Hydrodynamic Considerations for Semi-Submersible Minehunting Vehicles

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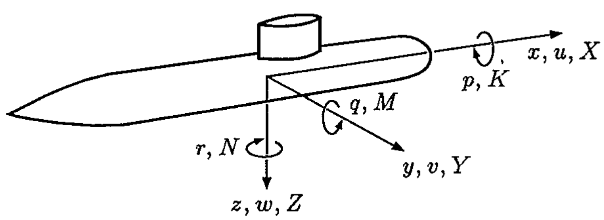
ABSTRACT

This paper evaluates hydrodynamic issues associated with the development of a Remote Minehunting Vehicle (RMV) for the Canadian Navy's remote minehunting system. The RMV will be an unmanned, autonomous, snorkeling semi-submersible with a deployable towfish which will be used for route surveying and locating mines on the sea floor. The hydrodynamic characteristics of the DOLPHIN Mark I semi-submersible are reviewed and some innovative changes proposed for consideration for the RMV design. These changes address resistance, propulsion, and hydrodynamic control issues at the preliminary design level. Cost is not considered.

NOMENCLATURE

Vehicle body fixed axes are shown below, with the origin on the hull centerline opposite the center of gravity.



K, M, N	body axes moments.
K', M', N'	moments nondimensionalized with $\rho U^2 \ell^3/2$.
ℓ	overall vehicle length.
m, m'	vehicle mass; $m' = m/(\frac{1}{2}\rho\ell^3)$.
p, q, r	body axes angular velocities.
u, v, w	body axes velocities.
U	overall velocity: $\sqrt{u^2 + v^2 + w^2}$.
x, y, z	vehicle body fixed axes.
X, Y, Z	body axes forces.
X', Y', Z'	forces nondimensionalized with $\rho U^2 \ell^2/2$.
δ_r	rudder deflection angle; positive clockwise as viewed from above.
ρ	fluid density.
ϕ	vehicle roll angle, same sign as p and K.

1 INTRODUCTION

DREA and ISER are evaluating hydrodynamic issues associated with the development of a Remote Minehunting Vehicle (RMV) for the Canadian Navy's Remote Minehunting System (RMS). The RMV is an unmanned, autonomous, snorkeling semi-submersible with a deployable towfish which will be used for route surveying and locating mines on the sea floor [1].

A remote minehunting operation is desirable because it eliminates risk to both personnel and capital equipment. A semi-submersible tow vehicle is more covert and, except for surge, less susceptible to wave induced motions [2] (which compromise towfish positioning) than a surface vessel, yet retains the air breathing endurance, high data rate real time communication link, and accurate global positioning characteristics of surface vessels. The towfish houses and positions sonar equipment used for imaging the bottom; the images are either recorded or sent to a mother ship for real-time processing. A deployable towfish allows quick transit to the hunt area (with the towfish undeployed) and adjustment of the towline length for bottom depth and speed-of-advance.

A candidate semi-submersible for a Canadian RMV is the DOLPHIN. Built by International Submarine Engineering Ltd., DOLPHIN (Deep Ocean Logging Platform Instrumented with Navigation) is a 24 foot long, 7400 pound vehicle with a snorkel supplying air to its diesel engine allowing the hull to stay at depths of 10 to 15 feet (3 to 5 hull diameters). DOLPHIN has been in service as a hydrographic survey vessel (without a towfish) since 1983. DND has contracted ISER to build a Mark II version of the DOLPHIN as a test bed for Canadian RMS development.

In this paper, the hydrodynamic characteristics of the DOLPHIN are reviewed and design changes proposed for consideration for the final RMV. These changes address hydrodynamic control, resistance, and propulsion issues for RMS operation. A design capable of withstanding launch and recovery from the deck of a small ship in high sea states is sought. A multi-use design option is also considered.

2 DOLPHIN MARK I AND MARK II

Figure 1 shows the Mark I version of the DOLPHIN. The hull is aluminum with a cylindrical midsection, a hemi-spherical cap for a nose, and an asymmetrical tail. Two large doors open on top of the hull giving access to a forward payload compartment and an aft engine bay. The mast/snorkel is a 4 inch inside diameter pipe surrounded by self-aligning fairings to reduce drag. A faired forestay helps support the mast and deflects objects floating in the water. To lower the center-of-gravity, a 1000 lb keel is suspended on a large appendage well below the hull.

DOLPHIN has four independent all-movable control surfaces. Two of these are the foreplanes which are used for depth and roll control. The sternplanes are coupled to one actuator and provide vertical plane stability and pitch control. The rudder is a single vertical appendage beneath the sternplanes and provides horizontal plane stability and directional control. In addition to these control appendages, horizontal plane stability is enhanced by a fixed vertical stabilizer above the sternplanes which encases the engine exhaust outlet. All appendages have thick NACA 0025 sections.

The propulsion system consists of a 150 hp diesel engine driving a single screw propeller at around 900 rpm. Top speed is 16 knots.

The Mark II RMV concept demonstrator is based on Mark I. It uses the same control surfaces and control strategy. The hull has the same 39 inch diameter but is 4 feet longer. Modifications include: a 350 hp engine and contra-rotating propellers, a larger 5 inch mast, a weight increase to 9450 lb, a tow cable drum and traction winch in the keel, and a towfish housing at the keel tip. Mark II will have a top speed of about 18 knots with the towfish housed at the keel and an endurance of about 16 hours when towing. To compensate for weight changes that occur when the towfish is deployed, for density changes when fuel is used up and replaced with sea water, and to give additional pitch control to counteract the tow tension, Mark II will incorporate 2000 lb of variable ballast.

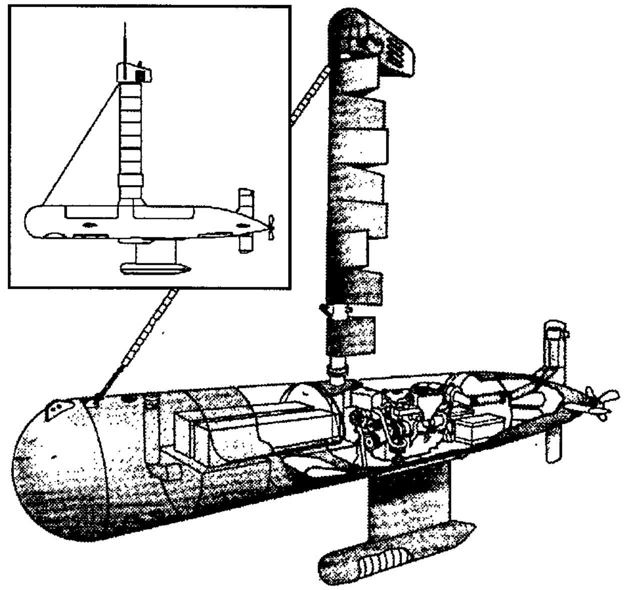


Figure 1 *DOLPHIN Mark I*

When turning during towing operations, cable tow-off angles result. This causes a rolling moment on the vehicle and affects towfish positioning. Operational procedures can minimize these effects but the effects still influence turning efficiency. This paper only considers roll control. DOLPHIN's roll control is described in the next section and an enhanced roll control capability in §5. Towfish displacement control will be addressed in a future paper.

3 CONTROL ISSUES

Shupe and McGeer [3] present a linearized hydrodynamic analysis of DOLPHIN that shows it is unstable in pitch and heave and has underdamped stability in roll and yaw. They present an active control system design which gives the desired stability in all degrees of freedom. Full scale trials with DOLPHIN show that, indeed, its control surfaces are continuously active with amplitudes of a few degrees.

3.1 Yaw and Roll Control

DOLPHIN has always had reliable directional control; its minimum turning diameter is about 10ℓ. As shown below, this turning capability is well predicted by linear analysis. Recent trials with a short 200 foot tow attached at the keel did not compromise this turning performance. Thus, although a detailed 'turning while towing' analysis should be carried out at some point, it is not a critical issue at this time.

On the other hand, roll problems are exacerbated in towing applications, especially during turns, and recent Mark I trials (discussed below) show unexplained roll characteristics. This merits analysis.

During a turn to port, say, the rudder generates a force to starboard and a negative rolling moment. A strong crossflow develops as the hull yaws about a longitudinal 'pivot point'. Forward of this point the crossflow comes from port, it is zero at the pivot point, and it comes from starboard aft of it with ever increasing strength towards the stern. The rolling moments generated by the crossflow over the hull (with its asymmetrical stern), mast, keel, and asymmetrical tailfins are all a function of their axial location relative to the pivot point, as well as their geometry. To predict the net rolling moment in a turn, the pivot point location must be known.

For ships or submarines, the pivot point is close to the nose. As shown below, DOLPHIN's pivot point is much further aft than this, mainly because of the keel.

DOLPHIN's linear horizontal plane hydrodynamic characteristics are estimated in Table 1. These are based on semi-empirical techniques from both the open [4,5,6,7,8] and closed literature. Sidewash from the keel in a crossflow has a major impact on the rudder, reversing the flow there in fact, and this is accounted for using a simplification of the technique presented by Spreiter and Sacks [9].

The Y_v' and N_v' estimates agree well with static towing tank measurements made by the Institute for Marine Dynamics (IMD) in 1991 using a 1/4 scale fully configured DOLPHIN model. However, the K_v' estimates are almost twice the values suggested by these tests, the implications of which are examined below. Rotary data (r derivatives) are unavailable.

The horizontal plane control derivatives are estimated as:

$$Y_{\delta_r}' = 0.01, \quad N_{\delta_r}' = -0.0041, \quad K_{\delta_r}' = -0.00056$$

also with an error of about 20 percent.

In a horizontal plane steady turn, the linearized lateral force and yawing moment equations of motion for an underwater vehicle [10] give:

$$v' = \frac{(m' - Y_r')\,r' - Y_{\delta_r}'\delta_r}{Y_v'} \tag{1}$$

$$r' = \frac{(Y_v'N_{\delta_r}' - N_v'Y_{\delta_r}')\,\delta_r}{N_v'(Y_r' - m') - Y_v'N_r'} \tag{2}$$

where $'$ indicates quantities made dimensionless by ρ, U, or ℓ (24.1 ft); $m' = 0.0164$. The local crossflow is given by $v(x) = v + rx$ and the pivot point (where $v(x) = 0$) is independent of δ_r:

$$x_p = \frac{-v}{r} = \frac{-v'\ell}{r'} = 0.5 \text{ ft.} \tag{3}$$

$100 \times$	Y_v'	N_v'	Y_r'	N_r'	K_v'	K_r'
Hull	−1.3	−1.93	0.42	−0.19	.002	−0.03
Keel	−11.8	0.34	1.40	−0.06	1.24	−0.15
Vertical Stabilizer	−3.1	1.30	1.41	−0.58	−0.16	0.07
Rudder	0.5	−0.20	0.34	−0.12	−0.02	−0.02
Mast	−0.2	−0.01	−.01	−.001	−0.07	−.004
Totals	−16.0	−0.50	3.56	−0.95	1.01	−0.12
Errors	15%	100%	12%	15%	25%	30%

Table 1 *The DOLPHIN Mark I horizontal plane static derivatives. Yawing moments are about the CG, 0.449ℓ aft of the nose, while rolling moments are about the hull centerline. The error estimates are based on a relative error of 20 percent in each component. The error in N_v' is large because it is close to zero.*

This puts the pivot point about midway between the mast and keel leading edge. Without the keel's contribution to the derivatives in Table 1, the pivot point would be 0.25ℓ further forward.

All of this is relevant to the roll angle ϕ attained during a steady turn. Ignoring foreplane roll control, the rolling moment equation of motion [10] shows that ϕ is dependent on v and r:

$$\sin\phi = \frac{m'z_G'r' + K_r'r' + K_v'v' + K_{\delta_r}'\delta_r + Q'}{m'g'(z_G' - z_B')}. \tag{4}$$

Here, z_G and z_B are the vertical coordinates of the centers of gravity and buoyancy, g is the gravitational constant, and Q is the torque applied to the vehicle by the propeller. With $z_G = 1.15$ ft, $z_G - z_B = 0.8$ ft, $Q = 570$ ft lb, $\delta_r = 25$ degrees, and v and r given by equations 1 and 2, $\phi = -1$ degrees. During a turn to port, DOLPHIN should have negligible roll.

This result is relatively insensitive to the coefficient errors noted above which give:

$$\phi = -1 \pm 4 \text{ degrees.} \tag{5}$$

If the K derivatives are reduced by 50 percent, as suggested by the IMD experiments, then $\phi = -8$ degrees.

This roll angle prediction disagrees with results measured in 1995 full scale sea trials with DOLPHIN. With test parameters identical to those used above and a vehicle speed of 11.5 knots, these trials produced a yaw rate $r' = -0.23$ and a roll angle $\phi \approx 15$ degrees. The measured yaw rate agrees reasonably well with the equation 2 prediction of $r' = -0.19$. However, the ϕ prediction and measurement are quite different, and strikingly so since the prediction ignores any foreplane influence while the control system in these trials had the foreplanes fully deflected to ±25 degrees to right the vehicle.

This roll angle discrepancy may be attributable to the foreplanes themselves. Figure 2a shows the vortex field generated by the foreplanes when oppositely deflected to counteract a roll to starboard, as in the above example. Figure 2b shows the trailing vortex field opposite the keel with, for simplicity, the hull bound vorticity treated as one coalesced vortex on the centerline. The trailing vortex positions assumed at the tail are shown in Figure 2c, but are only nominally correct in the presence of a crossflow, the boundary layer, and afterbody separation. This vortex field is used to estimate the effect of foreplane downwash on rolling moment.

The lateral velocity induced by the trailing vorticity along the vertical plane of symmetry ($y = 0$) containing the keel, rudder, and vertical stabilizer is:

$$v_i = \frac{\Gamma}{\pi z(1 + z^2/b^2)}. \qquad (6)$$

The normal velocity induced across the sternplanes is:

$$w_i = \frac{-\Gamma}{\pi y(1 - y^2/b^2)}. \qquad (7)$$

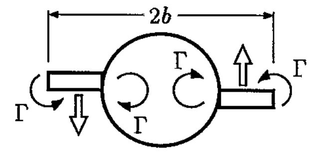
Integrating the local lift generated by the incidence angles v_i/U and w_i/U along the spans of the trailing appendages, and normalizing by an identical integration along the foreplanes, gives an estimate of foreplane downwash *induced* moment K_i relative to the moment generated at the foreplanes alone K_f, as shown in Table 2.

	Keel	Vertical Stabilizer	Rudder	Stern-planes	Total
K_i/K_f	−0.8	−0.2	−0.1	−0.3	−1.4

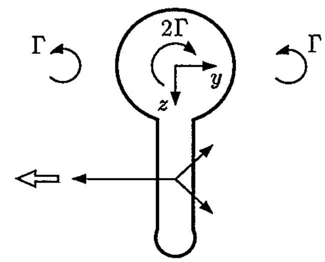
Table 2 *Torque induced on trailing appendages by the ideal, simplified vortex field of Figure 2.*

This simplified analysis should be best in straight line sailing conditions when crossflow is absent. Such conditions were present for about 100 seconds at the beginning of the above trial. During this period, the foreplanes were quite active and DOLPHIN's average roll angle was gradually decreased from about 4 degrees (propeller torque induces about a 5 degree roll angle) at $t = 0$ to about 1 degree at t=100. At no time did the roll angle cross zero. This also suggests that foreplane roll control is not what it should be but, at least, that $K_i/K_f > -1$.

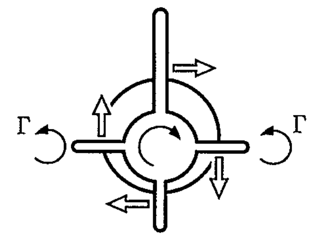
During turns, crossflows shift vortex trajectories and large interactions can suddenly occur, especially



a) *Foreplane generated counter-clockwise moment.*



b) *Induced clockwise moment at the keel.*



c) *Induced clockwise moment at the tail.*

Figure 2 *Looking forward at the vorticity ( ), velocity fields ( ), and forces ( ), generated by opposing deflections of the foreplanes.*

if a vortex intersects a control surface. This may have occurred during the tight turn in the above trial. At the 11.5 knot trial speed, the foreplanes directly generate $K_f \approx 2700$ ft lb of torque when differentially deflected 25 degrees, 40 percent of which can roll DOLPHIN about 10 degrees. Thus, Table 2 shows that, even without vortex/appendage intersection, the trailing foreplane vorticity has the energy to affect large vehicle roll angle changes.

One way to overcome this foreplane downwash interference problem is to shift roll control to the sternplanes, a solution which is accomodated in the Mark II design. An alternative strategy is discussed in §5.

3.2 Depth and Pitch Control

DOLPHIN's depth and pitch control capabilities are reported to be good. Depth control is achieved with symmetrical foreplane deflections. In this case, the two hull bound vortices have opposite sense and cancel. A continuous vortex filament extends through the hull from one foreplane tip to the other and is shed from the tips as shown in Figure 2, except that the shed vortices are now of opposite sense — the classical lifting line horseshoe vortex.

As a result, the trailing vorticity generates no lateral velocity component at the vertical plane of symmetry. It does, however, reverse-load the sternplanes generating a pitching moment in a direction which aids the depth change. This would not necessarily be bad if the downwash interaction was consistent, but the interaction is strongly dependent on the locations of the trailing vortex trajectories and these depend on the maneuvering history of the vehicle.

While the current depth control is adequate, it is desirable to reduce foreplane downwash interaction with the tailfins. This will stabilize the linearized hydrodynamic characteristics used by the control system.

4 DRAG

The DOLPHIN drag D (in lb) is estimated as:

$$D = 9V_k^2 \quad \text{Mark I with no tow} \tag{8a}$$

$$D = 47V_k^2 \quad \text{RMV with tow at full depth} \tag{8b}$$

where V_k is the speed in knots. The latter case is the RMV design condition. It shows that vehicle drag is only 1/5 of the total so that improvements in vehicle drag have a diminished impact on overall system performance. However, vehicle drag is still worth examining. Propulsive efficiency is discussed in §6.

4.1 Mast Drag

In 1989, full scale trials were carried out with DOLPHIN (without a tow) to measure its drag with and without the mast attached (an electric motor was used for propulsion). The results showed that at 10 knots the total vehicle drag was 895 lb with the mast generating 353 lb, or 40 percent of the total. The mast drag components are roughly estimated as:

2D Section	Ventilation	Spray	Wave	Forestay
21%	34%	4%	40%	1%

The section drag estimate assumes a drag coefficient of 0.06, based on frontal area. Ventilation drag occurs when a surface vent opens up behind the mast, going several feet into the water. It is estimated empirically [11] using the minimum pressure coefficient for the section. Spray drag is estimated using Horner [12]. Wave drag is taken to be the remainder.

An effort was made to calculate wave drag using a numerical potential flow linearized free surface program. The results were unrealistically high, probably because of the high Froude numbers and the nose bluntness of the mast section. However, relative differences in these calculations are worth mentioning. They show, not surprisingly, that wave drag is much smaller when the 25 percent thick self-aligning mast section is replaced with NACA 0021 and 0015 sections (keeping the maximum thickness constant).

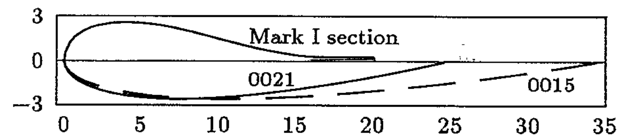


Figure 3 *Mast profiles (in inches) for the Mark I and two NACA sections, all with the same thickness.*

Using these, albeit questionable, wave drag relative change predictions, together with section and ventilation drag estimates for the longer NACA sections, leads to the following total mast drag estimates:

Mast	Total Drag	Reduction
IFS 61 TR 25 (Mark I)	353 lb	
NACA 0021	194 lb	45%
NACA 0015	161 lb	54%

Figure 3 shows these profiles. The higher drag of the TR 25 profile probably results from its blunt nose and afterbody contraction.

The NACA profiles do not make good self aligning fairings because their narrow noses force the mast pivot axis aft towards the center of pressure, thereby reducing the self-aligning moment arm for the fairing. However, a NACA 0015 profile is used in the next section as the basis for a rigid, flapped mast with a 22 inch chord length. Using the methods discussed above, this mast is estimated to have a total drag of only 106 lb.

4.2 Hull Drag

Based on streamlined submarine-like hull data, the DOLPHIN hull viscous drag is estimated at 218 lb at 10 knots. Using standard drag estimation techniques [6,8], the appendage drag is estimated at 93 lb. Numerical predictions of hull wave drag at 3 m depth are 51 lb and are considered reliable. So, mast-less vehicle drag is estimated at 67 percent of the 542 lb that was actually measured in the above trials.

This discrepancy is reasonable given the hull form, thick appendages, external weldments and attachments, and holes, slots, and gaps in the hull. Increased hull drag was a DOLPHIN design consideration accepted to reduce manufacturing and repair costs and to increase ease of use.

4.3 Reduced Overall Drag

By using the rigid mast of the next section and reconsidering the above hull drag compromises, it may be possible to reduce DOLPHIN's drag by about a third, which is about a 5 percent reduction in overall RMV drag (equations 8). Further improvements are possible by using a thicker and shorter body ($\ell/d = 6$, say) designed around its propulsor.

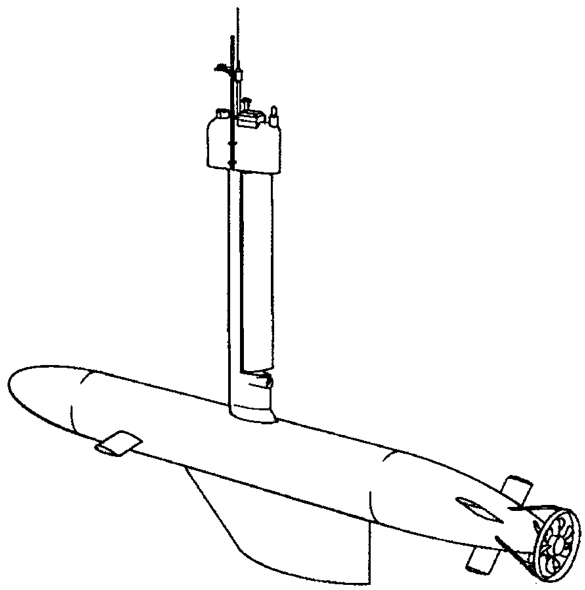


Figure 4 *RMV1 with a 27 foot hull; retains the concept of a devoted RMV vehicle.*

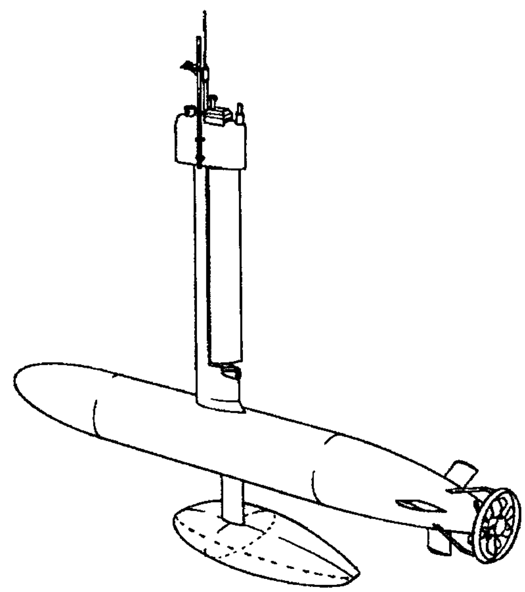


Figure 5 *The multi-use RMV2 with a 24 foot hull; a general purpose vehicle convertable to an RMV capability with specialized payload and pod modules.*

5 IMPROVED HYDRODYNAMIC CONTROL

In this section, two RMV designs are presented which provide enhanced roll control to counter large cable tow-off moments. These designs are shown in Figures 4 and 5. Both use a rigid airfoil section mast (without a forestay) which addresses the mast drag issue discussed above. The mast uses an actively controlled plain flap over most of its span for roll control. As well, the base of the mast is used for the engine exhaust passage, allowing the tailfin design to be optimized. The RMV hulls use streamlined nose and tail sections.

5.1 Roll Control

Tow-off angles during tight turns are difficult to predict but may reach 30 degrees, the maximum tow-off angle the tow cable winch will accomodate. For the heavy tow condition, this results in a tow-off moment of 9000 ft-lbs.

One way to get the roll control necessary to overcome these moments is to increase the moment arm at which the roll control surface acts. Increasing the spans of horizontal appendages is undesirable because of increased vulnerability to damage during launch and recovery operations. A bottom appendage is at risk in shallow water and may interact undesirably with towfish operations. At the top, the mast already has a long span and would benefit from improved fairing, so it is the natural choice for a control surface. The mast is also vulnerable to damage but has the advantage of always being visible.

Figure 6 shows a section of the proposed mast. It has a shorter chord length than the self-aligning fairing used for the large 5 inch inside diameter DOLPHIN Mark II mast, and about half the frontal area. Yet it has as large an air inlet and an equally large exhaust area. As noted in §4.1, the total drag on this mast, with the flap undeflected, is expected to be about 1/3 that of the Mark I mast.

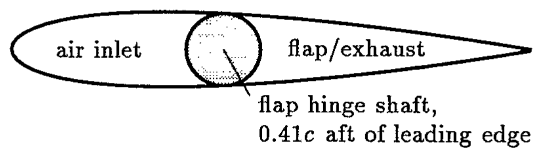


Figure 6 *The mast's 22 inch chord, NACA 0015 section with air inlet, engine exhaust, and a plain flap.*

If the engine exhaust uses the first 2 feet of the mast (keeping the exhaust outlet at its current height), and the flap extends along the remainder of the span, then at the minimum 10 ft operating depth (6.5 ft of submerged flap), the flap will generate about 300 lb of force per degree of deflection at 10 knots, or 2000 ft lb of torque. This means the worst-case cable tow-off moment can be countered with a 5 degree flap deflection. Small flap deflections ($<$ 10 degrees) are desirable because mast drag will likely be high at even moderate deflections.

The axial location of the mast should be near the turning pivot point of the vehicle so that crossflows are small. A flap deflection 1.6 times larger than the local crossflow angle is required to zero the rolling moment generated by crossflow over the mast, neglecting any counterbalancing from the keel.

An additional effect when using the mast for roll control is side force. For each degree of flap deflection, about 2/3 of a degree of sideslip angle is induced on the vehicle, at any speed. However, large flap deflections are only necessary when the flow or tow is asymmetrical, when large side forces of unknown magnitude are present in any case. When cruising at 10 knots, the induced sideslip angle will be less than 0.05 degrees per degree of vehicle roll being corrected, and less at higher speeds. The prevailing currents in the operations area are probably of greater concern.

Propeller torque imbalance may require the largest steady state roll correction during straight ahead cruising, as discussed in §6.2.

Most of the deflected flap generated vorticity should be shed from the inboard edge of the flap and is not expected to interact appreciably with downstream appendages (see tailfin design below).

DOLPHIN trials have shown that rigid fairings generate large side forces that are destabilizing in roll, so the RMV will require active flap control. DOLPHIN roll control is also active.

5.2 Depth, Pitch, and Directional Control

Figures 4 and 5 show different depth control mechanisms. In the RMV1 design, the foreplanes are retained for depth control, but the tailfins are replaced with a symmetrical 'X-tail' configuration (fins are 45 degrees above and below the horizontal). This has several advantages over the Mark I tailfins:

- interaction with foreplane vorticity is minimized;
- if a maneuver results in a close encounter between trailing vorticity and two tailfins (as has occured with Mark I), the impact on pitch control, to which all four tailfins contribute equally, is halved;
- for the same appendage dimensions, the X-tail has $\sqrt{2}$ times more horizontal and vertical plane control than a conventional 'plus' configuration;
- for a given span length, X-tail fins protrude horizontally less than do sternplanes, making damage during launch and recovery operations less likely.

The RMV2 design uses a deflectable underslung pod for depth control. The pod houses the cable drum, winch assembly and alignment system, the towfish capture mechanism, and ballast control hardware. The pod has several advantages over the foreplanes and keel:

- it provides twice the lift per degree of deflection as the foreplanes but with only 2/3 their span, making it less susceptible to damage;
- it is less delicate than the foreplanes;
- it provides room for the cable winch to align itself with the tow-cable at large tow-off angles;
- it reduces hull length requirements by housing the cable drum and some ballast control capability;
- it provides a multi-use vehicle by confining RMV specific hardware to a detachable pod replaceable with a much smaller one matching Mark I depth control and keel ballast characteristics, or perhaps a fuel tank pod for extended operations.

The pod and RMV1 keel generate about the same drag. Although the keel moves the turning pivot point aft, minimizing the crossflow at the mast, the rigid mast itself may accomplish the same thing. Further analysis is required here.

The RMV2 is shown with a Y-tail configuration. Pitch control is provided by the arms of the Y which are canted 30 degrees above the horizontal. These fins also help with directional control. The main advantage of the Y-tail is its lower susceptability to trailing vortex interference. The arms of the Y give better clearance to both mast and pod trailing vorticity than does the X-tail, and the stem of the Y is unaffected by the symmetrical wake from the pod. With 3 actuators, the Y-tail has less redundancy than the X-tail (with 4 actuators) but more than DOLPHIN.

RMV1 can also use a Y-tail if the foreplanes are lowered to the keel to reduce interference.

The Y-tail lacks the control of the X-tail for identical fin dimensions. For equivalent control, Y-tail fin dimensions need to increase to provide about 50 percent more lift per fin than X-tail fins.

6 THE PROPULSION SYSTEM

In this section, the propulsion system is examined with a view to increasing operational endurance. The calculations below show that substantial efficiency improvements are possible by increasing propeller diameter and reducing rpm. Interestingly, the endurance improvement that results comes not from fuel savings but from reduced power requirements allowing the engine duty rating to be upgraded.

Various propulsor configurations and sizes are examined for desirable characteristics in several areas: efficiency at both tow and transit conditions, torque balance, weight, tow cable protection, and installation issues. Some attributes are identified for the following configurations.

Conventional Single Propeller With Fixed, Radially-Varying Pitch: rugged, single-shaft, efficient at light loading, low weight, limited capability at widely different loads, requires a cable guard and torque balance, most efficient if hub diameter is about 0.2 of rotor tip diameter.

Pre- and Postswirl Propulsors: high efficiency if lightly loaded, single shaft, stator vanes provide torque cancellation at one operating point but not at widely different points, vanes may have integral ring for cable protection, hub/tip ratio of about 0.5 is acceptable, additional blade-rate frequency from stator wake, stator may reduce wake deficits to improve cavitation performance.

Contra-Rotating (C/R) Rotors: highest efficiency at light loads, intrinsic torque balance to within about 5 percent with two shafts at equal rpm, complex and heavy shafting, complex gears and seals, smaller diameter than single rotor, requires a cable guard, hub/tip ratio of about 0.5 satisfactory, excellent cavitation properties, blade-rate signal includes both blade-number harmonics and additional blade-interaction frequencies.

Accelerating Duct Configuration with Any of Above: highest efficiency at high loads, small diameter, heavy and complex mounting, may have assembly problems, duct provides cable protection.

6.1 Propulsor Concept Evaluation

The propulsor performance estimates which follow are based on the classical concept of separate estimates of wake fraction and thrust deduction combined with open water performance measurements of actual scale model propulsors. The estimates are improved by incorporating results from experiments and calculations carried out for the DOLPHIN Mark II propulsor design. A representative family of open propellers is the classical B-Series (or Troost Series) described by van Lammeren *et al* [13], high-load ducted propulsors have been examined by Oosterveld [14] and Caster [15], and C/R propulsors have been evaluated by van Manen and Oosterveld [16]. The Burrill and Emerson [17] Cavitation Diagram is used to confirm that excessive cavitation will not lead to thrust breakdown. The rotor is selected for minimum shaft horsepower (SHP) at 10 knots using a blade area that has acceptable cavitation characteristics. This determines pitch, shaft horsepower, and diameter for the propulsor and these determine a unique open-water performance curve.

No performance data are available for the preswirl propulsors so designs are based on a lifting-line model [18]. Powering and efficiency estimates are not made at the transit condition but should be similar to those of the other propulsors (Table 3).

Table 3 shows the powering and efficiency estimates for various candidate propulsors. Two conditions are considered: the 10 knot heavy tow design condition (equation 8b), and an in transit condition with the towfish housed at the keel ($D = 10V_k^2$). The 300, 600, and 900 rpm values are maintained at both tow and transit. It is possible to increase engine rpm to increase speed at transit, with no great loss of efficiency, if the heavy tow operating point is assigned to the low rpm side of the engine optimum efficiency peak. Transit speeds of at least 17 knots for all propulsor configurations are realizable in this way.

All propulsor power requirements reduce with rpm. In general, a 350 HP engine is required at 900 rpm, a 300 HP engine at 600 rpm, and a 250 HP engine at 300 rpm. The accelerating duct absorbs the least power while towing, but its efficiency is matched by the C/R and preswirl designs (Figure 7) at low rpm. Note that some C/R propulsor efficiency is lost in increased shaft friction.

		Propeller Geometry		Heavy Tow (10 kts)		Transit (at heavy tow rpm)		
Prop Config	rpm	Dia (ft)	EAR	SHP	η	SHP	V (kts)	η
Open	300	4.8	0.45	223	0.66	122	13.8	0.69
	600	3.5	0.60	266	0.55	162	14.9	0.64
	900	2.8	0.70	298	0.49	194	15.8	0.63
C/R	300	4.1	0.60	216	0.70	109	13.7	0.76
	450	3.4	0.65	247	0.62	140	14.6	0.72
	600	3.0	0.70	266	0.57	161	15.3	0.71
	900	2.4	0.95	305	0.50	205	16.4	0.70
Pre-swirl	300	4.5	0.35	212	0.69			
	600	3.2	0.50	266	0.55			
	900	2.5	0.75	302	0.49			
Accel Duct	300	4.3	0.70	206	0.70	no data base		
	600	3.0	0.70	240	0.60	229	17.8	0.77
	900	2.5	0.70	270	0.55	210	16.2	0.63

Table 3 *Propulsor powering and efficiency estimates. Shaft losses (2% for single-shaft and 5% for C/R) are accounted for. SHP is at the engine, efficiency η at the propeller. At transit, only about 0.5 to 0.7 of the tow horsepower is required at the heavy tow rpm. (EAR $\equiv$ Expanded Area Ratio.)*

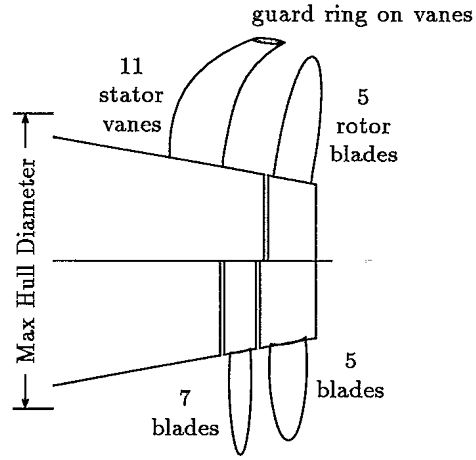


Figure 7 *The 300 rpm preswirl (top) and contra-rotating (bottom) propulsion configurations.*

6.2 Torque Imbalance

The open propeller design is also viable with a large diameter propeller, but generates substantial torque, as shown below. The torque generated by the preswirl stator vanes scales as vehicle velocity squared, while the preswirl rotor torque seldom exceeds that at the heavy tow condition, so stator torque is designed to be low at the tow condition and high during transit by equal amounts. The C/R design requires only small residual torque corrections.

Residual Torque Estimates (ft lb)

rpm	Open	Preswirl	C/R
300	4114	1430	206
600	2495	870	125
900	1955	725	98

The mast flap can negate these torque imbalances at 10 knots with deflections of 2, 0.7, and 0.1 degrees for the 300 rpm designs. At 17 knots, the deflections are 1/3 these values since flap force also scales as speed squared. These deflections are small and are unlikely to increase drag appreciably. However, ventilation and wave drag are a concern and are hard to predict, so full scale experiments should be conducted to quantify mast drag at small deflection angles at 10 and 17 knots.

There is a concern that tailfin downwash may reverse load the large stators of a low rpm preswirl propulsor, reducing tailfin effectiveness. This could be evaluated with a lifting line model such as was used to predict preswirl propulsor effectiveness [18].

6.3 Suggested Propeller RPM

Table 4 gives total weight estimates for the C/R and preswirl engine/transmission/propulsor configurations, including fuel requirements for 10 and 20 hour heavy tow missions. Open propulsor weight estimates are essentially the same as for the preswirl designs (preswirl stator vanes are assumed to be a light weight Kevlar composite). The table shows that despite fuel weight reductions from efficiency improvements, there is little change in total system weight because 1) engine weight does not decrease substantially with reduced power requirements and 2) transmission weight increases offset fuel weight decreases. In other words, for a given vehicle weight (same as size for an underwater vehicle), the reduced fuel consumption has little effect on endurance.

Nevertheless, increasing efficiency substantially increases endurance at the heavy tow design point because engine power requirements are reduced to a level at which continuous operations can take place. This level is just achieved with a 450 rpm propulsor, and so Tables 3 and 4 include C/R data at this point. This is the propulsor rpm suggested for the RMV.

Weights (lb)	900 rpm		600 rpm		450	300 rpm	
	C/R	p'swl	C/R	p'swl	C/R	C/R	p'swl
engine	1720	1720	1650	1650	1650	1650	1650
gearing	210	210	620	620	620	830	830
C/R gears	260	0	260	0	260	360	0
shafts	60	40	70	40	70	90	90
propulsor	70	90	120	130	180	220	230
fuel: 10 hr	1160	1190	910	910	840	770	760
fuel: 20 hr	2320	2380	1820	1820	1680	1540	1520
Total: 10 hr	3480	3250	3630	3350	3620	3920	3560
Total: 20 hr	4640	4440	4540	4260	4460	4690	4320
Engine Duty Rating*	inter-mittent		medium continuous		cont	continuous	

*Engine usage at 10 knot heavy tow design point.

Intermittent: 100% power 10% of the time, 90% power 90% of the time.

Medium Continuous: 100% power 50% of time, 90% power 50% of the time.

Continuous: 100% power 100% of the time.

Table 4 *Complete C/R and preswirl propulsion system weights. Fuel allotments are for the heavy tow design point. At 300 rpm, a two stage transmission gives the lowest weight, but reduces the efficiency below that of Table 2.*

6.4 Suggested Propulsor Options

A RMV using sternplanes for roll control will have limited capability for handling residual propulsor torque. The C/R propulsor, a 900 rpm version of which has already been successfully implemented in Mark II, is the best option in this case.

With a roll control capability that can negate large tow-off moments, such as the flapped mast, the C/R option is unnecessary. The simplest single shaft propulsor option is the open propeller, which should not compromise tailfin effectiveness. Adding an accelerating duct is an increasingly attractive option since new work is showing that optimized duct geometry may improve both tow and transit efficiencies at 450 rpm. However, duct/tailfin interaction and the effect of the duct on maneuvering are unknown. Further analysis is required to decide between these options.

7 IMPLEMENTATION

Many of the concepts presented here have undergone preliminary analysis to establish their feasibility. This includes structural analyses of the flapped mast and pod support strut, and the preliminary design of a pod deflection mechanism. Several pod winch mechanisms that self-align with the cable at tow-off angles up to 30 degrees have been devised.

8 CONCLUDING REMARKS

New RMV design options have been presented which address hydrodynamic control, drag, and propulsion issues. They are based on the DOLPHIN semi-submersible which has a good track record in these areas; an exception is roll control, where analysis suggests that a sternplane based roll control capability would be beneficial, a solution which is accomodated in the Mark II design.

The RMV designs rely on a new flapped mast for reduced drag and sufficient roll control to handle the large tow-off moments experienced in tight turns. This mast should be located close to the horizontal plane turning pivot point of the vehicle so that crossflows are minimized. Full scale testing is required to confirm the mast's effectiveness and to establish its drag characteristics. The design incorporates the engine exhaust in the base of the mast.

The RMV designs are based on existing DOLPHIN Mark I and Mark II layouts. New X- and Y-tail configuration options provide more efficient pitch and yaw control with reduced downwash interference. The existing foreplanes can be used for depth control, located as they currently are with an X-tail, or lowered to the keel with a Y-tail. Alternatively, depth control can be provided with an underslung pod, used preferably with a Y-tail and optionally with an X-tail.

The RMV2 design proposes a multi-use capability. RMV specific capabilities are contained in the payload bay and pod and can be replaced to adapt the vehicle to other uses. In addition to depth control, the pod provides ballast control and additional options for winch and cable drum designs.

If the RMV's adopt a roll control capability that can negate large tow-off moments, such as a flapped mast, then single shaft open and accelerating duct propulsors can be considered. Increasing efficiency by reducing propeller rpm to about 450 reduces engine power requirements and increases endurance.

RMV1 and RMV2 have no long appendages protruding horizontally or vertically downwards risking damage during launch, recovery, or shallow water operations.

The proposals in this paper are those of the authors and do not necessarily reflect the policy of DND or ISER.

REFERENCES

1. Walman, J.P., "Dolphin Warriors: Contrasting MCM Systems Show Promise," Surface Warfare, pp 16–17, January/February 1995.
2. Preston, J.M. and Latchman, S., private communication, July 1994.
3. Shupe, L. and McGeer, T., "A Fundamental Mathematical Model of the Longitudinal and Lateral/Directional Dynamics of the DOLPHIN Type Unmanned Semi-Submersible," DREP/RRMC Military Robotic Applications Workshop, August 1987.
4. Lyons, D.J. and Bisgood, P.L., "An Analysis of the Lift Slope of Aerofoils of Small Aspect Ratio, Including Fins, with Design Charts for Aerofoils and Control Surfaces," ARC R & M 2308, January 1945.
5. Pitts, W.C., Nielsen, J.N., and Kattari, G.E., "Lift and Center of Pressure of Wing-Body-Tail Combinations at Subsonic, Transonic, and Supersonic Speeds," NACA Report 1307, 1957.
6. Whicker, L.F. and Fehlner, L.F., "Free-Stream Characteristics of a Family of Low-Aspect-Ratio, All-Movable Control Surfaces for Application to Ship Design," DTMB Report 933, December 1958.
7. Dempsey, E.M., "Static Stability Characteristics of a Systematic Series of Stern Control Surfaces on a Body of Revolution," DTNSRDC 77–0085, August 1977.
8. Aucher, M., "Dynamique des Sous-Marins," Sciences et Techniques de l'Armement, 4^e fascicule, Paris 1981.
9. Spreiter, J.R. and Sacks, A.H., "The Rolling Up of the Trailing Vortex Sheet and Its Effect on the Downwash Behind Wings," Journal of the Aeronautical Sciences, vol. 18, no. 1, January 1951.
10. Feldman, J., "DTNSRDC Revised Standard Submarine Equations of Motion," DTNSRDC/SPD–0393-09, June 1979.
11. Blevins, R.D., "Applied Fluid Dynamics Handbook," Van Nostrand Reinhold Company Inc., New York, 1984, pp. 361-366.
12. Hoerner, S.F., "Fluid-Dynamic Drag," published by the author, Midland Park NJ, 1958.
13. van Lammeren, W.P.A., van Manen, J.D., and Oosterveld, M.W.C., "The Wageningen B-Screw Series," Trans. SNAME, Vol. 77, 1969.
14. Oosterveld, M.W.C., "Wake Adapted Ducted Propulsors," NSMB Publication No. 345, 1970.
15. Caster, E.B, "Ducted Propeller Designs for Improved Backing Performance," RINA Symposium on Ducted Propellers, 1973.
16. van Manen, J.D. and Oosterveld, M.W.C., "Model Tests on Contra-rotating Propellers," 7th Symposium on Naval Hydrodynamics, Rome, 1968
17. Burrill, L.C. and Emerson, A., "Propeller Cavitation: Further Tests on 16-inch Propeller Models in the King's College Cavitation Tunnel," Trans. NECIES, Vol 79, 1963.
18. Brockett, T.E. and Korpus, R.A., "Parametric Evaluation of the Lifting-Line Model for Conventional and Preswirl Propulsors," International Symposium on Propellers and Cavitation, Wuxi, China, 1986.

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