enhance their appreciation for the propulsor's task and to encourage optimum design.

Here's my propulsor power ranking scale:

Power	Description
75-185 watts (0.1-0.25 hp)	Look for a replacement.
185-260 watts (0.25-0.35 hp)	In the field.
260-335 watts (0.35-0.45 hp)	Serious contestant.
335-410 watts (0.45-0.55 hp)	A contender.
410-485 watts (0.55-0.65 hp)	Marry, kidnap, adopt ...

PHYSICAL SIZE: There isn't much to say here, except small is better only when it can be achieved without detracting from power. It is this fact that has driven many designers to consider the push-pull mechanism over the bicycle crank-arm. The large diameter requirement of the crank-arm type, which is usually located near the stern where most designers would like to be tapering off their offsets for good propeller flow, generally tends to increase the maximum hull diameter. The design space required for a pedal-crank mechanism, including the heal and toe

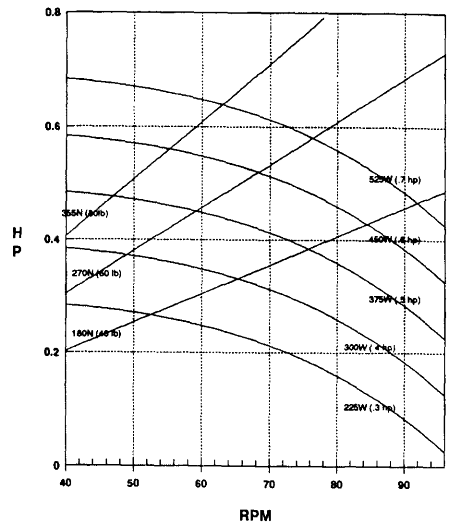

Figure 1. U/W Power and Leg Force Curves

clearances, is a circular arc of approximately 60 cm (24 in) in diameter whose center is shifted toward the heal half the difference between pedal-toe and pedal-heal lengths. On a men's size 9-10 shoe this distance is about four centimeters (1.5 in). The push-pull system, void of this large diameter requirement, lends itself to a fair hull geometry and normal crew orientation. I noticed in the last race an increased number of such systems. It is important for push-pull system designers to remember that the propulsor power design points of cadence and leg-force discussed for the crank-arm remain critical elements of their system's design.

Speaking of size, a significant number of designer tend to over-size their control planes. For many of the entries I have seen the drag on the control planes, even at zero angle of attack, would nearly match the hull drag. I guess its the "if you like a little, you'll love a lot" design theory. Designers tend to build large planes and, if time permits, cut them back from the outer extremities. The drag reduction resulting from the decrease in surface area may be offset by the increase in aspect ratio. My recommendation is to use a small, high aspect ratio plane, with a thickness to chord ratio of approximately 10%, placed at the greatest distance to the center of gravity and rely on high angle of attack maneuvers. Here is my greatly simplified equation for estimating the CHORD and SPAN of control planes (both rudder and dive) with aspect ratios (span/mean-chord) of three.

$$C = K_5 \cdot \sqrt{\frac{D \cdot L^{2.5}}{N \cdot X_p^{1.5}}}$$

$$S = 3 \cdot C$$

C = Mean Chord Length (cm)	(in)
S = Span Length (cm)	(in)
L = Boat Length (m)	(ft)
D = Max Diameter (m)	(ft)
N = Number of Planes	
X_p= Plane to CG distance(m)	(ft)
K_5= 2.5	(K_5= 1)

The 3.4 m (10 ft) example given above, with upper and lower rudders (N=2) located 1.2 m (4 ft) from the center of gravity, would require each plane to have a mean chord of 16 cm (6 in) and a span of 48 cm (19 in).

PROPELLER EFFICIENCY: For some reason this is the most overlooked aspect of design, yet it is possibly the most important. I have observed many contestant entries, with ingenious designs and exquisite construction in nearly every aspect, have propellers which doom all such efforts. Propulsion devices fall into two categories, those who wish to go fast and those who wish to demonstrate an alternative to the propeller. For the later, the demonstration is usually an alternative to going fast. Having done both, SQUID in '89 (fast) and SUBDUE in '91 (alternative), I can attest to the joy of going fast and the dismay of not. Since the alternatives are numerous, I will limit my discussion to propellers, except to make one observation. It is my opinion that the single major flaw in all the alternatives (Whale's Tail, Articulated Linear Thrust, Radially-Extended Rotary Impeller, etc.) is the inherent inability of their drive mechanisms to operate at sufficiently high frequencies to generate a reasonable thrust force. Enough said.

Large diameter propellers (diameters somewhere slightly less than the maximum hull diameter) are theoretically very efficient. The maximum efficiency possible is dependent again on the physical size of the sub if you assume that the propeller will not extend past the hull's maximum diameter. The relationship below, derived from momentum theory, gives a reasonable estimate of attainable propeller efficiency.

$$\eta_p = \frac{2}{1+\xi}$$

Where ξ is the same as given in the equation for estimating the sub speed. Of course, theory must meet reality at some point and I find that the maximum achievable efficiency is in the 90%-91% range. The efficiency is defined as the fraction of shaft power developed in thrust at terminal velocity. The false assumption that you need a Ph.d in hydrodynamics to design, and a three-axis CNC milling machine to construct an efficient propeller leads many to put off a concerted effort to do so. Good designs, for the relatively large diameter propeller discs that these subs allow, should be in the 85-90% range. There are a number of good propeller design papers in publication[2,3,4,5] which provide sufficient information to design and construct a reasonably efficient propeller. I'm a big fan of simple momentum theory design and fiberglass over hot-wired foam construction. If "reasonable" doesn't suit the need, then you may want to consult that Ph.d or the nearest prop guru in your neighborhood. If you can't find one, call me.

CONCLUSIONS: Building a H-P submarine is easy, designing and constructing a H-P submarine that is competitive in the biannual sub races is not. Of the roughly 100 entries in the three races held to date, only about 35% have gone over 1 m/s (2 kt) in the 100 meter time-trial and less than 25% have actually run in head-to-head competition around the course. Even more surprising, given the pre-race predicted speeds, is that less than 10% have actually broken the 1.5 m/s (3 kt) mark. Narrowing in on the propulsor's power will give more reasonable estimates of speed, allow proper design of mechanical drive and control systems, and better reflect the effects of minimizing the submarine size.

Poole's Rules for H-P Sub Design:

No.1	If your calculated speed is 2.73 m/s (5 kt) or greater, you made a mistake.
No.2	Bow planes are a must.
No.3	Buoyancy, stay positive.
No.4	A bluff nose is good, pointy bad.
No.5	Control by articulating propeller alone is no control.
No.6	With high static stability you can ignore dynamic.
No.7	Laminar flow ends at the nose cone, don't count on it anywhere else.
No.8	Propeller RPM > 120
No.9	Nozzles/Ducts/Rings **cannot** improve the efficiency of high efficiency, unfouled, unbroken propellers.
No.10	Keyways are better than shear pins.

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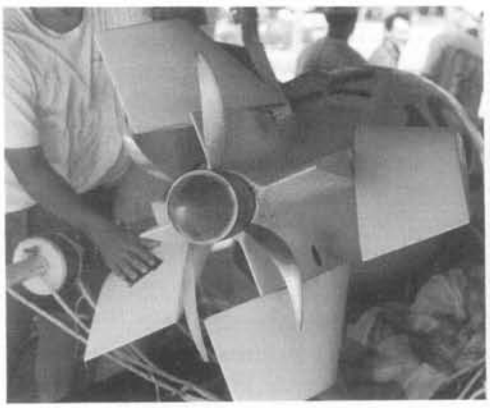
(ed-Pat is acting editor of H-P until Dave Wilson returns with the next issue. He is a consulting engineer in the areas of hydro-propulsion, underwater work systems, and hydrodynamics.)



A High Efficiency C-R Propeller (UNH's "SPUDS 3") and An Alternative Propulsion Mechanism (Independent–"Aqua Melon") *photos by John Chiffer*

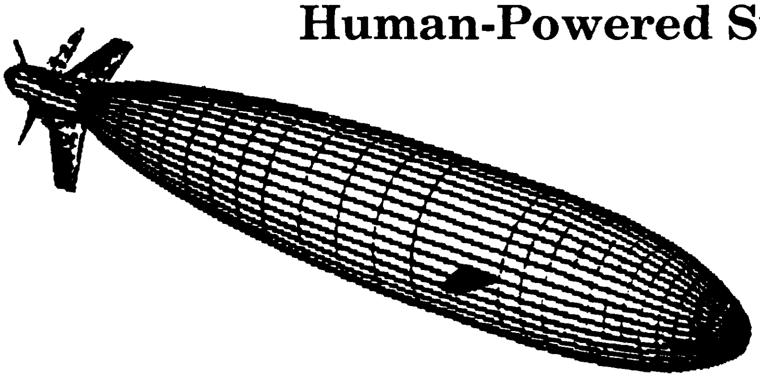
OFFICIAL TIMES / SPEEDS

3rd International Submarine Races- June 1993

SUB NO.	TEAM NAME	SUB NAME	TIME 100M secs.	SUB SPEED kts.	RANK	TIME 400M secs.	SPEED 400M kts.
8	Florida Atlantic University	F.A.U-Boat	45.58	4.26	1	3:53.86	3.32
33	Mass. Institute of Tech.	Sea Beaver II	48.75	3.99	2		
31	Tennessee Tech University	Tech Torpedo II	54.55	3.56	3	4:57.12	2.62
15	Team Borborygmi	Pelagic Cruiser II	57.84	3.36	4	4:57.38	2.61
6	Battelle Institute	Subjugator	60.65	3.20	5	4:28.59	2.89
34	Sub-Human Group	SubHuman II	61.29	3.17	6		
6	Benthos, Inc.	Subasaurus	63.34	3.07	7	4:24.65	2.94
39	Cape Fear Community College	Cape Fear	65.00	2.99	8	6:02.39	2.14
18	Gary Straughan	C-Scan II	65.21	2.98	9	5:26.89	2.38
9	Fla. Institute of Technology	SeaFIT	66.18	2.94	10	6:13.81	2.08
38	U of Calif/Santa Barbara	Love Missile	69.32	2.80	11		
17	Am Society of Mech Engineers	Project Neptune	69.50	2.80	12		
12	University of New Hampshire	Spuds 3	74.13	2.62	13		
24	German Sub Team	Borti II	76.88	2.53	14	5:56.61	2.18
3	Epcot Center	Submousible	79.37	2.45	15	4:44.56	2.73
10	University of South Florida	Sea Bullet	80.51	2.41	16		
42	Marine Institute/Newfoundland	Terror Nova	81.87	2.37	17	5:47.48	2.24
40	ETS Sub/Ecole de Technologie	Omer	90.45	2.15	18		
16	U of British Columbia	Killer Instinct	93.99	2.07	19		
21	University of Washington	Deep Purple	96.58	2.01	20		
1	U.S. Naval Academy	Spirit of Annapolis	97.73	1.99	21		
27	University of Southampton England	Submission Impossible	103.89	1.87	22		
30	California State Poly	Impatience	123.12	1.58	23		
22	Millersville University	Hoagie II	130.16	1.49	24	7:15.55	1.78

West Coast Invitational '94

Human-Powered Submarines



Scheduled for March 22 - 31, 1994
Offshore Model Basin, Escondido, San Diego County, California

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