UNIVERSITY OF CAPE TOWN

DEPARTMENT OF MECHANICAL ENGINEERING

A TOWED SUBMERSIBLE

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A thesis submitted in partial consideration
for the degree of M.Sc.(Eng) . December 1972.

Contents.

1. Introduction.

One of the main problems in oceanographic research relating biological disturbances to chemical and physical parameters is that of combining observation and synopticity. At the moment parameters such as sound and plankton levels can be measured continuously, whereas geological and temperature readings are discrete (1,2). The difficulty is particularly acute for acoustic measurements. These give very detailed continuous records of underwater biological disturbances such as fish shoals or whale movements, and it is quite impossible to correlate these with data such as local temperature or salinity which may be crucial factors in their interpretation.

Consider the present way of measuring ocean temperatures. The instrument used is the bathythermograph, developed by Spillhaus in 1938 (3). This carries a liquid in glass metal thermometer or bimetallic strip which, operating through a linkage, causes a metal pointer to etch a trace on a smoked glass slide, which is itself being gradually moved perpendicular to this trace by a Bourdon tube or a pressure sensitive bellows. Since the depth of sea water can be assumed to be proportional to the pressure for depths down to 1000 m a continuous trace of temperature against depth is generated. In operation the bathythermograph is lowered to the predetermined depth where the impulse due to stopping the descent by sharply braking the towing cable causes the recording mechanism to start. The device is then rapidly hauled to the surface. There is a 90% temperature response in 0,4 seconds, and cycling takes about 15 minutes (3,5). In this way a discrete temperature profile of a section of the sea at a particular time is attained. The profile position will not be exact due to the ship's motion, either through drift or intended manoeuvring, and the temperature profile is usually time dependant as well.

One of the most interesting and significant regions of the sea is the thermocline (Fig. 1,1). This is the dividing

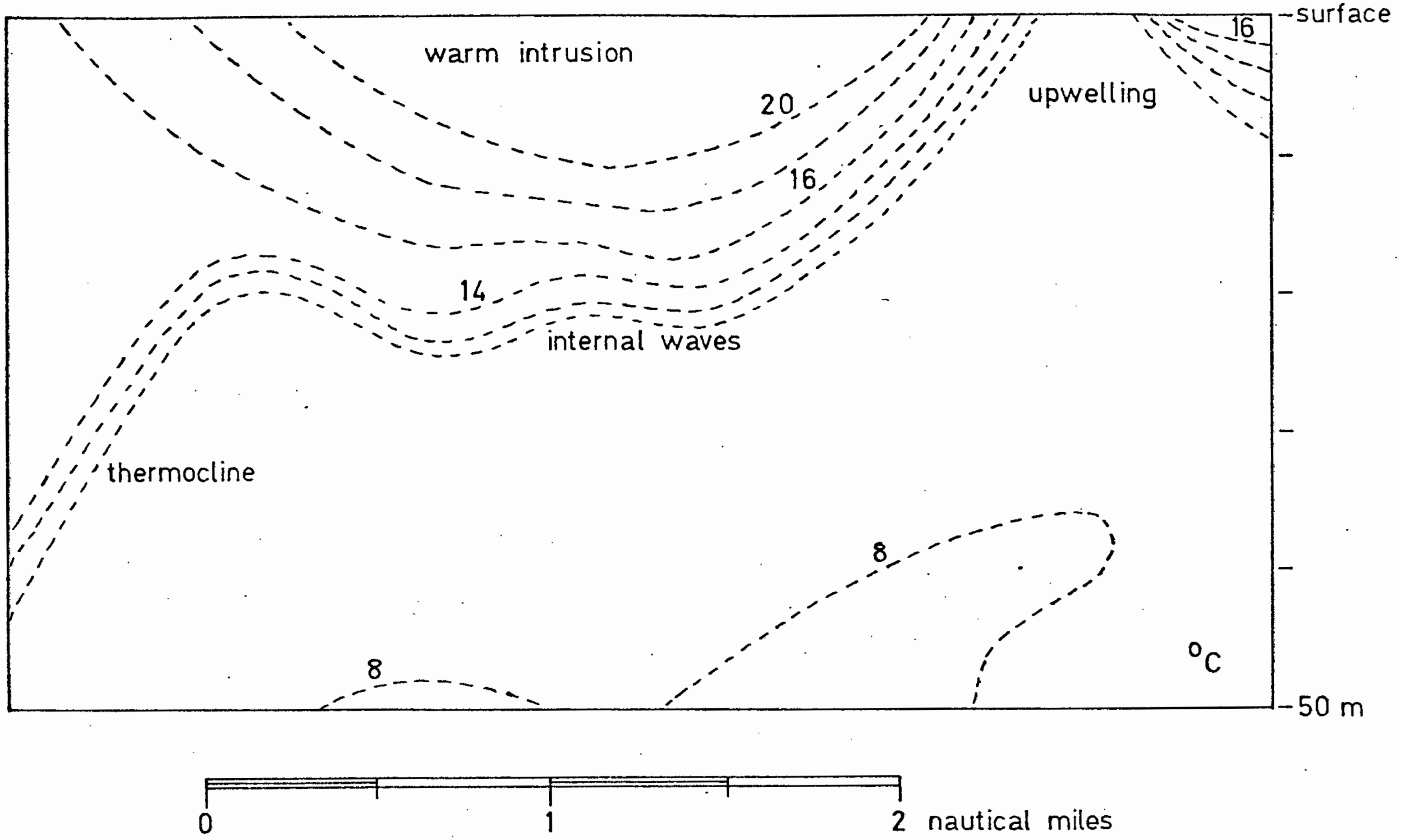


FIG 1.1 TYPICAL THERMOGRAPHIC SECTION

region between the cold ocean bottom currents and the warm surface layers, and since it delineates the interface between two liquids of different densities waves are formed, in this case of long period since the density difference is small. Off the Cape Coast of South Africa this temperature difference may be 15 C^{o}, and the wave length of order 1 km (5). To depict this region accurately a continuous temperature record is needed scanning both horizontally and vertically. This has been tried using a weighted cable carrying spaced thermistors feeding information to deck recorders (3). This is expensive, uses complex electronic equipment, and the multi-cored signal cable needs special winching facilities. Even when the cable is faired to reduce drag (4) a velocity gradient in the water or a variation in ship speed causes the cable profile to alter and the depths of the thermistors to change.

One way to continuously scan the sea would be to tow a submarine vehicle that would cycle vertically in the water between set depths. Its period of vertical oscillation would be short compared with the changes in parameters to be measured, in this case long internal waves. It would carry continuously recording depth and temperature measuring instruments; if possible storing the results on a magnetic tape or paper drum or else transmitting the data to the towing vessel. It could be designed so that it was relatively independent of ship speed and self contained so that it could operate unattended for long periods of time. Normal oceanographic surveys require regular stops for geological core samples to be taken or for sampling reversing bottles to be cast, and during these moments the vehicle could be hauled aboard and its recording sheets or power sources renewed as necessary. Accordingly a specification was determined in consultation with members of the Oceanography Department of the University for a suitable towed body (Table 1.1). Initially it was to carry temperature measuring devices, but it should have the possibility of extending this to carry instruments to measure pH, conductivity, etc..

The only published work at present concerning such a device is by Glover (2), who is developing an undulating plankton recorder for long distance towing with an oscillating wave length of 20 km. It is not very appropriate to make direct comparison with Glover's work since there local instabilities are damped out during the long, slow oscillations, whereas for a device with a much shorter wave period such local perturbations crucially affect the performance.

A device with a relatively short undulation period for intense data collection would be a major advance in oceanographic recording techniques, and one that could have a wide varity of commercial applications.

Table 1.1 Specifications for Design.

Depth range	0 - 50 m
Pressure range	1 - 6 bars
Temperature range	5 - 25 °C
Oscillation wavelength	About 200 m
Ship speed	7 knots (3,5 m/s)
Accuracy of position	
i) Vertically	3 m
ii) Horizontally	Unimportant
Weight in sea water of recorder	4,5 kg
General comments	It must be safe and easy to handle on the aft deck of the T.B. Davie, the research vessel of the University of Cape Town, and should have an overall length of less than 1,75 m.

Corrosion.

Corrosion is the gradual transformation of a metal into one of its compounds due to the presence of particular nearby chemicals. The main corrosion mechanisms are direct chemical attack and electro-chemical attack.

For electro-chemical attack, or galvanic action, to occur there must be a continuous electrical path between the electrodes. Seawater is a good electrolyte and readily sets up a galvanic cell between dissimilar metals or between a metal and a local patch of its oxide. Corrosion occurs most readily in metals that are far apart in the electro-chemical series for seawater, given in Figure 2.1, with the anodic region becoming pitted and cathodic areas masked with hydrogen. If the hydrogen is swept away the local corrosion continues, but if it remains another cell is set up elsewhere on the material and general corrosion results. Galvanic corrosion can be minimised by not using dissimilar metals or by ensuring that the critical components are the more cathodic and are protected by plating or painting. Many sea going vessels have sand blasted hulls to remove any post-fabrication pitting and then use a sacrificial anode of a Zn-Mg-Al composition which renders the metal cathodic and protects it from attack.

Purely chemical attacks occur in a number of ways, of which the most common and least understood is stress corrosion. Here certain chemicals dissolve the surface layer of the material at selective spots, such as grain boundaries, and cause corrosion cracks to form. This occurs only under conditions of tensile stress and will affect a submarine vehicle at areas such as bolted joints, welds and heat treated regions.

Fatigue is important because metals do not have an apparent fatigue limit in seawater. In submerged vessels the structural fatigue falls into the low cycle range, less than 10 000 cycles, and the life is more dependent on the design and fabrication methods than the materials used.

Absor~~b~~ption of hydrogen and water into steels causes embrittlement and consequent brittle fracture. This danger is increased near to the sea bottom and at estuaries, where the oxygen content of the water usually decreases. It is usually confined to martensitic and precipitate hardened steels, and is not a problem with austenitic steels or non-ferrous alloys except for certain body centred cubic alloys of aluminium.

Besides elctro-chemical and direct chemical corrosion ocean appliances are attacked by marine organic life. Small crustaceons such as barnacles disturb the smooth flow of water around a carefully profiled body and cause pitting by encouraging a local deficiency of oxygen. Copper based anti-fouling paints are toxic to them but have to be regularly renewed, and even these do not discourage curious sharks or dolphins.

Other failure mechanisms such as cavitation pitting or waterline rusting are not considered here, for although they occur at sea they are usually associated with surface vessels or high speed submarines.

Non metallic substances are not prone to the corrosion problems outlined above, except organic attack, and so are particularly attractive for use in a marine context. The common substances are wood, plastic and glass, which in general are not as strong as metals and have other serious disadvantages which preclude their general use as major structural materials. Wood forming is not suitable for intricate parts, requires careful moulding or caulking and must be continuously maintained. Glasses such as 'Pyrex' have been used to effect in research machines, but are expensive, have negligible tensile strength and are highly notch sensitive. Even when in a purely compressive role they can only be used in conjunction with a titanium lattice which brings further material difficulties. Plastics have the greatest potential, so far unrealised commercially except for fibreglass which is readily available, cheap and of low density. Properties of the more commonly available materials are discussed in References 6, 10, 11.

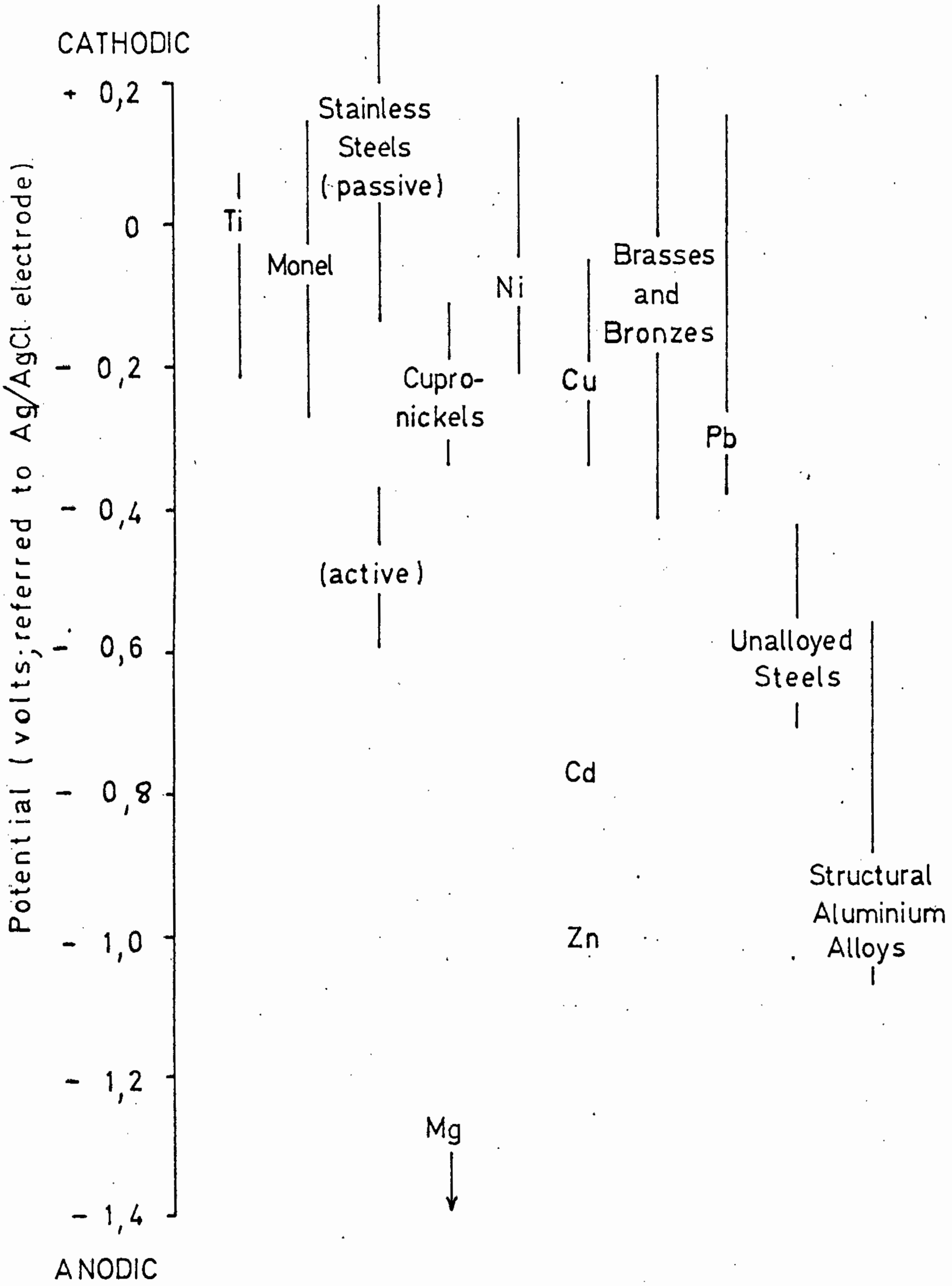


FIG 2.1 ELECTROCHEMICAL POTENTIALS IN SEAWATER

Towing.

There is little readily available literature on sub-surface towing. Surface towing is characterised by the distance between the vessels due to their lack of manoeuvrability, especially when the towed vessel cannot turn or stop independently. With towed submersibles there are the added dangers of the cable fouling the propellors of the ship and the submersible being disturbed by its wake. Both of these argue for a long tow. Small submersibles are affected by ocean currents, as described previously, but this drift can be detected and compensated for by a suitable control system. The greatest difficulty occurs when towing near to the surface. A ship speed of 7 knots is equivalent to the speed of fast ocean waves, so with a following sea a towed body would be in a metastable state. For a towed submersible this effect will be present to a depth of about 8 m, depending on the sea state.

2.2 Design Philosophy.

A vessel that does not have its own propulsion unit must rely on the forces generated by the interaction with its environment for motion. For a body moving in a fluid these impressed forces are due to differencies in mean density between the fluid and the immersed body and to the hydrodynamic effects, lift and drag, of the fluid flow around the body. For a towed vehicle there is also the force exerted by the towing vessel. Any design must utilise some of these forces, and this section looks at the merits of each.

Forces Due To The Towing Vessel.

The effect of these alone is to raise the towed body to the surface and keep it there. If the body has some weight and drag it will stabilise at a depth which depends on its weight and shape, the length of the towing cable and the boat speed. Changing the weight of the body whilst in the water is equivalent to changing its mean density, a hydrostatic effect, and changing its shape induces a change in hydrodynamic forces. Both of these cases are examined later. Continuously varying the boat speed to change the depth is impracticable due to the difficulties in manoeuvring the T.B. Davie at sea, and undesirable since the specifications implied a design that could operate without the need for continuous control from the ship. Varying the cable length seems a clumsy solution. It involves constantly operating winches on deck with at least 50 m of cable being wound and unwound every cycle. It does not utilise the ship's motion except to increase the drag and the power consumed. However it is a simple method that can be continuously monitored, with the instruments packed into a pressure vessel and lowered over the side on a weighted cable.

Hydrodynamic Forces.

Any increase in the lift or drag on a towed body will increase the force tending to raise it to the surface. Conversely a decrease in these quantities will cause it to sink. The properties can be altered by changing the profile of the body in the water or altering its attitude, either by moving attached hydroplanes or by adjusting its mass distribution.

Most constant depth submarine devices, for instance the paravane, prepare their equilibrium depth by altering their fins before being launched. However a moving body has a number of inherent disabilities that must be overcome before it can be successfully used as a precise research instrument. The most important of these is stability. If a towed vehicle is swept sideways by a cross current or a region of underwater turbulence it has no automatic righting system and will continue to move in a direction dependant on the final orientation of its fins in relation to the surface. If it has been rolled onto its side it will continue to move sideways unless a stabilising system is incorporated into the control circuitry. Also a body that relies on hydrodynamic forces alone has to be carefully streamlined for slight changes in profile to have a noticable effect. A preliminary design was evaluated including a stability control but was eventually discarded due to its increasing complexity and cost. Details of this design are given in Appendix A.

Hydrostatic Forces.

If legend is to be believed the principle of floatation has been recognised by scientific man since Archimedes (c. 400 B.C.). More recently it has been used for oceanographic research work by M. Picard in the bathyscape 'Trieste' (14) and for a free vehicle it has many advantages. However heavy the scientific package it can always be rendered effectively weightless in seawater by attaching a suitable volume of less dense material, and vertical motion assured by a slight change in weight. Its disadvantages are that the float is usually large and so vunerable to cross currents, and that most low density materials are noticably compressible and lose buoyancy as the bathyscape descends. For a towed vessel the large float would mean that there would probably be a correspondingly high drag, increasing the towing load and the moment of the force towards the surface.

Any design would basically consist of a large towed drum carrying an air cylinder and recording instruments inside but open to the sea at the base. The only controls would be two valves controlled by a depth gauge. One of the valves would be used to open a port at the top of the drum to allow air to escape, water to pour in the base and the body to sink, when the valve would close. At the appropriate depth the other valve would open to flood the drum with air, force out the sea and cause the body to rise. The advantages of such a system are clear. The shell need not be a pressure vessel and so could be manufactured cheaply. The control would be simple and could be encapsulated in a small pressure vessel inside the shell. Even if the entire structure was rolled over and the air spilled out it would right itself due to a towing eye at the top giving it a pendulum like stability, fall to its control depth and refill with air. This motion would be noted on the recorded and allowed for when interpreting the data.

The final design took the best points from the hydrostatic and hydrodynamic cases, and is considered in a general outline in the next section.

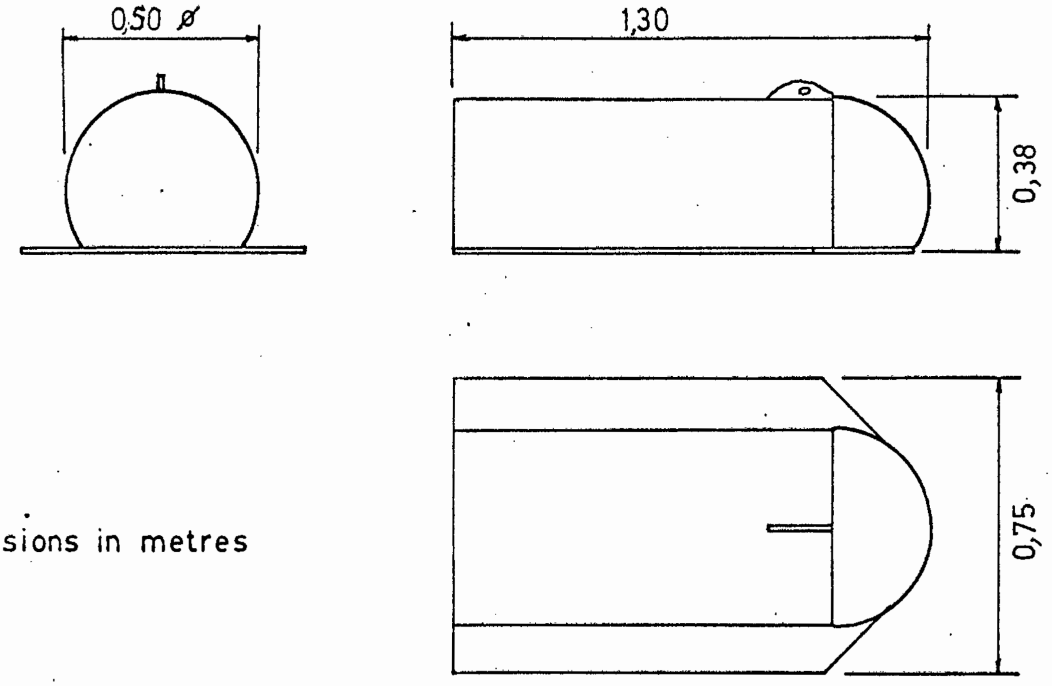
2.3 General Design.

General Arrangement.

The final design uses both hydrostatic and hydrodynamic effects for motion. Basically it is a hollow cylinder closed at one end by a disc and with the other capped by a hollowed hemisphere. The whole is cut along a plane parallel to the common axis and at a distance from it at the closest point of half the cylinder radius. Attached along this cut is a perforated plane base shaped to extend either side of the body into a pair of fixed fins (Fig. 2.2) A towing eye is attached to the top.

Internally the vehicle is divided into two compartments by a transverse watertight bulkhead (Fig. 2.3). The aft section houses two scuba diving aqualungs connected to a common manifold whose exit is through a solenoid valve to a pipe that is sealed into the bulkhead and passes through into the forward compartment. This forward chamber has a vent in the roof and air can leak out via another solenoid valve. The control circuitry and batteries for the valves are in watertight containers in the stern compartment, which also houses the recorders. The aqualungs, batteries and recorders are conveniently mounted for easy renewal.

The structure is built around a galvanised low carbon steel plate mounted vertically along the centre of the aft section. The towing eye and all the equipment are bolted to this to ensure that in the event of some disaster that damaged the shell of the vehicle and made it unfit for further service the expensive items of equipment could be recovered and would not be washed away. Except for two stiffening and locating hoops the rest of the structure is made from moulded fibreglass pieces. The curved part of the stern section is made in two pieces, covering each side with a separate cowling that can easily be removed for access. The whole structure can be lightened in the water if necessary by inserting shaped wooded or polyurethane pieces. Thus the vehicle is cheap to build and simple to construct.



Dimensions in metres

FIG 2.2 GENERAL VIEW OF VEHICLE

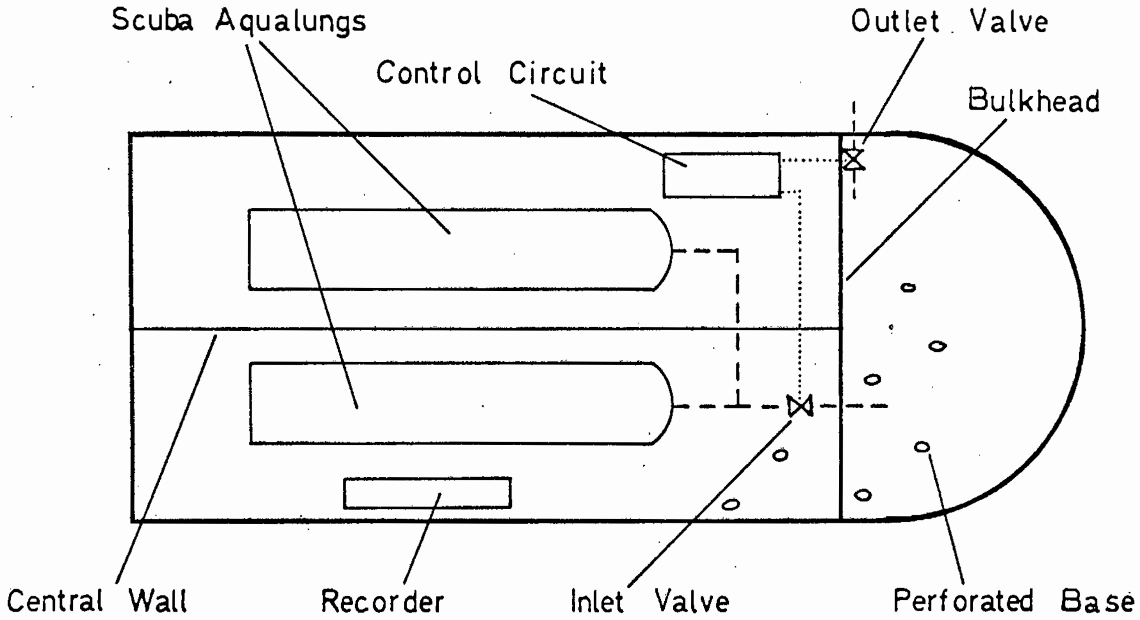


FIG 2.3 SCHEMATIC VIEW OF VEHICLE INTERIOR

Mode of Operation.

When on the surface the weight of the body will cause it to sink. The base is perforated so that it rapidly fills with water, and the valve in the roof of the forward section opens for an interval to allow the air to escape. The position of the towing eye is such that when the nose is flooded the vehicle takes a nose down attitude and the resulting negative lift facilitates descent. When at the lower specified depth the valve controlling the air cylinders opens to admit air to the forward section, flushing out the water and enabling the nose to lift. The effective weight of the body is reduced by this air, and with the lift on the fins due to the altered incidence it rises. On reaching the upper control depth the cycle is repeated.

Due to its particular mass distribution the body is inherently stable in descent provided that the centre of lift is always astern of the towing eye, but during ascent the stability position is more complex. The forces on the vehicle are its weight, its dynamic lift and drag and the force at the towing eye due to the cable. As indicated in Section 3 the angle to the horizontal of the cable at the eye depends on the resultant horizontal and vertical forces on the vehicle. If this angle is less than the inclination of the body to in the water then it will probably roll onto its back and be stable upside down. The exact point of change from one stable state to the other is difficult to determine since the vehicle's inclination in the water will be uncertain. If the depth recorder showed that irregular behaviour was occurring in the upper regions of the path then the towing position would have to be changed to alter the force distribution.

The time of oscillation of the body will depend greatly on its effective weight. This is considered quantitatively in Section 3. Too light a body will not reach the desired depth, whilst if it is too heavy the air consumption rate will be too great during the ascent. The developed analysis can only be considered as a guide to the best weight to use, and the final corrections must be left for the sea trials. Associated with both the weight of the body and its attitude is the rate of flow of air into the nose at the lower level of descent. The inertia of cable and body cause a response lag to such weight changes, and if the flow is too fast so that too much air is released into the nose the ascent will be rapid and the rate of vertical oscillation will correspondingly increase. If the flow rate is too slow it might not even keep abreast of the increasing compression of the air already in the nose. Other important variables are the trim of the vehicle, including the position of the towing eye, and the amount of towing cable used. Like the body weight both of these parameters can be assessed to some extent by calculation, but the refinements for optimum performance can only be made during sea trials.

3. Analysis.

This section makes an analytical examination of the performance of a towed body; its stability depth and descent time and their dependence on such factors as body weight and cable length. The first part of the analysis determines the profile in the water of a cable towed at a constant speed with various vehicles attached. This is the 'Steady State' analysis. Different cable/vehicle combinations give different cable profiles and stability depths for the towed vehicle. Correspondingly, once the desired lowest depth has been specified a cable/vehicle combination can be chosen whose stability depth is equal to, or slightly greater than, the desired depth, when the change in vehicle weight to raise it to the surface, and hence the amount of air used each cycle, will be a minimum.

Having decided the cable length and vehicle weight the second part, the 'Dynamic' analysis, estimates the descent time. Both analyses are expressed using non-dimensional parameters whose nomenclature is given in Table 3.1.

Table 3.1 Nomenclature

M	Mass of towed vehicle in sea water.
A	Effective area of the vehicle.
Cd_1	Drag coefficient for the vehicle.
u	Ship speed.
ρ	Density of sea water.
w	Mass/unit length of the cable in sea water.
d	Cable diameter.
Cd_2	Cable drag coefficient.
L	Cable length.
s	Distance along the cable from the ship
θ	Inclination of the cable to the horizontal.
T	Tension in the cable at s.
T_o	$T = T_o$ at $s = L$
ϕ	$\theta = \phi$ at $s = L$
g	Gravitational constant.

$K1 = 0,5 \cdot \rho \cdot Cd_1 \cdot A$

$K2 = 0,5 \cdot \rho \cdot Cd_2 \cdot d$

$I = (M + w \cdot L/3) \cdot L^2$

$P1 = w \cdot L/M$

$P2 = Cd_1$

$P3 = A/L^2$

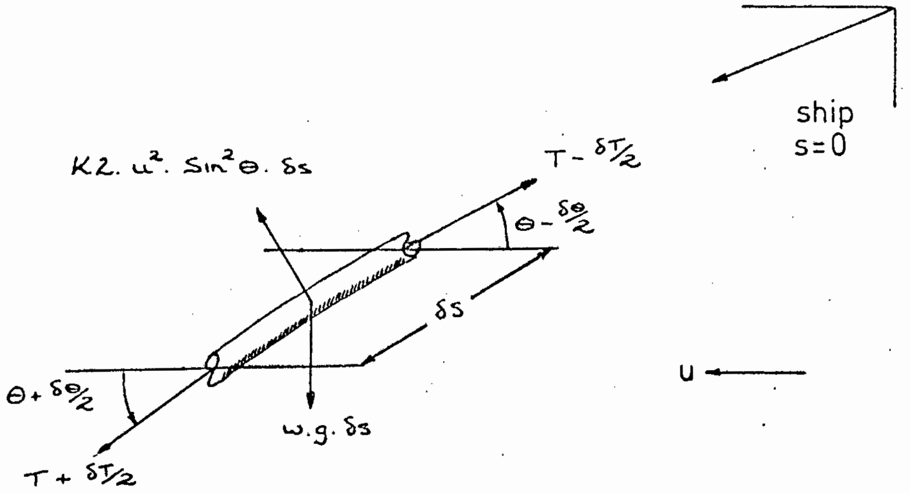
$P4 = Cd_2$

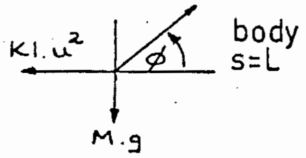
$P5 = d/L$

$P6 = u/(L \cdot g)^{1/2}$

$P7 = \rho L^3/M$

3.1 Steady State Analysis.





Assume the forces on a cable element shown. The assumption of the force $K2 . u^2 . \sin^2\theta . \delta s$ perpendicular to the element is a close approximation to the actual experimental result unless θ is very small (14).

Resolving perpendicular to the element; for equilibrium

$$K2 . u^2 . \sin^2\theta . \delta s - w.g.\cos\theta . \delta s - 2T . \sin\left(\frac{\delta\theta}{2}\right) = 0$$

If $\delta\theta$ is small compared with θ

$$T\frac{\delta\theta}{\delta s} = K2 . u^2 . \sin^2\theta - w.g.\cos\theta \qquad (1)$$

Resolving along the cable; for equilibrium

$$\left(T - \delta T/2\right)\cos\left(\frac{\delta\theta}{2}\right) - \left(T + \delta T/2\right)\cos\left(\frac{\delta\theta}{2}\right) - w.g.\sin\theta . \delta s = 0$$

$$\therefore \frac{\delta T}{\delta s} = - w.g.\sin\theta \qquad (2)$$

The boundary conditions for these equations are

$s = L \qquad T = T_0 \qquad \theta = \phi$

Where $T_0^{\,2} = (K1 \times u^2)^2 + (M \times g)^2$

$\tan\phi = (M \times g) / (K1 \times u^2)$

The equations can be solved analytically, but the resulting expression is cumbersome, and since it relates s and θ, has to be integrated numerically to give the cable profile in x-y co-ordinates. Accordingly the entire analysis was performed numerically, starting from the conditions at s = L and deriving T and Θ after a step length δs from equations 1 and 2. Knowing the inclination of the cable at points along its length an integration using Simpson's Rule enabled the vertical and horizontal co-ordinates describing the cable shape to be found. An account of the computer program used is given in Appendix 2.

3.2 Dynamic Analysis.

The system to be analysed consisted of the submersible and flexible cable. In the dynamic case these sink from the surface to the design depth, and since the cable is assumed perfectly flexible the system has an infinate number of degrees of freedom.

To analyse it gross assumptions must be made:

1) The cable lies in the plane of motion, i.e. that the system occupies a two dimensional space at any instant.

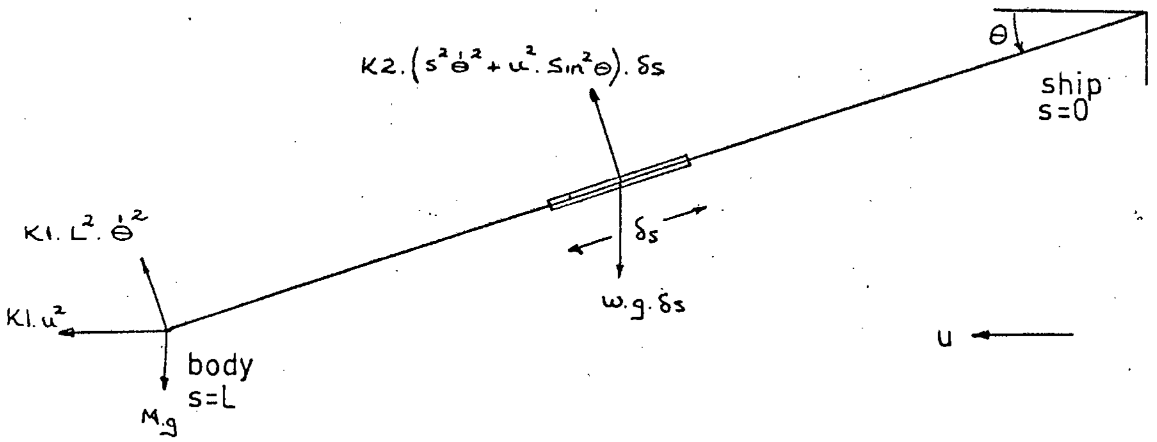
2) Forces across the plane of motion are negligible.

3) The towing cable is a rigid rod with the drag and weight properties of the cable but freely hinged at the boat and at the body. Figure 3.1 shows that for typical configurations the cable profile is approximately linear for the steady state case, and our assumption is that this profile is valid at all moments in the descent. If the dynamic analysis is performed for a long period of time the vehicle will reach a steady state stability depth. Table 3.2 compares this depth with one obtained using the steady state analysis for different cable/vehicle configurations, and it is clear that although the analyses are different their results are close enough to justify the linear assumption as a first order approximation.

4) As in the previous case the drag coefficients for cable and vehicle are assumed constant over the narrow range of Reynolds' Numbers considered, and for simplicity it is assumed that the drag coefficient for the body is the same for all flow directions. Reference to the force diagram will show that this is not such a gross assumption as at first would seem since the two forces that include the drag coefficient are almost equiaxial.

Table 3.2 Steady State - Dynamic Correspondence.

Mass of Towed Vehicle (kg)	Stability Depth.	
	Dynamic Case	Static Case
u = 10 knots	$Cd_1 = 1,5$	
12,5	0,0909	0,0909
25,0	0,1096	0,1096
37,5	0,1269	0,1268
50,0	0,1431	0,1428
u = 7 knots	$Cd_1 = 1,5$	
5,0	0,1292	0,1294
15,0	0,1529	0,1532
25,0	0,1748	0,1749



Consider a cable element length δs as shown. Then, as in the steady state case, there will be a drag force perpendicular to the element of $\frac{1}{2}.\rho.Cd_2.d.u^2.\sin^2\theta.\delta s$ due to the motion of the water alone. Since the cable has an angular velocity $\dot\theta$ the total drag force will be $K2.(s^2\dot\theta^2 + u^2\sin^2\theta).\delta s$ where $K2 = \frac{1}{2}.\rho.Cd_2.d$ .The moment about the stern of the ship due to the resultant forces on the element is $s.(w.g.\cos\theta - K2.(s^2\dot\theta^2 + u^2\sin^2\theta))\delta s$ and the total moment due to the cable is

$$\int_0^L (w.g.\cos\theta - K2.(s^2\dot\theta^2 + u^2\sin^2\theta)).s.\delta s$$

At any instant $\theta, \dot\theta$ are constant and the moment is

$$w.g.\frac{L^2}{2}.\cos\theta - K2.\dot\theta^2.\frac{L^4}{4} - K2.u^2.\frac{L^2}{2}.\sin^2\theta$$

The inertia of the cable and vehicle is $I = (M + w.L/3).L^2$

Taking moments

$$I\ddot\theta = (M.g.\cos\theta - K1.(u^2.\sin^2\theta + L^2.\dot\theta^2)).L + w.g.\frac{L^2}{2}.\cos\theta - K2.(u^2.\frac{L^2}{2}.\sin^2\theta + \frac{L^4}{4}.\dot\theta^2)$$

i.e.

$$I\ddot\theta + (K1 + K2.L/4).L^3.\dot\theta^2 + K1.u^2.L.\sin\theta + K2.u^2.\frac{L^2}{2}.\sin^2\theta = w.g.\frac{L}{2}.\cos\theta \qquad (3)$$

The equations can be solved using a Runge-Kutta method for small time increments. The program in Appendix 2 was written to solve this equation and gives an output of depth against time for any specified system.

3.3 Non-Dimensional Analysis.

In non-dimensional analysis the basic dimensions of mass, length and time are usually redefined in terms of variables of the system being analysed that are unlikely to change rapidly. In our case ρ and g are obviously such variables. Non dimensionalising an equation is equivalent to scaling it, however, and if the equation is to be solved by numerical methods the scaling must result in coefficients that are of comparable size, lest small coefficients result in terms being ignored. The first choice of basic variables was L, ρ, g but this gave a dimensionless group of $M/\rho . L^3$ which for most cases was very small, leaving the equations insensitive to changes in M. Trying u, ρ, g or A, ρ, g both gave workable values for the groups but very unequal coefficients in the equations. The choice of basic quantities that gave an equation responsive to all variables was M, ρ, L. Since M is a variable that is often changed this choice is restrictive, but will give the most accurate results. The programs written for both steady state and dynamic analyses used the dimensionless groups indicated in Table 3.1

3.4 Results of the Analysis.

Figure 3.1 shows some typical cable profiles. These are identical to those found by Glaubert (14) whose work was used before the development of computers to give standard design parameters.

Figure 3.2 illustrates the importance of towing speed on any towed body. From the wind tunnel tests (Appendix C) the drag coefficient of the vehicle was found to vary between 0,85 and 1,50 for the angles of incidence chosen, and the specifications show that speeds within 3 knots of the 7 knot design speed must be considered. Doubling the speed quadrouples the drag, and the stability depth varies accordingly. If the towing ship cannot maintain the design speed or the subsurface currents persist the vehicle depth can be altered by hoisting inboard and changing the mass, by adding weights or floats, or by simply varying the length of the towing cable. Figure 3.3 shows the effect of drag variation on the descent time.

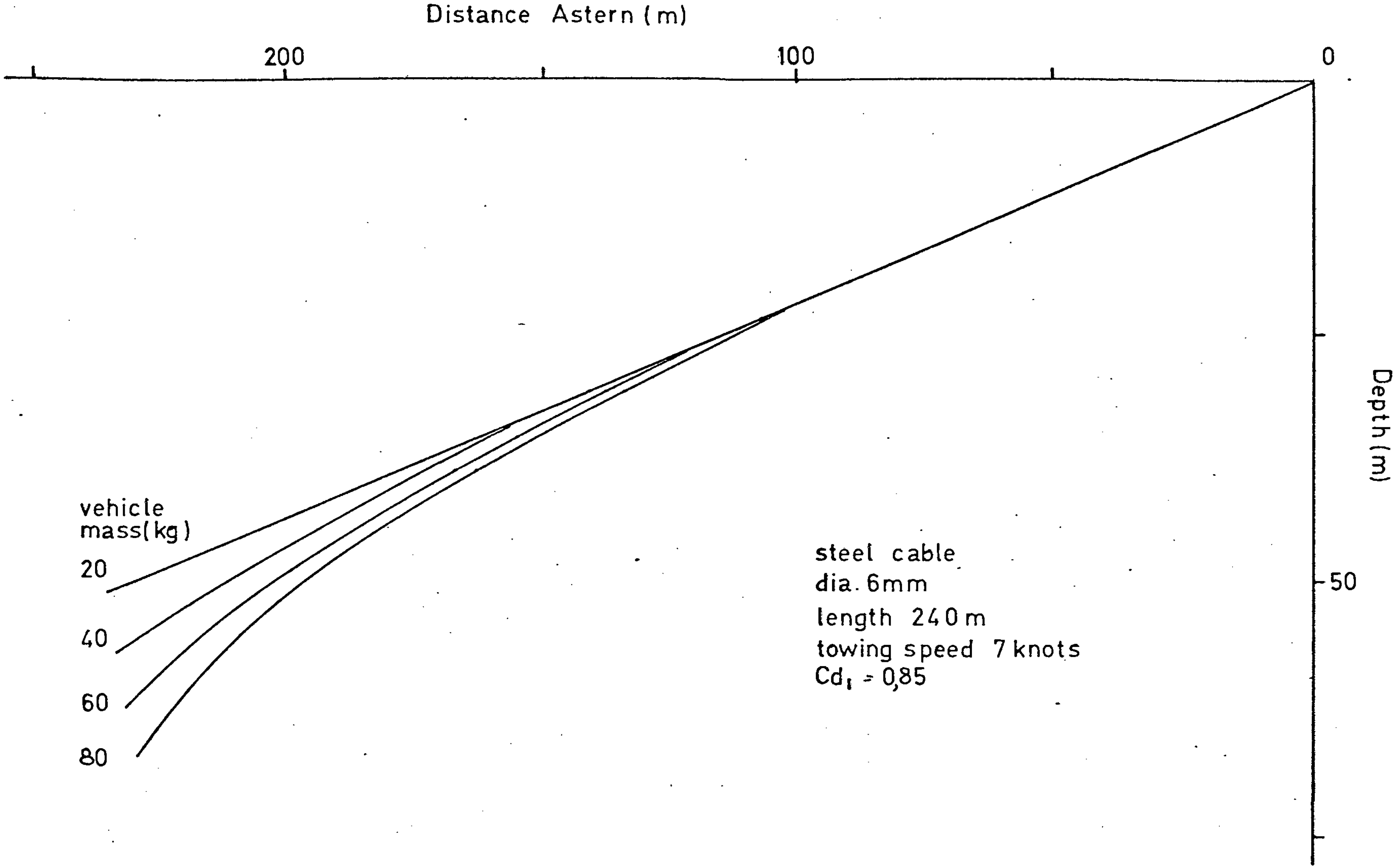


FIG 3.1 TYPICAL CABLE PROFILES

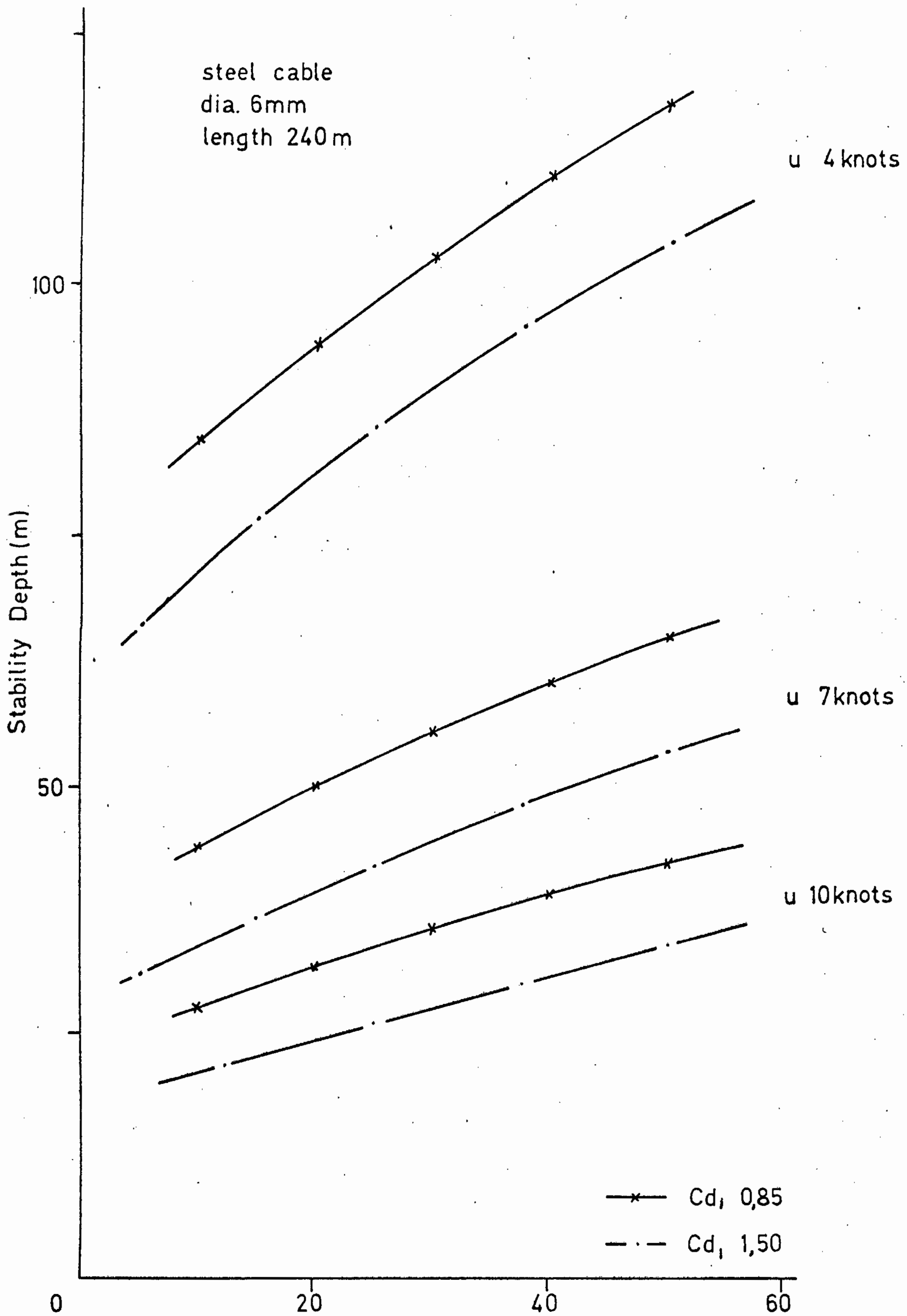


FIG 3.2 CHANGE IN DEPTH WITH VEHICLE DRAG

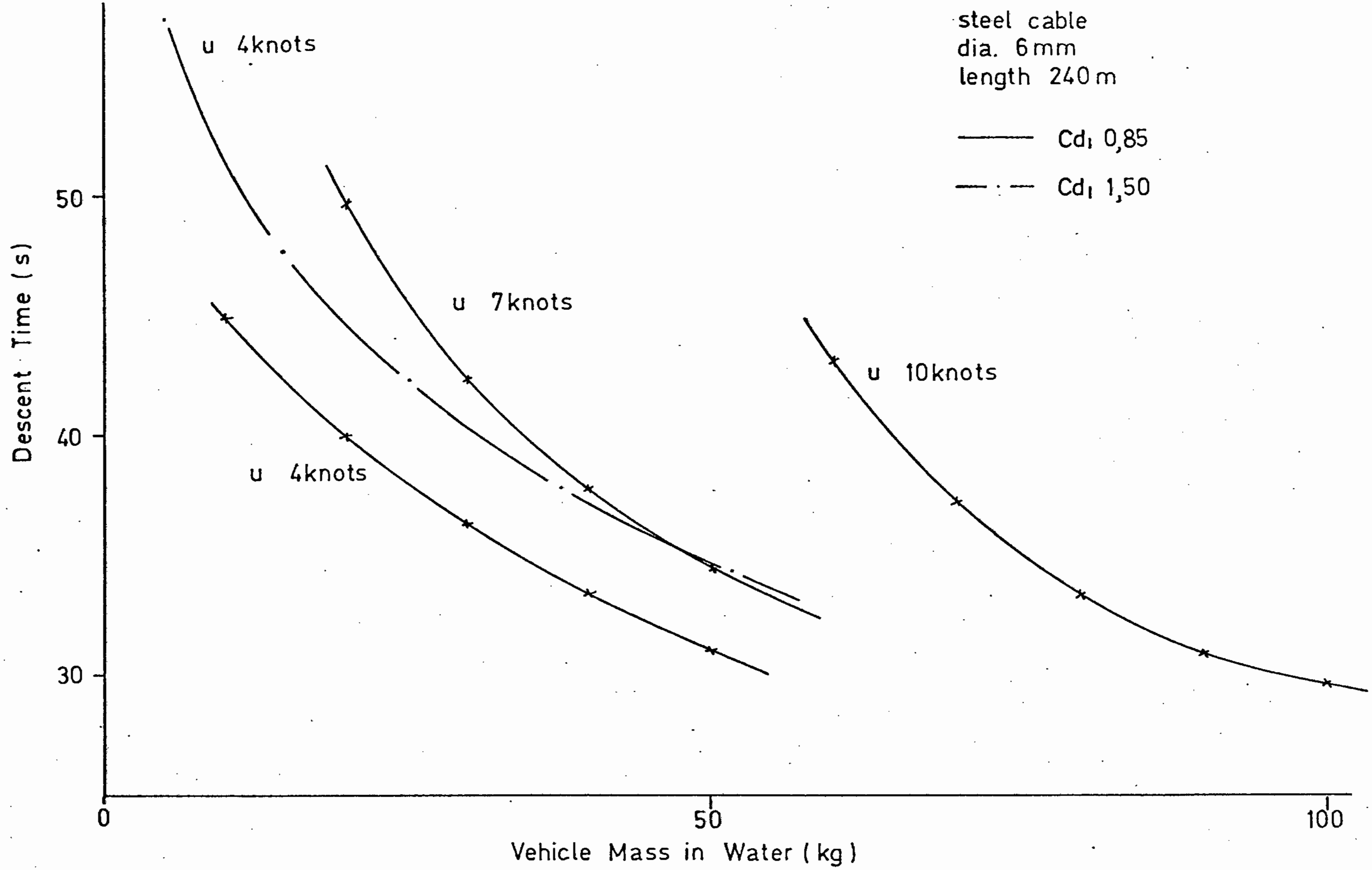


FIG 3.3 CHANGE IN DESCENT TIME WITH VEHICLE DRAG

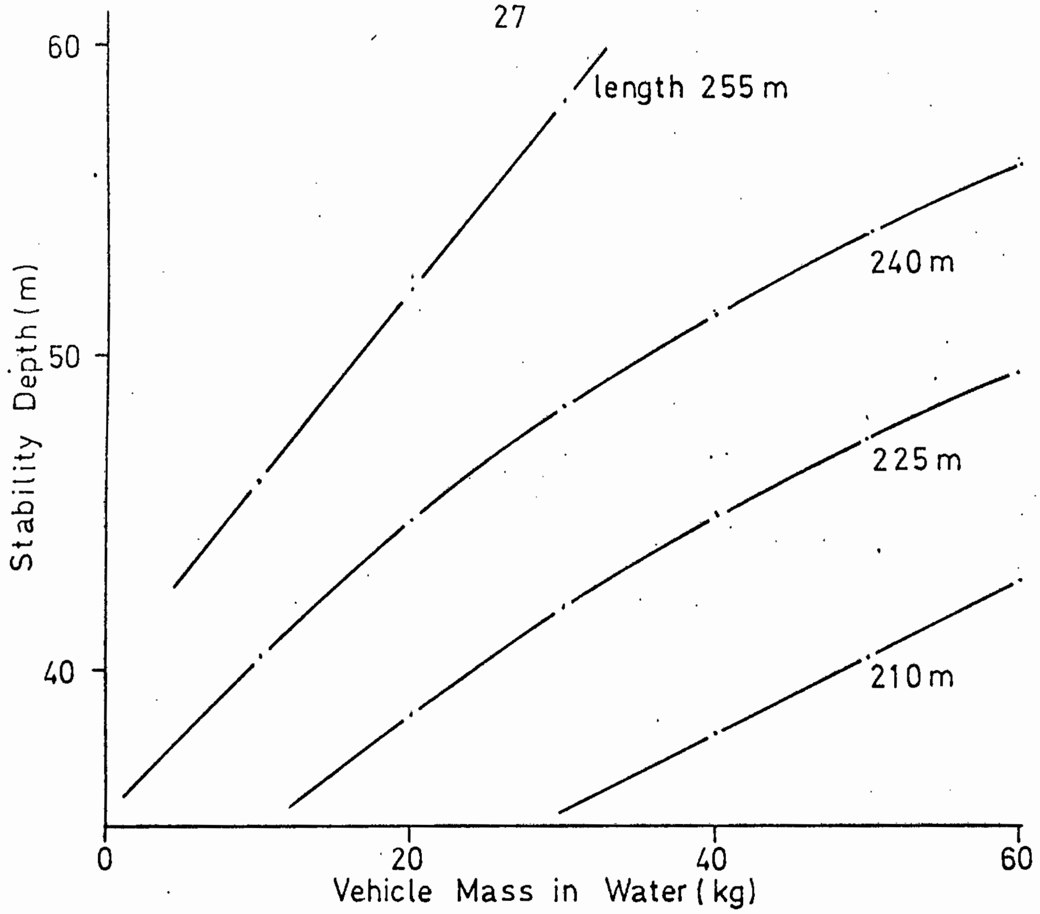


FIG 3.4 CHANGE IN DEPTH WITH CABLE LENGTH
6 MM DIA. STEEL CABLE

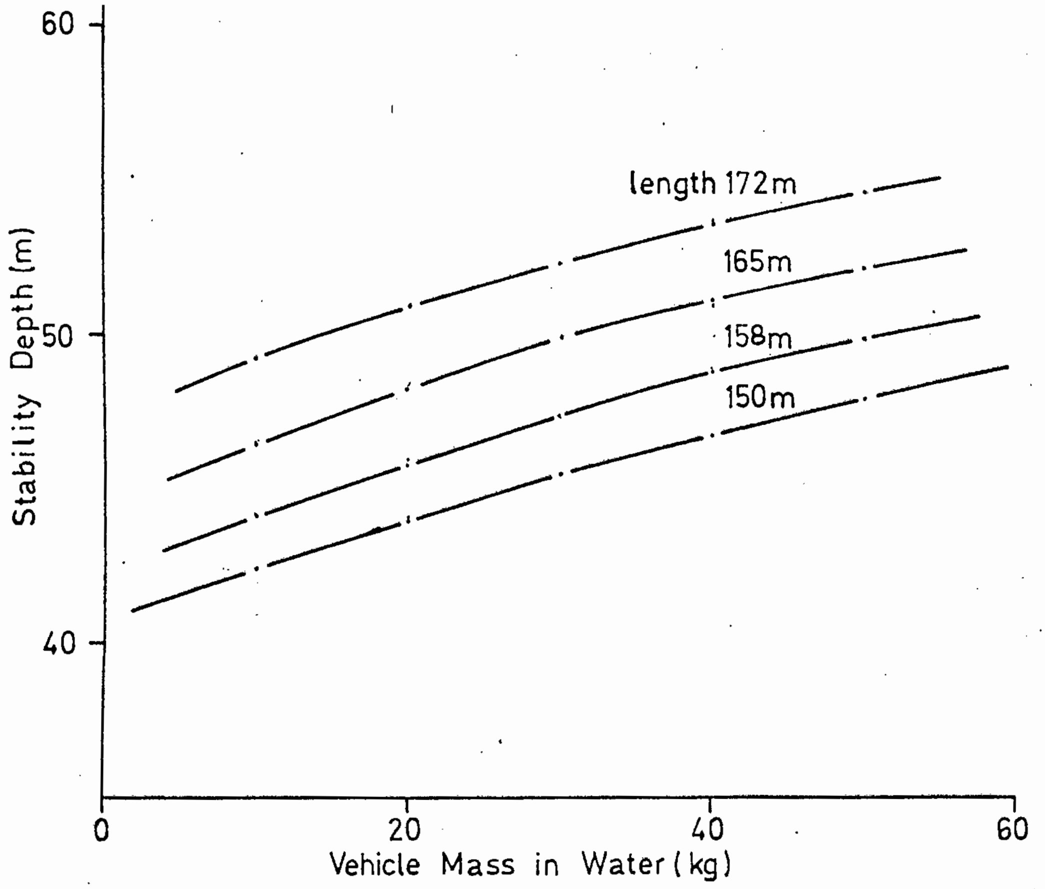


FIG 3.5 CHANGE IN DEPTH WITH CABLE LENGTH
12MM DIA. STEEL CABLE

Figures 3.4 and 3.5 are the most directly practical graphs drawn. The two towing cables available on the T.B. Davie are steel, of 6 mm and 12 mm diameter, and these graphs show the stability depths attained using different cable lenghts and vehicle masses in each case. It is clear from them that changing the actual body mass does not change the stability depth very greatly. For instance, if a vehicle of 40 kg mass towed by a 240 m long 6 mm steel cable reduced its effective mass to zero by a sudden ingress of air its stability depth would change from 53 to 35 m. This illustrates the importance of the hydrodynamic lift on the body, as the lift forces must raise both the vehicle and 240 m of cable to the surface. If a cable of polypropylene was used this problem would not arise since polypropylene has neutral density in sea water. It is weaker than nylon but does not have its elasticity, making winching easier.

3.5 Vibrational Stability.

The best researched area analogous to that of the towed submersible is that of the tethered balloon. As in the wind tunnel tests (Appendix C) conducted on a scale model of the towed vehicle the effects of changing Reynolds' Number can be allowed for, and a tethered balloon is an identical system to the towed submersible if the tethering cable is considered to have negative mass. Neumark (17) analysed the stability of balloons and found that there were two modes of instability in directions in the plane of the cable and one mode of lateral oscillation. This lateral and one of the longitudinal modes had a frequency approximately that of a simple pendulum of length equal to the towing cable, and oscillations of large amplitude in a balloon, where viscous damping effects were small, but for a submersible the viscous forces are much greater and large oscillations are unlikely to occur. For a weighted cable of length between 150 and 250 m the natural frequency for pendulum instability would lie between 15 and 20 s. The desired cycle time for this design is of order 1 to 2 minutes, so forced resonance is improbable.

The other mode of instability is in the direction of the cable, and is unlikely to be important as it has high frequency and small amplitude. The only source of forcing would be from vortex shedding from the rear of the towed vehicle, a phemomenon that is difficult to calculate for on a complex profile without extensive testing.

4. Detailed Design.

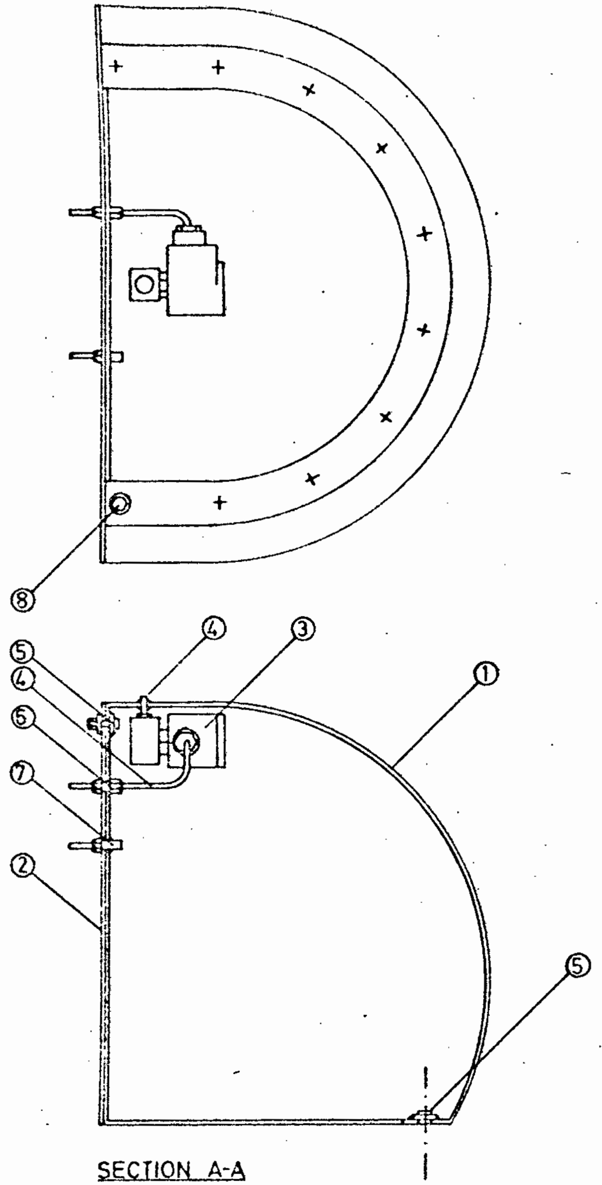
4.1 Forward Section.

The forward section is the only part of the main structure of the vehicle that must be watertight over all its surface, except the base, so that the air trapped within during ascent does not escape. It was decided to cast the section in an approximately hemispherical shape since this contains a large volume of air for its length and is able to withstand a certain amount of internal pressure should the exit holes in the base be temporarily blocked. The hemispherical shape was extended back into a cylinder to give a total enclosed volume of 0, 04 m^3, giving a lift of 400 N to the submersible when filled with air. To facilitate moulding the nose was made in two sections; a curved part and a flat bulkhead, cut as part of a disc, which were subsequently bonded together. Dunlop (20) suggested the use of 3 mm cross laid fibreglass matting for the structure, which is illustrated in Figure 4.1. Blind brass threaded caps were glued with epoxy resin at 50 mm intervals to the inside of the rear flange on the curved section, and the bulkhead was tightened onto these against the flange to ensure a tightly bonded joint. When the resin bond had hardened the caps were used as location points to attach the nose to the stern section.

To ensure that as much air as possible was exhausted from the nose at the upper switching depth the solenoid control valve was positioned as close to the roof as possible. The greatest pressure across the valve seat would then be a head of seawater equal to the over all height of the body, that is a pressure of about 0,04 bar. Details of the valve chosen are given in Table 5.1. Before fitting in position it was totally immersed in an oil bath and pressurised to 15 bar to test for possible leakage of the coil casing in service. The casing proved watertight but had there been a slight leak it would have been filled with heavy oil or wax and retested. These would have kept the electrical contacts free from seawater

PARTS LIST

1 FIBREGLASS NOSE
2 FIBREGLASS BULKHEAD
3 SOLENOID VALVE
4 COPPER TUBING
5 BRASS BLIND CAP FOR WATERTIGHT JOINT
6 TUBING CONNECTOR FOR ELECTRICAL LEADS TO 3
7 TUBING CONNECTOR FOR AIR LINE FROM THE STERN SECTION
8 BRASS BOLT FOR ATTATCHMENT TO BASE



SCALE

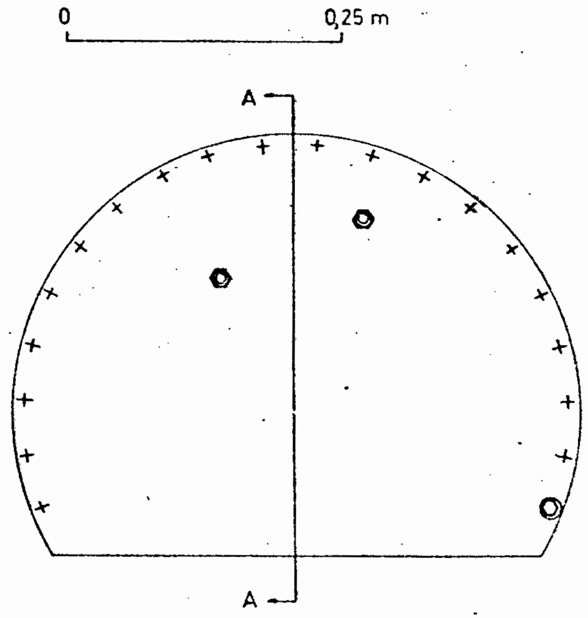


FIG 4.1 NOSE SECTION

but would not have hindered the operation of the valve since there are no exposed moving parts. A short length of copper tubing was bonded into the roof with epoxy resin and the valve outlet clamped to this against an O-ring seal. The valve inlet was left open.

The bulkhead was pierced in two further places, and two brass tubing connectors fixed in place with 'Araldite' epoxy glue to allow contact between the forward and stern compartments. A 6 mm O.D. copper tube was attached between one of these and the electrical imput to the valve. It carried the supply leads and protected them from seawater and the external pressure. The other connector was joined to the other solenoid valve and supplied the compressed air to the forward section.

The flange around the bottom of the nose section was used to locate it to the base. This joint was not watertight; blind caps were glued to the inside of the flange at 90 mm intervals and the base bolted fast against them.

4.2 Stern Section Structure.

It has already been mentioned in Section 2 that the vehicle is built around a vertical steel I-beam, and that the stern section is enclosed by two fibreglass covers shaped to be part of a cylinder. Once the original concept of the design had been formulated the dimensions were calculated from those of the air cylinders most readily available at the time. These had a diameter of 200 mm and an over all length, including pillar valve, of 660 mm. The cross sectional area of the vehicle is the most critical dimension, since this determines the drag forces and the towing load. Two bottles were to be carried, and to allow for fittings and the thickness of the central plate a maximum external width of 500 mm was decided on. This meant that the side covers had to be moulded to fit inside a cylinder of 250 mm radius. The exact dimensions of the other equipment to be mounted within the structure, such as the recorder and the pressure resistant containers for the control mechanism, were not known at that stage, so a length of 910 mm was chosen for the plate. This seemed to give ample room inside for the fittings, and so it proved later. This gave an over all length of 1,30 m to the body, well within the specifications.

In designing the central beam it must be remembered that it is essentially a load bearing tensile member. It transmits the dynamic loading on the vehicle fins to the towing cable, and also the weight of the body when it is lifted onto the towing vessel, when it may be filled with water. Besides this it must be a rigid member, corrosion resistant and as light as possible. The ideal material would be an aluminium alloy, but the design incorporated a certain amount of welding, and aluminium welding is a specialist task. Stainless steel did not justify its cost for a prototype, although for a production model it might be essential, so a welded low carbon steel structure was decided upon which could be galvanised and painted for protection. 100 mm wide steel strips were welded along the top and bottom of the vertical plate to form the I-beam, the corners being stiffened by webs. After the vehicle had been completely fitted out the beam was removed and sections cut out to lighten it.

A steel D-ring was bolted to each end of the beam to form stiffening ribs to locate the side pieces, and flanges were welded to each of these hoops to support in one case the bow section and in the other the stern wall. An exploded view of this structure is given in Figure 4.7. All these steel pieces were cut from 3 mm plate and after fabrication the whole structure was galvanised and painted. Due to the time taken for manufacture outside the Engineering Department the fibreglass pieces and the I-beam were made concurrently. When the pieces arrived they were found to be oversized due both to the difficulties of accurate moulding of the material, and also to the manufacturing methods used. The first parts made were the bulkhead and stern wall. These were slightly too large and were then used to mark out the other parts. As a result the fibreglass pieces fitted together, but the I-beam had to be supplemented by wooden blocks which were screwed to its base to increase the height. The hoop sections were made 'to fit'.

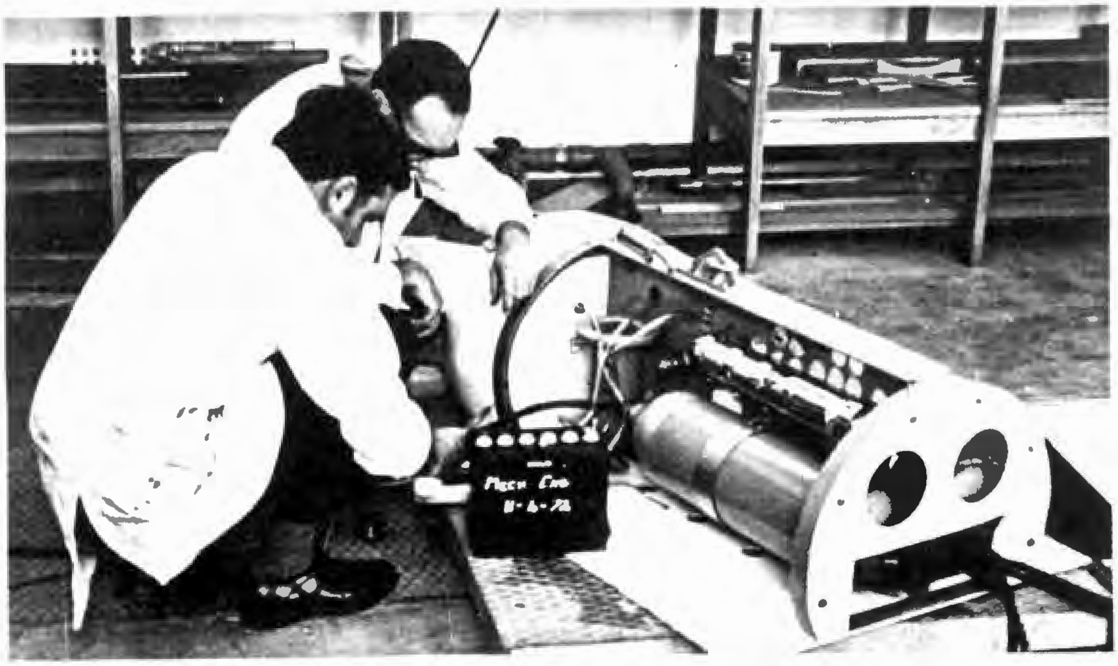


FIG 4.2 GENERAL VIEW

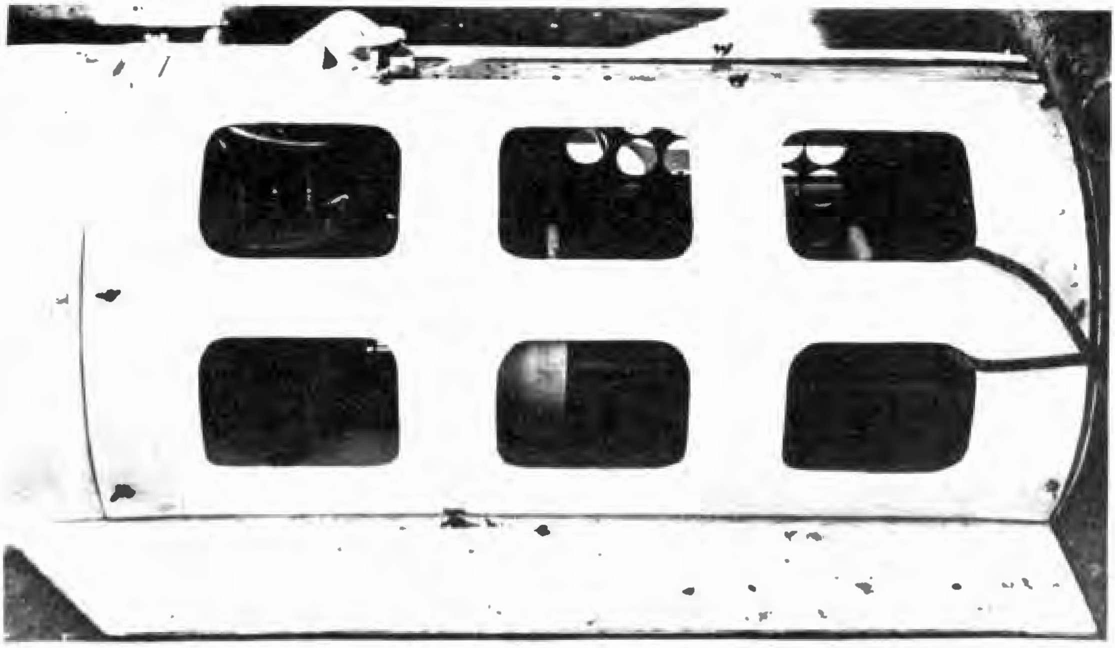


FIG 4.3 SIDE COVER

As mentioned in Sectioned Section 4.2 the front flange supports the bow section and holds it in place by closely spaced brass bolts. The curved side pieces were moulded into a flat at the top to fit onto the top of the I-beam, and each secured by 8 brass wing nuts. Access to the interior of the stern section to renew batteries or recorder reel is thus simple, and access for recharging the bottles unnecessary since there is an external connector, as explained in Section 4.3. These side covers were cut away to facilitate water flow through the stern section and past the recorder (Fig. 4.3). Similarly the stern wall had vents cut in it (Fig. 4.2). This wall is not load bearing and consequently was attached at only a few points around its circumference.

The towing eye was a galvanised, welded low carbon steel structure attached to the upper surface of the beam by 6 stainless steel nuts and bolts. A series of holes were drilled in the beam (Fig. 4.5) so that the eye position could be altered as the trim of the vehicle was changed, either by changing its equipment or internal arrangement. Because it was a welded structure whose safety was crucial for the success of the apparatus the eye was tested with the bolts in shear up to an end loading of 22,25 kN, which exceeds the breaking load of a 6 mm steel cable and is much greater than any anticipated load - see Section 4.5. No signs of failure occurred and the towing eye was judged to be safe for use.

The recorder used was ordered and purchased by the Oceanography Department of the University. It was made by Messrs. Ogawa Suiki Co. Ltd. of Japan and recorded temperature and pressure concurrently on a slowly unwinding sensitive tape with brass pens. This tape was driven by a clockwork motor with a run duration of nine hours. Attachment was made by bolting through an eye at each end onto two threaded stainless steel pillars in the central beam. The recorder must be removed and a pressure cap unscrewed to switch on and off and to remove the tape. It weighed 4,5 kg in seawater and fitted conveniently above the air bottle on the port side of the vehicle (Fig. 4.5).

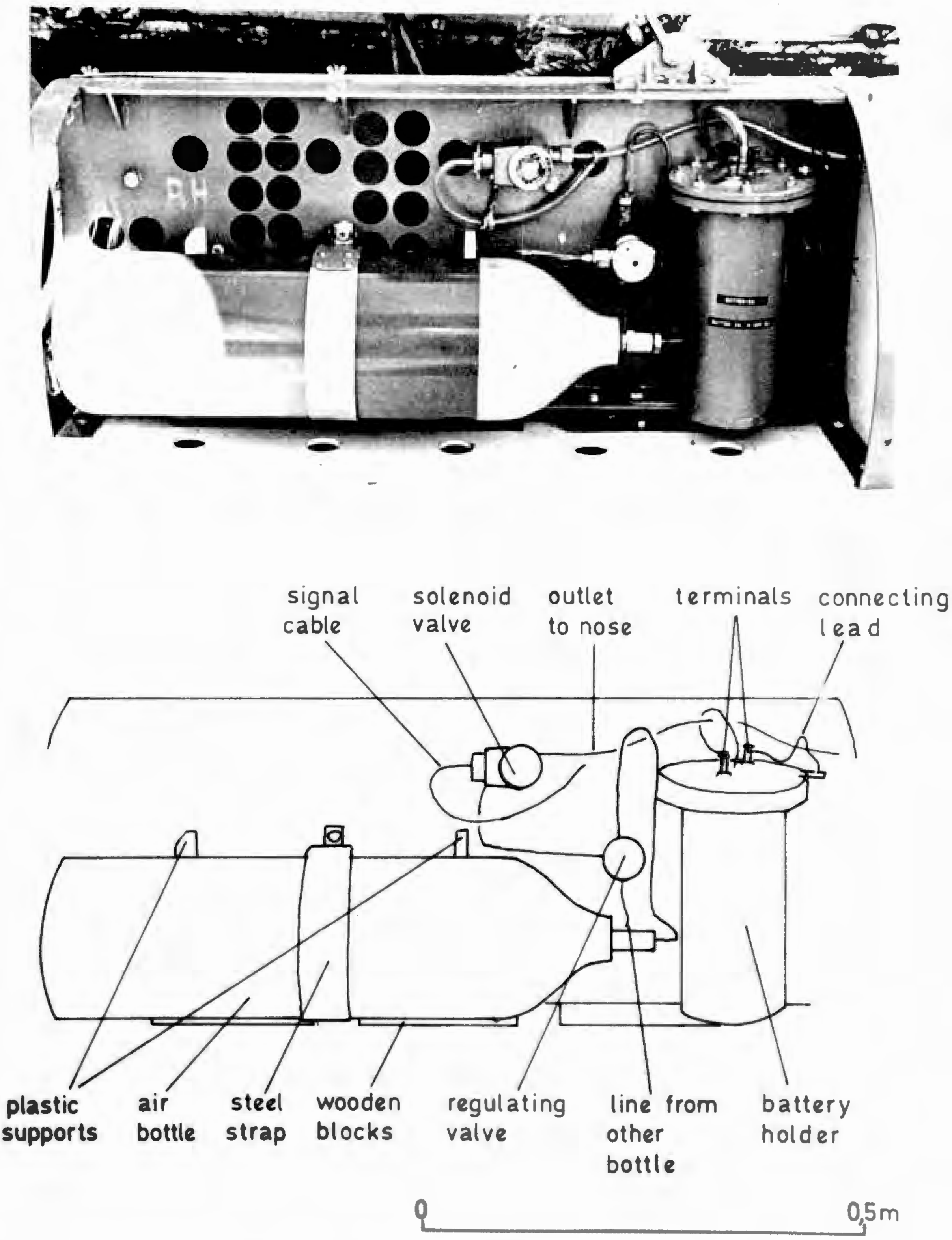


FIG 4.4 STARBOARD SIDE VIEW

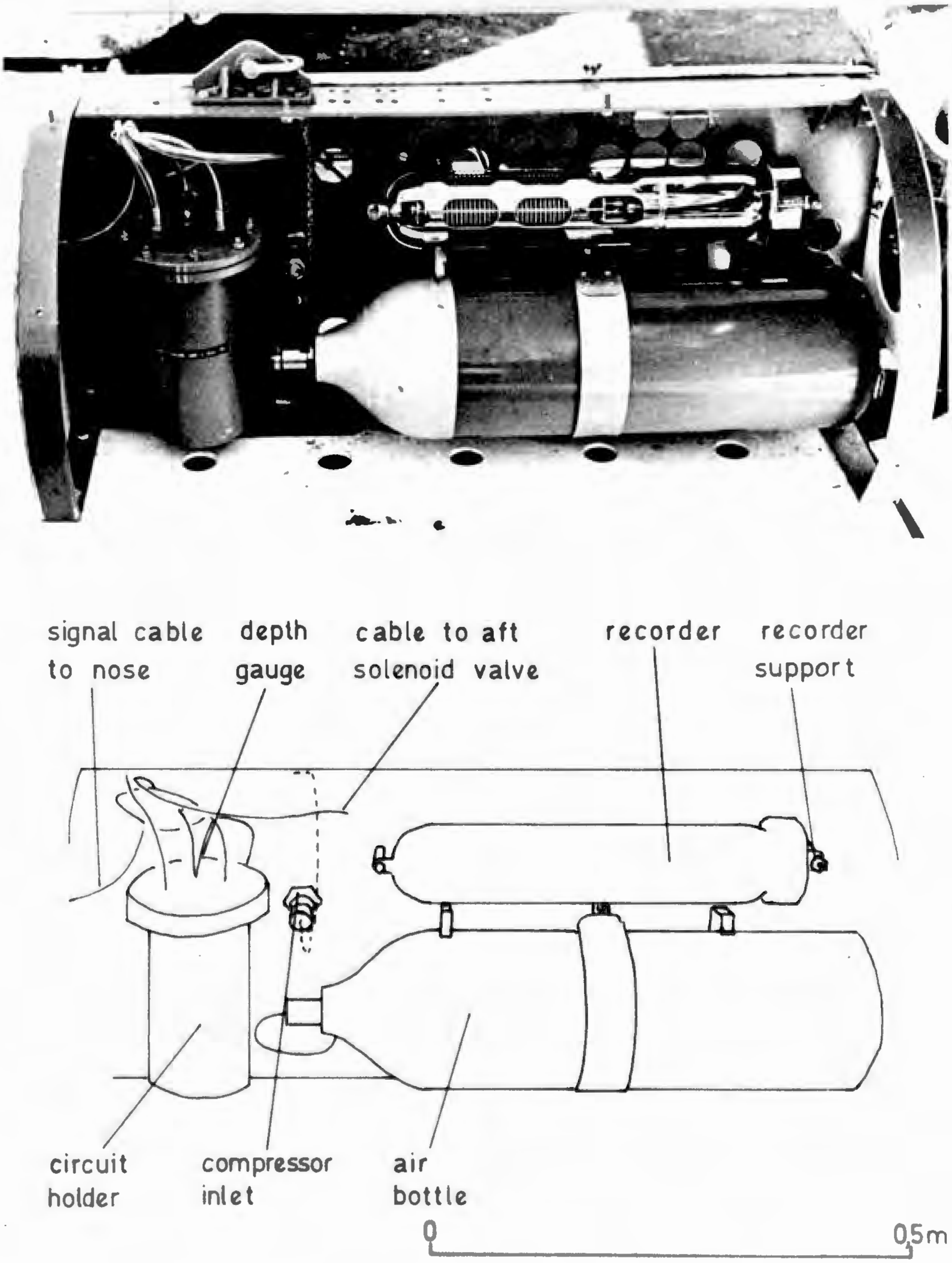
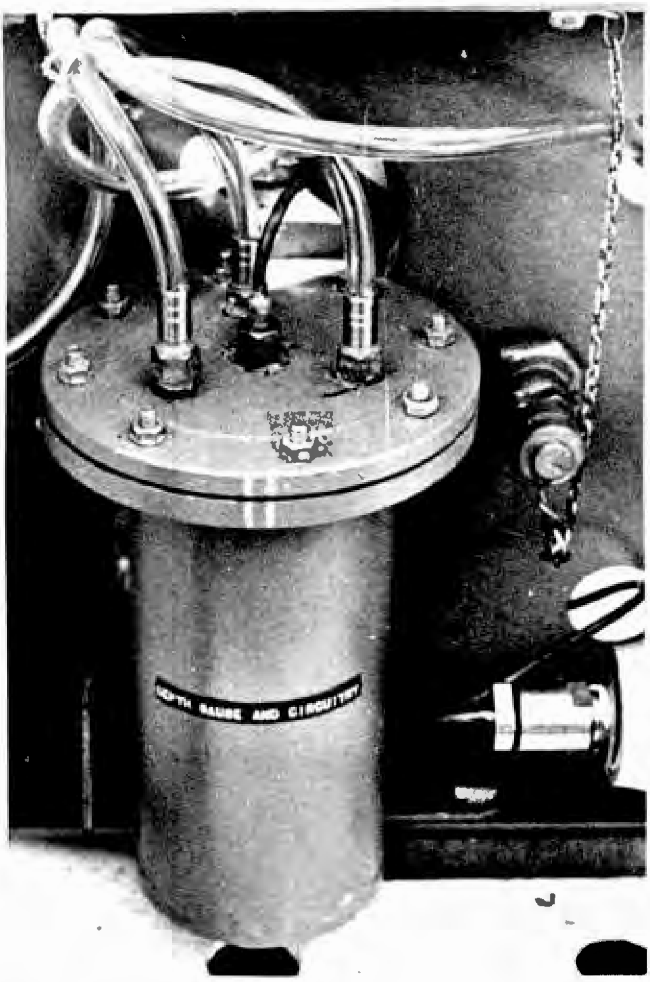


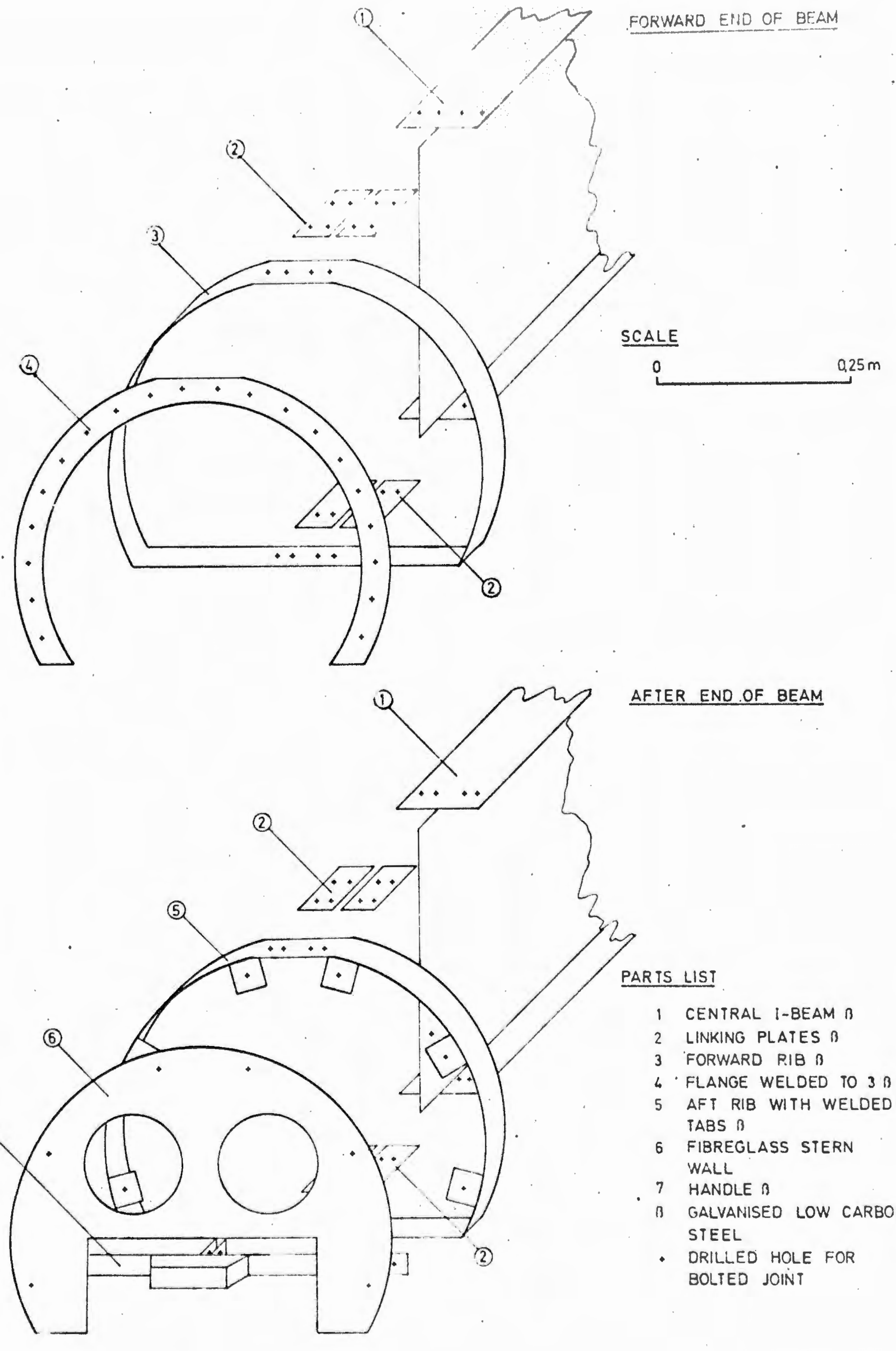
FIG 4.5 PORT SIDE VIEW



Depth gauge and circuit container and compressor inlet

Battery container and regulating valve

FIG 4.6 ELECTRICAL EQUIPMENT CONTAINERS



PARTS LIST

1	CENTRAL I-BEAM β
2	LINKING PLATES β
3	FORWARD RIB β
4	FLANGE WELDED TO 3 β
5	AFT RIB WITH WELDED TABS β
6	FIBREGLASS STERN WALL
7	HANDLE β
β	GALVANISED LOW CARBON STEEL
◆	DRILLED HOLE FOR BOLTED JOINT

FIG 4.7 EXPLODED VIEW OF BEAM CONSTRUCTION

4.3 Air Circuit.

The time taken for a run with the vehicle depends on the battery life, the recorder time and the gas supply. Of these the most easily exhausted is the gas and the greatest consideration when designing the control circuit and towing position was to minimise the volume expended each cycle. Compressed air was eventually used as the lifting medium since it is readily available and easily compressed, whereas a gas such as carbon dioxide manufactured locally in a fire extinguisher needs constant replenishment of materials. Also, equipment to contain air underwater at high pressures is readily available for sub-aqua sports. The valves and cylinders are robust and well tested, besides being light and corrosion resistant. When the vehicle was eventually fitted out the bottles available had a smaller diameter than those for which the original design was calculated, and were longer. Two 10 litre bottles were used connected in parallel to a common manifold. The working pressure inside the bottles was about 185 bar and this was reduced to 7 bar above the ambient pressure by a first stage demand valve. This had an input rating of 200 bar and incorporated a safety valve on the input side. The demand valve is continuously open so the air flow was regulated by a solenoid valve whose maximum pressure across the valve seat was rated at 10 bar. These pieces of equipment were linked by 6 mm copper tubing, and the entire assembly is shown in Figure 4.8. To simplify recharging the cylinders, and hence to increase the turn around time of the vehicle, a one way valve and adaptor for the compressor inlet (Fig. 4.6) were incorporated into the circuit. The cuts in the fibreglass cowling pieces allowed the output from the compressor on the T.B. Davie to be attached without removing the cover.

The valves and piping were securely bolted onto the central beam, and each air cylinder fixed to it by a stainless steel strap lined with neoprene rubber and spaced from the beam by two polythene pieces. Non metallic materials were always used where possible in preference to metals to avoid corroding the

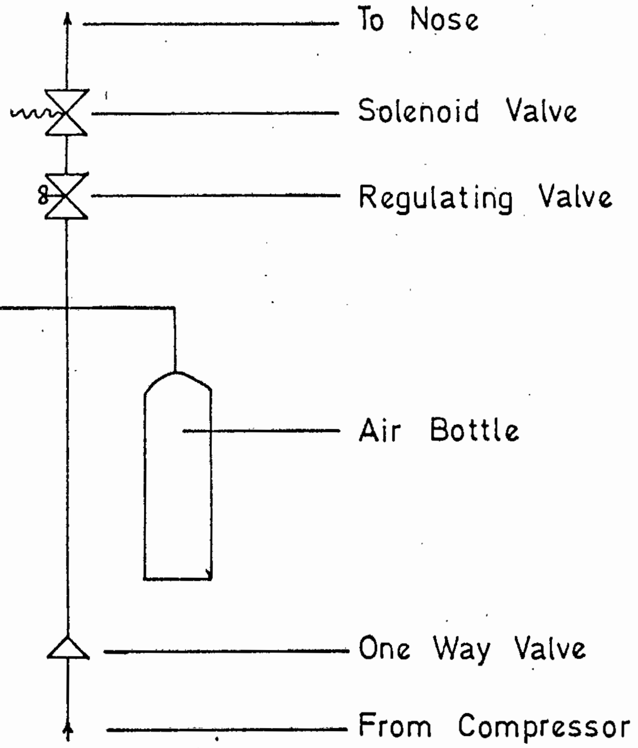


FIG 4.8 SCHEMATIC AIR CIRCUIT

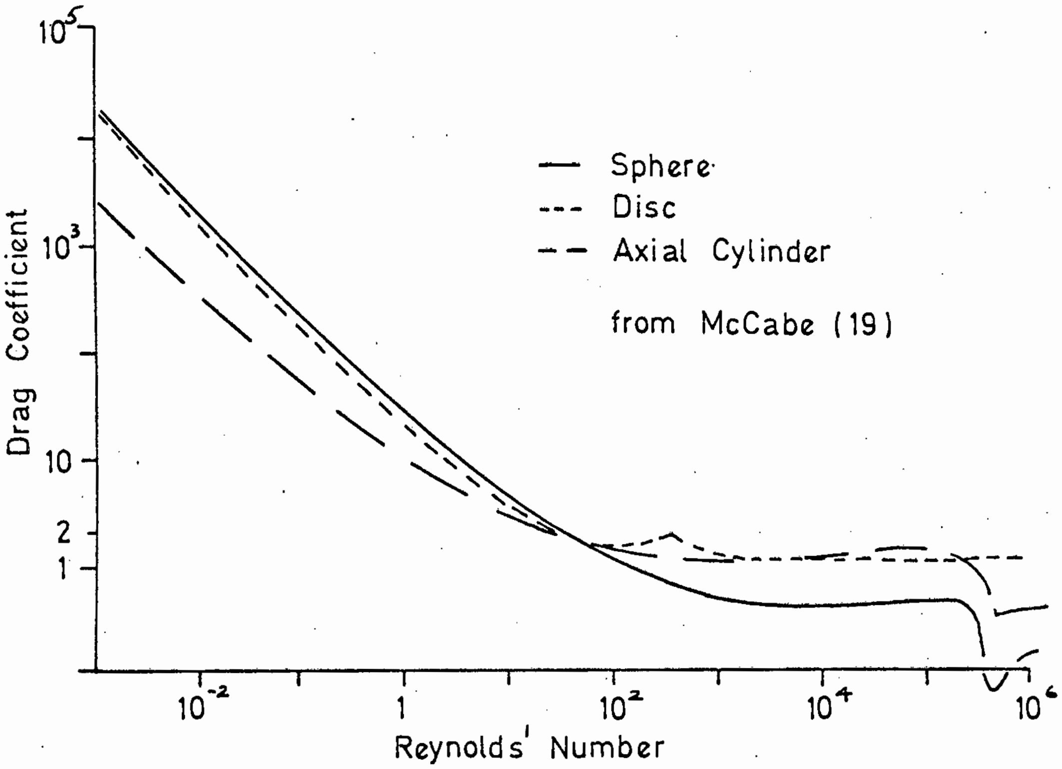
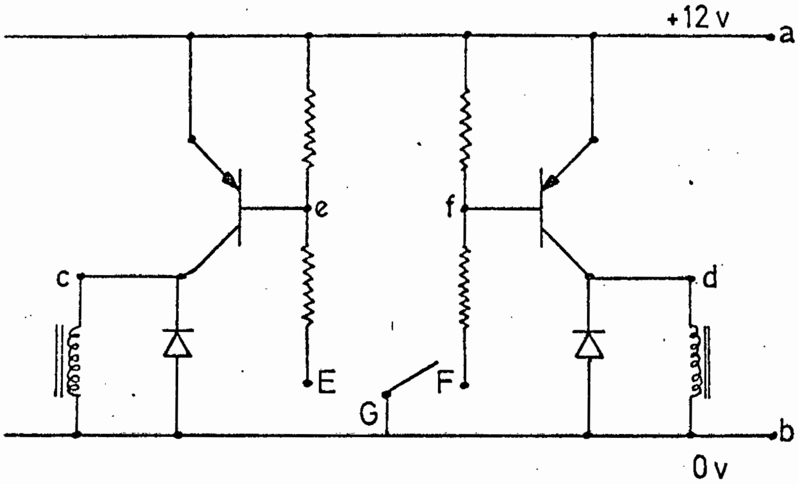


FIG 4.9 TYPICAL DRAG COEFFICIENTS

cylinders. The connectors into the necks of the aluminium alloy bottles were stainless steel protected from direct contact by P.T.F.E. tape. This is standard practice in Scuba diving, where the connectors are chrome plated brass. The bottles were pressure tested seperately at 230 bar and the entire assembly to 230 bar before operation.

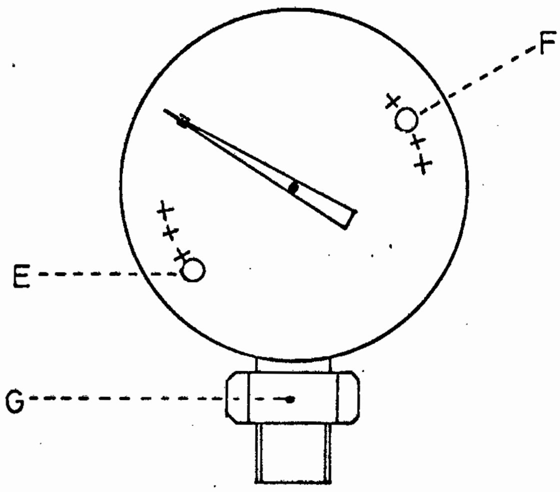
4.4 Control Circuit.

The essence of any control circuitry is that it should be as simple as possible. Since the control valves for the air flow were solenoid operated it was logical to use a transistorised circuit to operate them, rather than use the available compressed air to drive a fluid logic circuit. Since the switching is to occur at two discrete levels the obvious way would be to use a pressure switch with each valve. This would open the valve below a preset depth plus a differential depth that depends on the switch characteristics. Switches are available for use on moving vehicles, but they are expensive and do not have any degree of flexibility in control. So it was decided initially to use a pressure gauge to indicate depth and use the output from an attached potentiometer as the input to a suitable logic circuit. A circuit was constructed but a suitable gauge with an attached potentiometer was priced at R70. This might be appropriate for a working design, but for a prototype seemed an unjustifiable expense when cheaper methods were available. A depth switch was eventually constructed from a Bourdon pressure gauge, one contact of the switch being the moving arm of the gauge, the other a contact fixed at the control depth. When the body sinks the arm moves round to the contact at the higher pressure, completing a circuit to operate one valve. When the body rises again the arm moves off this contact, closing the first valve, until it reaches the higher controlling depth at the lower pressure when the other contact is touched, closing a separate circuit to operate the other valve. The contacts are 9 B.A. brass bolts threaded into the perspex gauge glass. Holes are tapped at intervals around the disc to provide for varying the depths, and the contact bolts and gauge arm are tinned for better contact. Each contact leads to the base of a PNP transistor,



resistors 1 kΩ
transistors TIP 30 A

Circuit



+ tapped 9 B.A. holes
O tinned contacts

gauge exit open to the sea

Depth Gauge

↑towards circuitry

c	b	d	a	E	F

Junction Box Connections

FIG 4.10 SWITCHING CIRCUITRY

and the solenoid valve is the collector load. To avoid surging in the load circuit when the transistor switches off and the valve closes, a diode across the coil acts as a voltage leak (Fig. 4.10).

The controlling equipment was enclosed in two watertight canisters, each mounted on either side of the central beam and at the front of the stern chamber (Figs. 4.4, 4.5). These were made from 90 mm I.D. steel piping with a blank plate welded into one end and a flange to the other. To this was bolted a disc carrying tubing connectors to bring the electrical leads into and out of the containers. One holds the batteries, and the negative output passes through a lead trapped between two terminals on top of the container. Unscrewing one terminal switches off the electrical circuitry without having to unbolt the container. The original batteries were two 6 volt dry batteries, but after the first sea trials (Section 5.2) these were found to have a low capacity and were replaced by rechargable wet cells. The other container holds the control circuit and depth gauge, which is threaded to tubing sealed into the disc. The gauge can then easily be removed by unbolting and removing this lid. Electrical leads were fed through reinforced plastic tubing to protect them from seawater, the ends of the tubing being glued to the tubing connectors with 'Araldite'. Both containers were galvanised, painted and internally pressure tested to 15 bar to check the welding, then bolted securely to the beam.

4.5 Base.

The base was cut from a flat piece of 6 mm cross laid fibreglass. It was shaped to fit the nose and bolted to its bottom flange (Fig. 4.1) and bolted to the stern section along the bases of the central beam and the stiffening hoops. A series of 35 mm diameter holes were cut through the base to facilitate water flow and a larger hole of 100 mm diameter cut under the nose to give access to join up the connectors to the forward solenoid valve. The position of these holes was unimportant since even if they extended to the very front of the nose air would only spill out from them when the vehicle was 'nose up'; once the nose dropped the holes would be covered by water and the nose would refill with air. The forward extent of the holes does, however, determine the maximum attitude the body can adopt in ascent.

Myers (6) suggests a figure for the yield stress of cross laid fibreglass matting of 200 MN/m^2. Assume that in the very worst case the vehicle is dragged perpendicular to the water at 10 knots with a drag coefficient of 2,0, i.e. that of a flat plate across a fluid path (19). Since the base is supported along the centre and for part of its width front and rear then for the purposes of this analysis it falls between being a flat plate totally supported along three edges and a simple cantilever. In the worst case it will be a cantilever, and calculations based on a cantilevered length of 0,375 m gave a safety factor of 2,8 in this case. For the design speed of 7 knots this becomes 5,7. Since the conditions represent an extreme case the factor was considered adequate for safety. If conditions became very bad the vehicle would probably be hauled aboard and stowed before the possibility of breakage occurred. The total force on the towing wire at 10 knots was 8,3 kN, considerably less than the breaking load of the 6 mm coring cable, estimated at 17,8 kN (5).

For a base width of 0,75 m each fin has a width of 0,16 m and the ratio w/r is 0,75 (Appendix C). Figures C.5 and C.7 show that for values of w/r between 1,00 and 0,67 the fins generate almost all of the dynamic lift except when they are greatly reduced in size, and that for most practical purposes the effect of the hull can be ignored. The fins were assumed to be two flat plates whose centre of lift for small angles of incidence was L/4 from the leading edge, where L was the plate length. In our case the leading edge was swept back at 0,79 rad across the width of the fin, so the centre of lift was measured from that edge along the mid line of each fin. Once the position of the towing eye had been determined the fins were shaped so that the calculated centre of lift was either at the towing point or just astern of it. This ensured that the dynamic forces tended to decrease the attitude of the body in the water and so prevented it from turning across the water flow.

Figure 4.9 shows the variation of drag coefficient with Reynolds' Number for typical regular bodies. In the range of Reynolds' Numbers between 10^4 and 10^5 the coefficients remain approximately constant, although there is some variation, as the wind tunnel tests have shown (Appendix C). The values of the coefficients found for the greatest available Reynolds' Number, that of 13 000, can thus be applied to the full scale body with reasonable accuracy.

4.6 Costs.

The choice of materials was determined both by the expediency of building the vehicle quickly and easily and by the considerations of corrosion due to the proximity of materials of different electro-chemical potentials. For this reason the main metal parts were all fabricated from galvanised low carbon steel rather than stainless steel or an aluminium alloy. Once the design had proved successful consideration of cost and durability would be more pertinent, as well as the inevitable modifications to the prototype. An exact analysis of costs was not possible since much of the building was done by technicians at the University who did not keep a detailed record of manufacturing time, and an estimate of the time spent by the author was similarly not attempted. However, a general guide would be as follows;

Manufacturing costs outside the University	R130
Materials and assembly	R 80
Fittings	R165
Technicians' time (estimated)	300 hours

Details of the fittings are given in Table 5.1.

Table 5.1. Specifications of Purchased Fittings.

Air Bottles	10 litre Luxfer aluminium alloy cylinders Working pressure 220 bar Over all length 627 mm Diameter 175 mm
Regulating Valve	1st stage Drager Pokemat demand valve Input pressure 200 bar Output pressure (Adjustable) 7 bar gauge
Forward Solenoid Valve	Lucifer type 121A51 $\frac{1}{4}$ in 2-way valve 12 volt D.C. coil with explosion-proof housing Current rating 0,8 amps Valve seat diameter $\frac{1}{4}$ in Valve seat pressure 0-1,5 bar
After Solenoid Valve	Lucifer type 133A54 $\frac{1}{4}$ in 3-way valve used as 2-way valve 12 volt D.C. coil with explosion-proof housing Current rating 0,8 amps Valve seat diameter in Valve seat pressure 0-10 bar
Depth Gauge	Ferris gauge 0-100 p.s.i.
Batteries	Furukawa Battery Co. Ltd. Model 6N2-2A Yamaha 6 volt batteries Capacity 2 ampere-hours Lead-acid rechargable cells

Once a faulty transistor had been replaced the electrical circuitry was found to work perfectly, except that the controlling batteries soon became flat. This was not unduly disturbing since they were probably old stock when acquired. However, there were no others available and the tests were powered by a car battery placed alongside on the dry floor with leads connected to the vehicle terminals underwater. For the first series of sea trials it was felt that the towing eye should be placed well forward to avoid the possibility of the vehicle being towed backwards, although this was considered a remote possibility because of both its rounded nose and bluff stern, and the wide fins set aft of the towing point. The eye was then moved forward, and 10 kg of lead pieces bolted into the nose to balance the vehicle. The batteries were renewed, the bottles filled to 160 bar and the accessories arranged for the first sea trials.

5.2 Sea Trials.

Due to the crowded timetable of the T.B. Davie the first series of sea trials were arranged at short notice while the ship was stationed in Langebaan lagoon. Because of the haste there was not time to fit more powerful batteries than those dry batteries already tested. Once aboard, the circuit was tested by closing the battery terminals and listening for the opening of the forward valve, and then the vehicle was securely lashed down for the trip to the open sea. This was necessary since the body was to be tested in at least 80 m of water to give an ample safety margin, and this depth was not achieved until about 1 km offshore. Once out of the shelter of the lagoon the surface waves were measured as being from 5 to 10 m high. A warp was tied to the stern handle to facilitate recovery, the recorder started and bolted into place and the electrical circuit made. The towing shackle was then attached through a swivel to the 6 mm coring cable, the boat speed reduced to 4 knots and the body cast over the stern, whence it disappeared in a mass of bubbles. 100 m of cable were run out, the speed increased to 7 knots and after 3 minutes cable was run out to 200 m. About 10 minutes later the vehicle was sighted on the surface some distance dead astern. The ship speed was reduced to 4 knots without any effect so it was hauled aboard.

On testing the circuit the forward valve would not open, and clearly the vehicle had filled with air at the lower depth but had not exhausted it near the surface, and the buoyancy and dynamic lift had kept it afloat. At first it was thought that water had seeped into the leads to the valve, but later tests showed that the batteries were quickly exhausted in seawater, contrary to previous thought, and that this had caused the malfunction. The after deck of the T.B. Davie was being frequently swept by waves and it was too dangerous to effect repairs there, so the vehicle was secured and the ship returned to the lagoon. There the recorded trace was checked and showed that 3 oscillations had taken place between 10 and 60 m with a cycle time of about 2,5 minutes. The thermometer response corresponded to the depth trace, showing that a sufficient flow had passed through the incised louvres and over the sensing elements.

Despite the disappointment of not having completed the proposed series of tests the trial was still partially successful. It showed that the vehicle was robust enough to stand up to very heavy seas and correspondingly difficult launching and landing conditions, and that it was symmetrically profiled and balanced to be towed directly astern. The cycles that were completed showed that with suitable adjustment for response at the lower depth the circuitry worked satisfactorily. That 200 m of cable were needed instead of the 275 m predicted was probably due to the limitations made in the program, for example that the dynamic lift was ignored, and to the very uncertainty of prediction in such heavy seas.

It was not possible to arrange for further sea trials before this thesis was printed, and an account of subsequent testing is included in a supplementary section.

6. Conclusions.

6.1 Assessment of Vehicle Performance.

The trials performed on the vehicle, although these were not fully completed when this thesis was presented, were sufficient to show that it would fulfill its function of oscillating between set depths whilst being towed behind a research ship. Even in rough weather it was able to be launched and recovered safely and access to the recorder and refilling points was easy. The basic simplicity of the design means that any repairs can be performed aboard ship.

The analysis proved extremely valuable as a guide to the length of towing cable needed and the period of the cycle. It was never intended as a precise analysis due to the uncertainty of the magnitude of the dynamic forces acting on the body, but is valid as an indication of what to expect.

6.2 Suggestions for Future Development.

The simplicity and the relatively low cost of operation and manufacture argue that the basic design of the vehicle is satisfactory. There are though clearly some areas where improvement is necessary.

The first of these is in the dimensions. As already mentioned in Section 4 the cross sectional area of the vehicle was greater than anticipated, and also greater than necessary. This could be reduced so that the covers fit more compactly around the equipment inside, and the drag and towing load be correspondingly reduced. Similarly the length could be shortened with an advantageous rearrangement of the weight distribution. It would not be possible to put equipment into the nose since access there is difficult, but smaller pressure vessels for the electrical equipment would result in the air bottles being moved forward and the length reduced.

The choice of materials has already been discussed in Section 4. Fibreglass proved satisfactory for the vehicle shell, but all the metal parts, including the air lines, should be made from stainless steel. It might prove easier to replace the I-beam by a vertical metal plate with lugs bolted to it for fastening the side covers, base and end hoops, and drill a hole through the plate for the towing point instead of having a separate external fixture.

The most sensitive part of the design is the control circuitry. The present equipment is a simple way of regulation that could be improved by a more sophisticated logic network which would have four levels of adjustment instead of the two at present. Each valve could then switch on and off at separately determined levels instead of at the same level. This would conserve air and increase the run time. However, the circuitry is complex, with consequent increased chance of failure, and needs a depth gauge with potentiometer attached to drive it.

Appendix A. General Design for Hydrodynamically Operated Body.

The design utilised a pair of independently controlled fins mounted horizontally towards the rear of the body, (Fig. A.1). These would be rotated through small angles of incidence by double acting cylinders driven by compressed air through solenoid actuated spool valves. Air was chosen in preference to electric motors due to the difficulty in obtaining batteries with sufficient capacity which would fit into the limited space available and be safe enough to operate upside down if necessary. Even the use of a small accumulator and a propellor driven charger was considered and discarded for this reason. The fins would be moved up or down at predetermined depths, the continuous signal from a potentiometer attached to a pressure gauge being converted in a Schmidt trigger to a square wave which would actuate the spool valves (Fig. A.2). The sensors for the stabilising circuit would be a pair of mercury filled U-tubes, to detect angle of roll, and a gyroscope, to detect rate of roll. Outputs from these would modify the signal to the fins as necessary. This equipment must be protected from the sea and enclosed in a suitably streamlined pressure vessel. To allow ready access would mean careful machining of seals and design of the moving parts. The body design proved to be heavy and costly with limited space available for instruments and was therefore discarded.

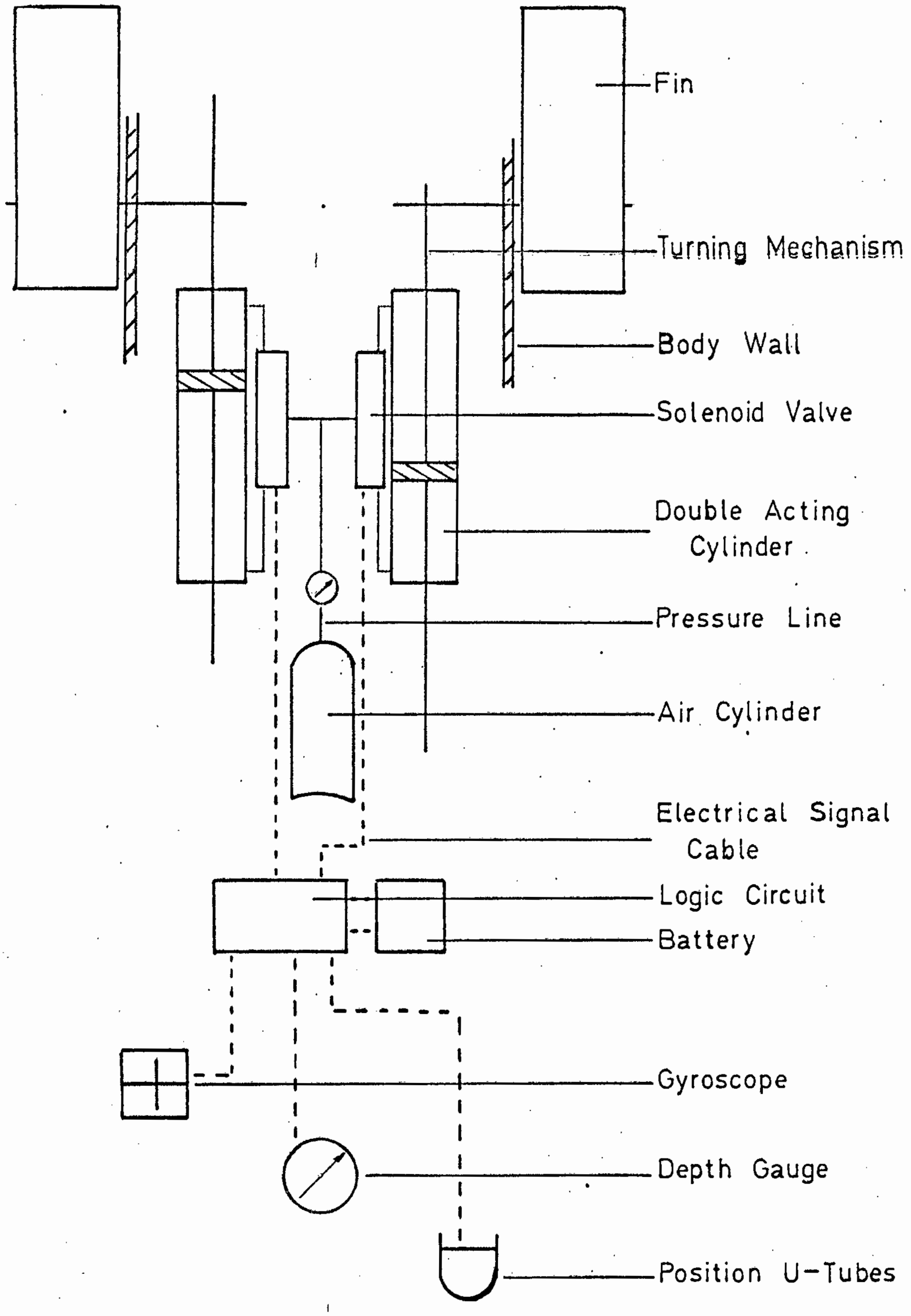


FIG A.1 SCHEMATIC ARRANGEMENT – HYDROPLANE OPERATED BODY

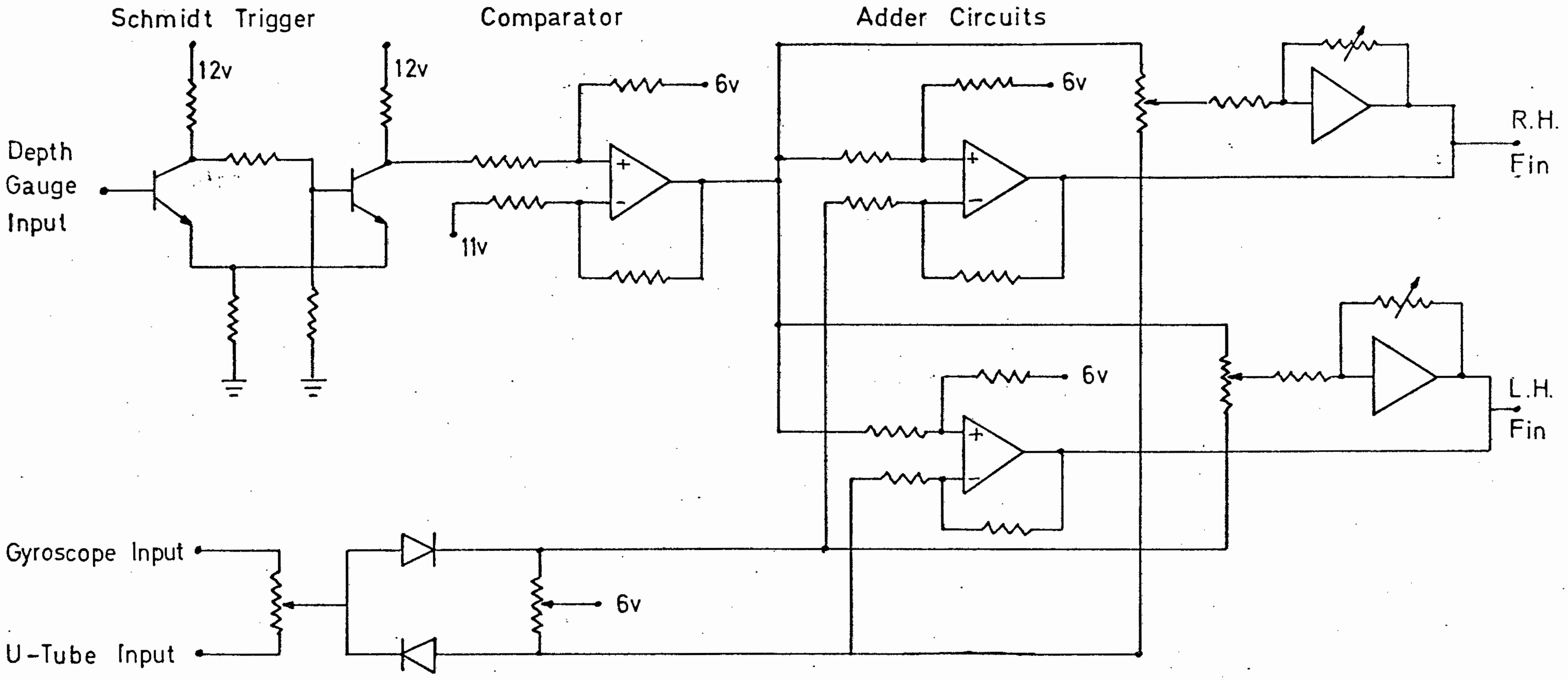
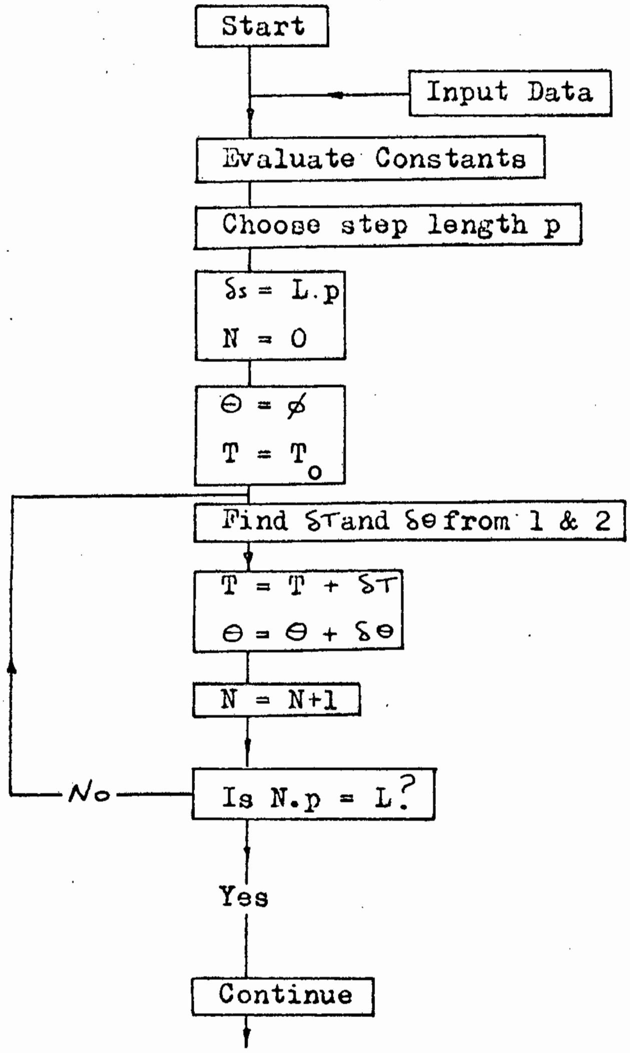


FIG A.2 CONTROL CIRCUIT — HYDROPLANE OPERATED BODY

Appendix B. Computer Solutions.

Steady State Analysis.

To solve equations 1 and 2 the following diagram is evoked.



This gives coincident values of s and θ for L/p discrete points along the cable. The second part of the program takes the towed body as the origin and uses Simpson's Rule to evaluate the integrals

$$y_s = \int_0^s \sin\Theta . \delta s \qquad x_s = \int_0^s \cos\Theta . \delta s$$

For this, of course, there must be an even number of points. Once the vertical and horizontal distances of the ship from the towed body are known the cable co-ordinates are re-evaluated with respect to the ship and printed. Any numerical analysis is subject to error both by virtue of the iteration process and the limitations of the computer. The only way to check accuracy is to vary the step length p for the same set of data and compare results. The program was run using values of p of 0,05 0,02 0,01 and 0,005. The last two series of results were equal to 3 decimal places, so p = 0,01 was used as being the most exact and economical. A typical print out is included with the program.

STEADY STATE ANALYSIS — PROGRAM

```
      DIMENSION ANGLE(101),ROPE(101),ANG(51),DIS(51),ANCOS(101),
     *ANSIN(101),X(51),Y(51)
  100 FORMAT(T5,'TOWING CABLE PROFILE ACCORDING TO DUNKLEY S ANALYSIS')
  101 FORMAT(//T5,'CURVE COORDINATES'/T5,'DISTANCE ASTERN',T35,'DEPTH',
     *T55,'DISTANCE ALONG ROPE'/
     *(T15,F8.4,T35,F8.4,T55,F8.4))
  104 FORMAT(T5,'PROBE DEPTH/CABLE LENGTH=',F6.4/
     * T5,'DISTANCE ASTERN/CABLE LENGTH=',F6.4)
  102 FORMAT(7F10.6)
  103 FORMAT(T5,'TOWING LOAD/(PROBE MASS*G)=',F10.4)
  105 FORMAT(/T5,'NON DIMENSIONAL GROUPS-INPUT'/
     * T12,'(CABLE LENGTH=L)'/
     *T5,'P1',T25,'CABLE MASS PER UNIT LENGTH*L/PROBE MASS=',F10.6/
     *T5,'P2',T25,'PROBE DRAG COEFFICIENT=',F10.6/
     *T5,'P3',T25,'PROBE AREA/L**2=',E10.6/
     * T5,'P4',T25,'CABLE DRAG COEFFICIENT=',F10.6/
     * T5,'P5',T25,'CABLE DIA./L=',E10.6/
     *T5,'P6',T25,'SHIP SPEED/SQRT(L*GRAV. CONST.)=',F10.6/
     *T5,'P7',T25,'WATER DENSITY*(L**3)/PROBE MASS=',E10.6)
      READ(8,102) P1,P2,P3,P4,P5,P6,P7
      WRITE(5,100)
C FORMULATE DISTANCE, ANGLE
      PP1=P1
      PP7=P7
      DO 1000 NNN=1,10,1
      P1=PP1*10./NNN
      P7=PP7*10./NNN
      WRITE(5,105) P1,P2,P3,P4,P5,P6,P7
      CON=0.5*P7*P6*P6
      D=CON*P4*P5
      DRAG=CON*P2*P3
      T0=1.+DRAG*DRAG
      T0=SQRT(T0)
      PHI=1./DRAG
      PHI=ATAN(PHI)
      P=0.01
      T=T0
      THETA=PHI
      ANGLE(1)=PHI
      ROPE(1)=0.000
      DO 1 N=1,100
      DIST=P*N
      ST=SIN(THETA)
      CT=COS(THETA)
      DT=P1*ST*P
      DTH=(P1*CT-D*ST*ST)*P/T
      THETA=THETA+DTH
      T=T+DT
      NN=N+1
      ANGLE(NN)=THETA
    1 ROPE(NN)=DIST
      WRITE(5,103) T
C FORMULATE SINE, COSINE
      DO 17 J=1,101
      H=ANGLE(J)
      ANCOS(J)=COS(H)
   17 ANSIN(J)=SIN(H)
C INTEGRATE BY SIMPSON'S RULE
      H=P/3.
      X(1)=0.000
      Y(1)=0.000
```

```
      DO 18 J=3,101,2
      KJ=J-1
      KK=J-2
      KL=(J+1)/2
      KM=(J-1)/2
      Y(KL)=Y(KM)+(ANCOS(J)+4.*ANCOS(KJ)+ANCOS(KK))*H
   18 X(KL)=X(KM)+(ANSIN(J)+4.*ANSIN(KJ)+ANSIN(KK))*H
      DO 19 J=1,51
      ANG(J)=Y(J)
   19 DIS(J)=X(J)
      DEPTH=ANG(51)
      ASTERN=DIS(51)
      DO 20 J=1,51
      I=52-J
      X(J)=ASTERN-DIS(I)
   20 Y(J)=DEPTH-ANG(I)
      DO 21 J=1,101,2
      I=(J+1)/2
      ROPE(I)=ROPE(J)
   21 ANG(I)=ANGLE(J)
      DO 22 J=1,51
      I=52-J
   22 ANGLE(I)=ANG(J)
C TELL IT LIKE IT IS
      WRITE(5,101) ASTERN,DEPTH
      WRITE(5,104)(Y(I),X(I),ROPE(I),I=1,51)
 1000 CONTINUE
      CALL EXIT
      END
```

—TYPICAL RESULTS

```
TOWING CABLE PROFILE ACCORDING TO DUNKLEY S ANALYSIS

NON DIMENSIONAL GROUPS-INPUT
      (CABLE LENGTH=L)
P1                  CABLE MASS PER UNIT LENGTH*L/PROBE MASS= 38.410000
P2                  PROBE DRAG COEFFICIENT=    .850000
P3                  PROBE AREA/L**2=.530000-05
P4                  CABLE DRAG COEFFICIENT=  1.200000
P5                  CABLE DIA./L=.724000-04
P6                  SHIP SPEED/SQRT(L*GRAV. CONST.)=    .087100
P7                  WATER DENSITY*(L**3)/PROBE MASS=.122000+10
TOWING LOAD/(PROBE MASS*G)=   31.3671
PROBE DEPTH/CABLE LENGTH= .2745
DISTANCE ASTERN/CABLE LENGTH= .9600


CURVE COORDINATES
DISTANCE ASTERN                DEPTH          DISTANCE ALONG ROPE
           .0000               .0000          .0000
           .0191               .0060          .0200
           .0381               .0121          .0400
           .0572               .0181          .0600
           .0763               .0241          .0800
           .0953               .0302          .1000
           .1144               .0362          .1200
```

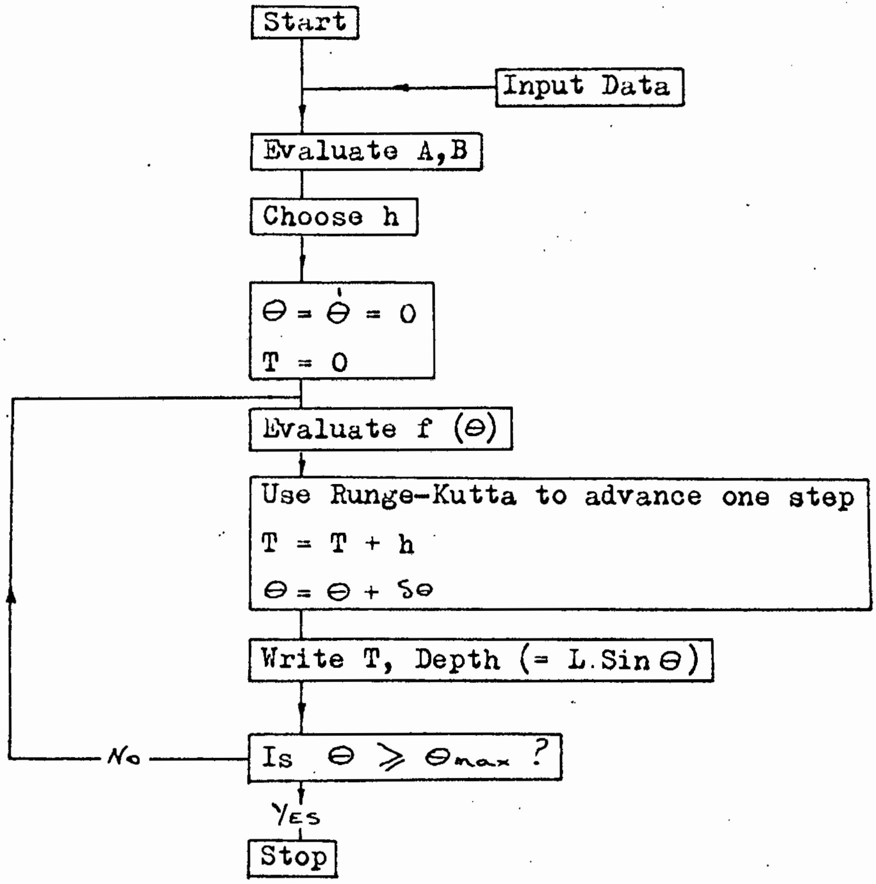
.....etc

Dynamic Analysis.

The equation to be solved is of the type $\ddot{\theta} = A\dot{\theta}^2 + B.f(\theta)$
This can be rewritten as two first order simultaneous differential equations

$$\dot{\theta} = x$$
$$\dot{x} = Ax^2 + B.\ f(\theta)$$

The program was written to utilise the Runge-Kutta method to advance the solution of these two equations by steps of $\delta t = h$ starting from the initial conditions when the body is on the surface of $\theta = \dot{\theta} = 0$



Runge-Kutta was used in preference to methods such as Adams-Moulton or Hanning predictor-corrector methods, because it gave accurate results at uniform time intervals. The other methods gave results at non-uniform time intervals due to the automatic changes in time interval associated with them.

If this automatic procedure was suppressed the accuracy or the computer time suffered because of the small values of step length required. For all the cases evaluated there was no difference to the fourth decimal place in the results between using double or single precision, so that the latter was adopted for economy. As in the previous case different values of h were tried to obtain the optimum solution. A program and sample solution are included.

DYNAMIC ANALYSIS — PROGRAM

```
      DIMENSION Y(2),YY(2),E(2),G(2)
100   FORMAT(T5,F10.4,T30,F10.4)
101   FORMAT(7F10.6)
102   FORMAT(1H1,T5,'NON DIMENSIONAL GROUPS - INPUTS'/
     1 T12,'(CABLE LENGTH=L)'/
     2T5,'P1',T25,'CABLE MASS PER UNIT LENGTH*L/PROBE MASS=',F10.6/
     3T5,'P2',T25,'PROBE DRAG COEFFICIENT=',F10.6/
     4T5,'P3',T25,'PROBE AREA/L**2 =',E10.6/
     5 T5,'P4',T25,'CABLE DRAG COEFFICIENT=',F10.6/
     6 T5,'P5',T25,'CABLE DIA./L=',E10.6/
     7T5,'P6',T25,'SHIP SPEED/SQRT(L*GRAV. CONST.)=',F10.6/
     8T5,'P7',T25,'WATER DENSITY*(L**3)/PROBE MASS=',E10.6/
     91H1,T5,'TIME',T30,'DEPTH')
      READ(8,101) P1,P2,P3,P4,P5,P6,P7
      PP1=P1
      PP7=P7
      DO 11 NN=12,20,2
      P1=PP1*10./NN
      P7=PP7*10./NN
      WRITE(5,102) P1,P2,P3,P4,P5,P6,P7
      A=1.+P1/3.
      CON=0.5*P6*P6*P7
      R=(P2*P3+P4*P5/4.)*P7/2.
      R=-R/A
      Y(1)=0.251
      Y(2)=0.000
      H=0.02
      X=0.000000
      DO 10 N=1,1000
      DO 1 J=1,4
      C=Y(1)
      AC=COS(C)
      AS=SIN(C)
      Z=(1.+0.5*P1)*AC
      Z=Z-(P2*P3+0.5*P5*P4*AS)*CON*AS
      Z=Z/A
      YY(1)=Y(2)*H
      YY(2)=(Y(2)*Y(2)*R+Z)*H
      DO 2 JJ=1,2
      GOTO(3,4,5,6),J
    3 E(JJ)=Y(JJ)
      G(JJ)=Y(JJ)-YY(JJ)/6.
      GOTO 4
    6 E(JJ)=G(JJ)-YY(JJ)/3.
      GOTO 4
    5 E(JJ)=E(JJ)+YY(JJ)/2.
    4 Y(JJ)=E(JJ)+YY(JJ)/2.
      G(JJ)=G(JJ)+YY(JJ)/3.
    2 CONTINUE
      GO TO(7,1,7,1),J
    7 X=X+0.5*H
    1 CONTINUE
      C=Y(1)
      DEPTH=SIN(C)
      WRITE(5,100) X,DEPTH
      IF (DEPTH.GE.0.251) GOTO 11
   10 CONTINUE
   11 CONTINUE
      CALL EXIT
      END
```

— TYPICAL RESULTS

```
NON DIMENSIONAL GROUPS - INPUTS
      (CABLE LENGTH=L)
P1              CABLE MASS PER UNIT LENGTH*L/PROBE MASS=  2.53000
P2              PROBE DRAG COEFFICIENT=    .85000
P3              PROBE AREA/L**2 =.48700D-05
P4              CABLE DRAG COEFFICIENT=  1.20000
P5              CABLE DIA./L=.62300D-04
P6              SHIP SPEED/SQRT(L*GRAV. CONST.)=   .08490
P7              WATER DENSITY*(L**3)/PROBE MASS=.36250D+02



    TIME                  DEPTH
        .0200                .2485
        .0400                .2488
        .0600                .2492
        .0800                .2497
        .1000                .2503
        .1200                .2509
        .1400                .2515
```

Appendix C. Wind Tunnel Tests.

Once the general shape of the body had been decided it was important to evaluate its lift and drag characteristics for different detailed profiles. These would be used in the analytical work and in determining the final design.

The most suitable apparatus for simulating flow conditions was the low speed wind tunnel in the fluid mechanics laboratory. This incorporated a lift and drag balance which the other available testing facilities, i.e. the flume tank and the towing tank in the Civil Engineering Department, did not have. The tunnel had a 240 mm diameter working section and an inlet length of 500 mm, of which 130 mm were flow straighteners. The maximum indicated air velocity was 22 m/s, which gave a Reynolds' Number based on diameter of $1,3 \times 10^4$ using a 3/20 scale model, whereas at the design speed in water the Reynolds' Number of the full scale body would be $1,5 \times 10^5$. There was little that could be done about this discrepancy except to examine the characteristics over a wide range of Reynolds' Numbers and extrapolate, as discussed in Section 4.5.

The first series of tests investigated the effect of changing the dihedral angle of the fins (Fig. C.1). The dihedral angle is important from considerations of lift and stability, negative dihedral causing instability in side slip. The model was equipped along its length with wings of width equal to its radius, the leading edges angled back at 0,79 rad. The dihedral angle was varied between ± 0,52 rad, and angles of incidence (Fig. C.2) between ± 0,26 rad. Results were obtained for Reynolds' Numbers of 5 000, 9 000, and 12 000, and predictably increasing the Reynolds' Number increased the lift/drag ratio for the body in every case due to the increased energy of the boundary layer resulting in later separation (Fig. C.3). The lift/drag ratios for the various wing angles are shown in Figure C.4 for particular angles of incidence. In all cases the highest ratio occurred at a dihedral angle between 0 and 0,17 rad. Fixing the fins at 0 rad meant that the base of the body could be moulded simply as a flat sheet, facilitating the assembly and access for the prototype model.

Having decided on the fin angle the only variable left was their area. Figures C.5 to C.8 summarise the results of the second series of tests, made on models with the fins set at 0 rad dihedral angle but with length and width varying as indicated in Fig. C.2. All the tests were performed at a Reynolds' Number of 13 000; lower air speeds gave lower values of lift and drag forces, increasing the error in the results. Four graphs are included for completeness. Figure C.6 shows the variation in lift/drag ratio for descent, and since the action of increasing drag is to raise the vehicle and that of increasing lift is to lower it the ratio must be large for rapid descent. Figure C.7 estimates the dynamic lift forces on the ascending body due to its fins and hull shape. As demonstrated in Section 3.4 dynamic lift is essential for a vehicle to approach the surface. If the magnitude of the force is too large the cycle time will be short, and if too small the cycle will never be completed as the body will stabilise during the rising part of its path at a depth below its upper switching depth. It is assumed that the magnitude of the forces is unaffected by the position of the fins along the body and depends only on their effective size. Since the forces on a finless vehicle are small when compared to those due to the fins this assumption is valid. The graphs show that in descent the body is better profiled hydrodynamically due to the cylindrical part interrupting and directing the flow before it reaches the fins. On ascent the body appears very like a flat plate to oncoming fluid and its lift/drag characteristics suffer accordingly.

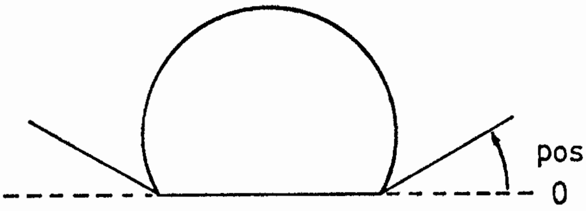
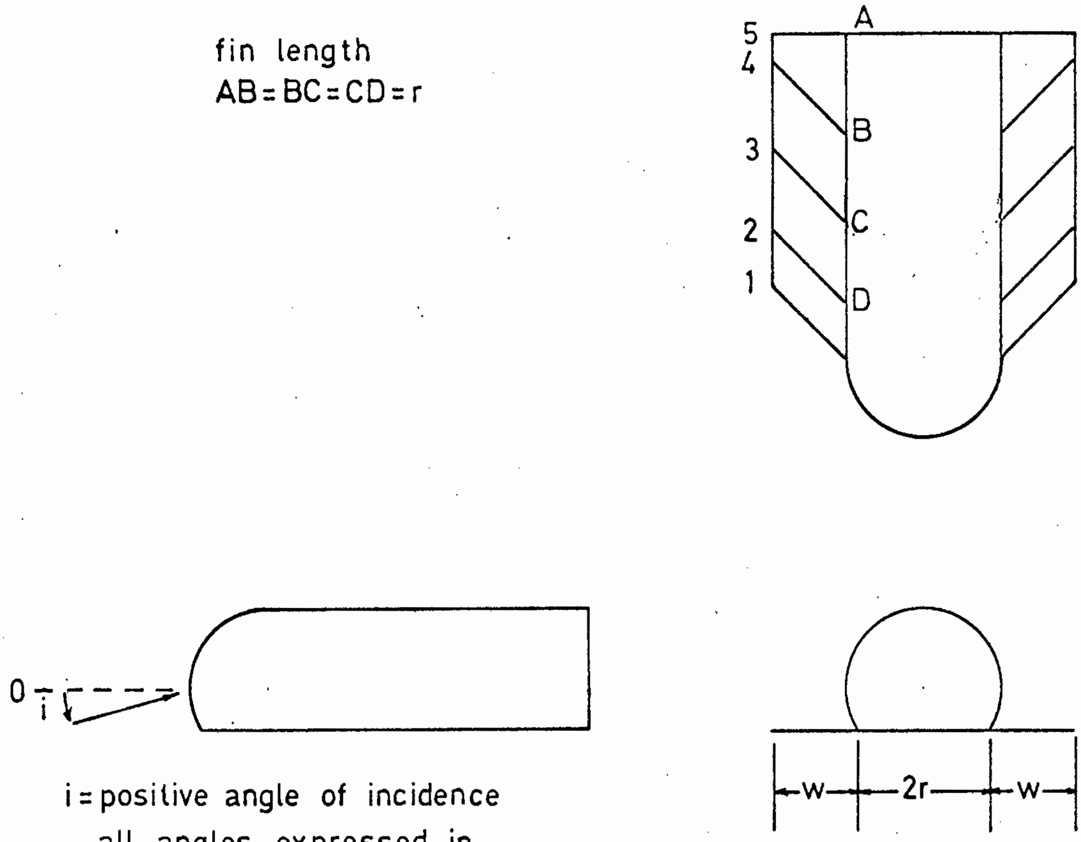


FIG C.1 DIHEDRAL ANGLE



fin length
AB = BC = CD = r

i = positive angle of incidence
all angles expressed in
radians

w = fin width

FIG C.2 WIND TUNNEL TEST DEFINITIONS

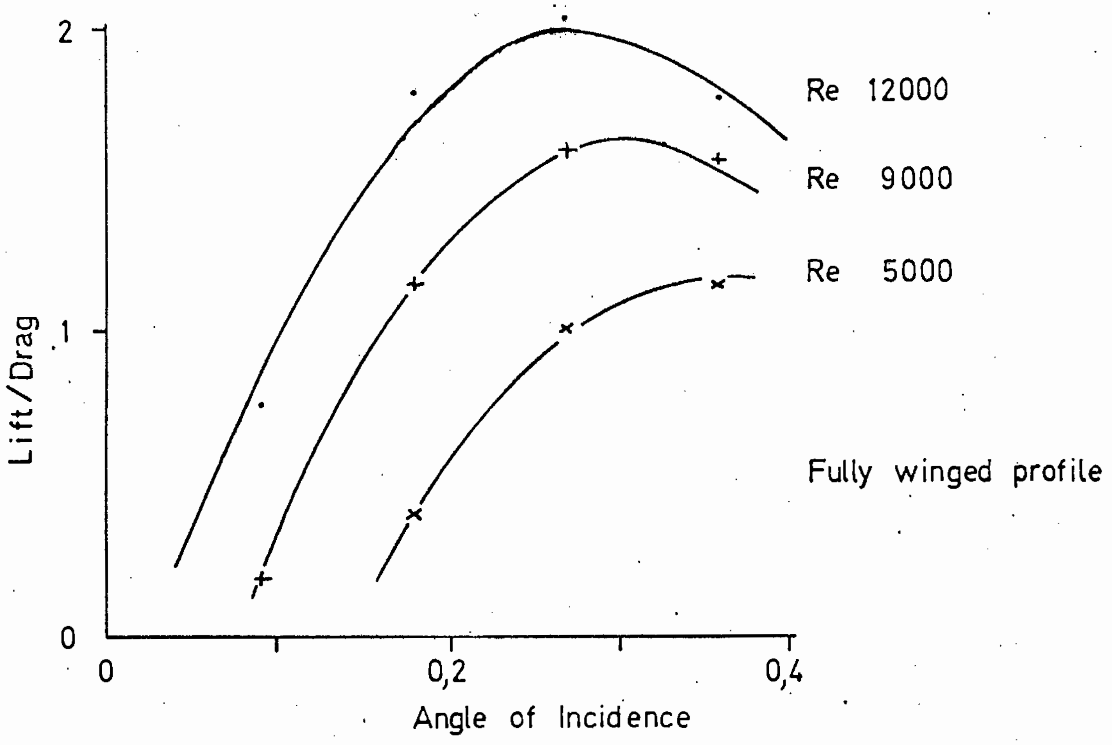


FIG C.3 CHANGE IN LIFT/DRAG WITH REYNOLDS' NUMBER

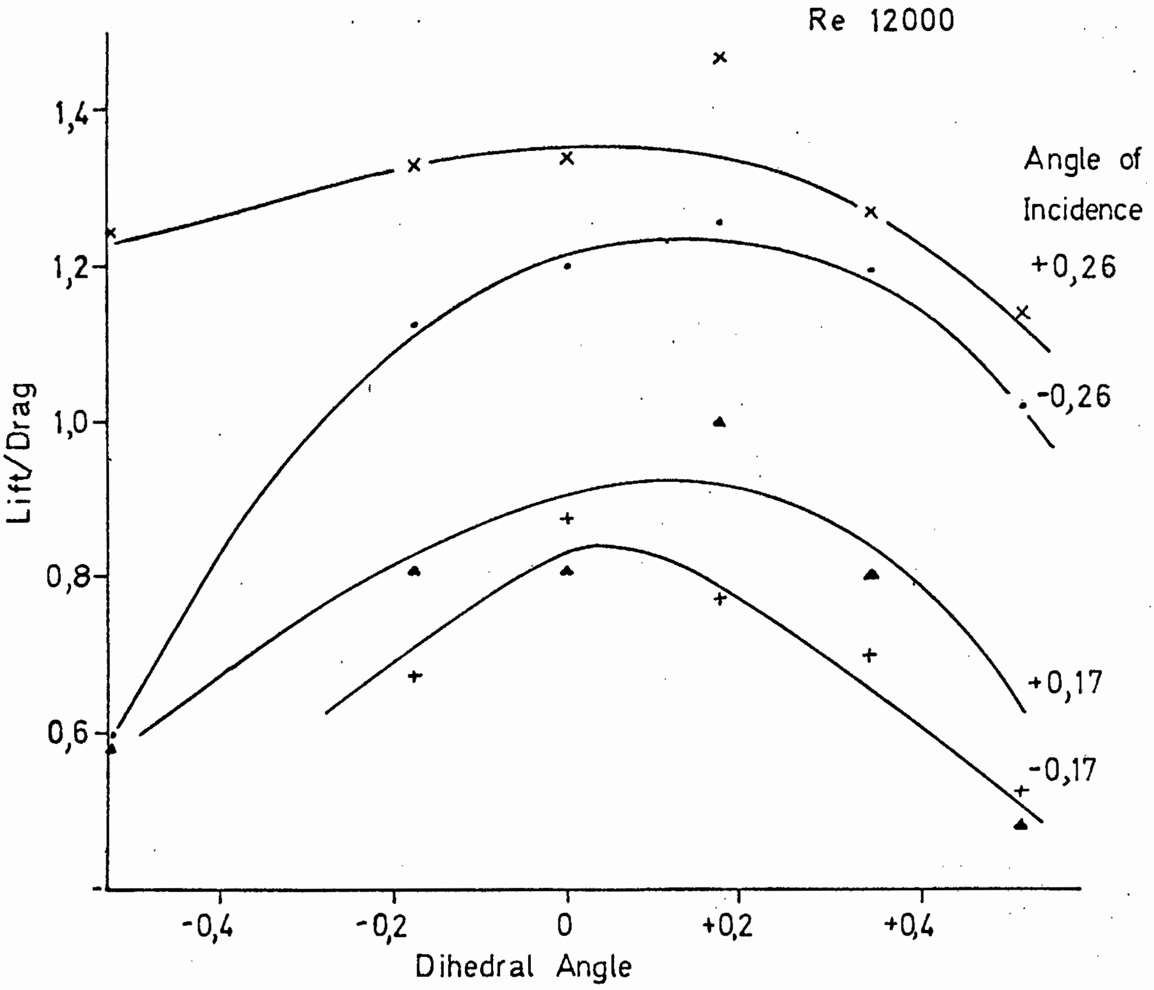


FIG C.4 CHANGE IN LIFT/DRAG WITH DIHEDRAL ANGLE

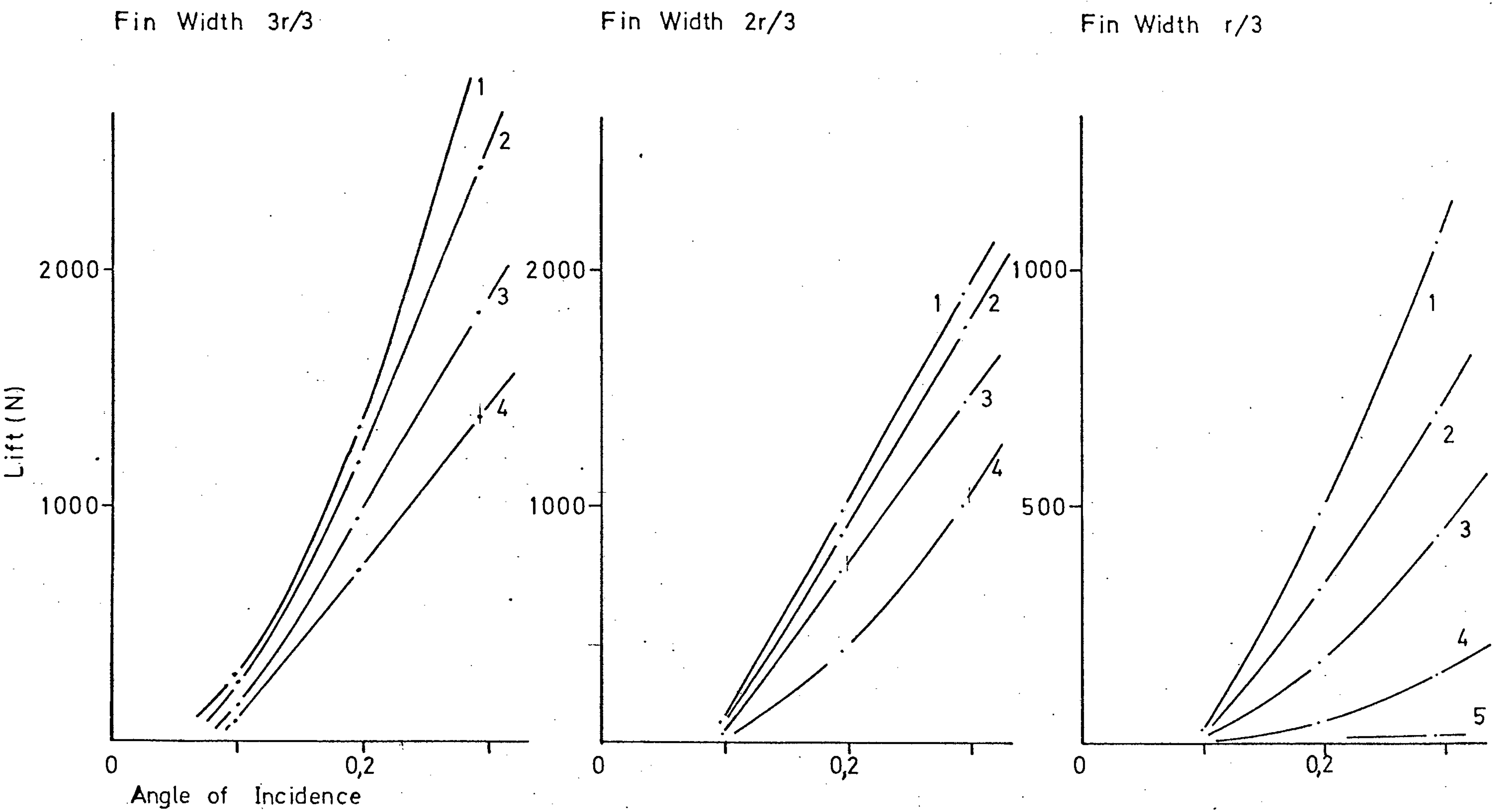


FIG C.5 LIFT FORCES FOR DESCENT

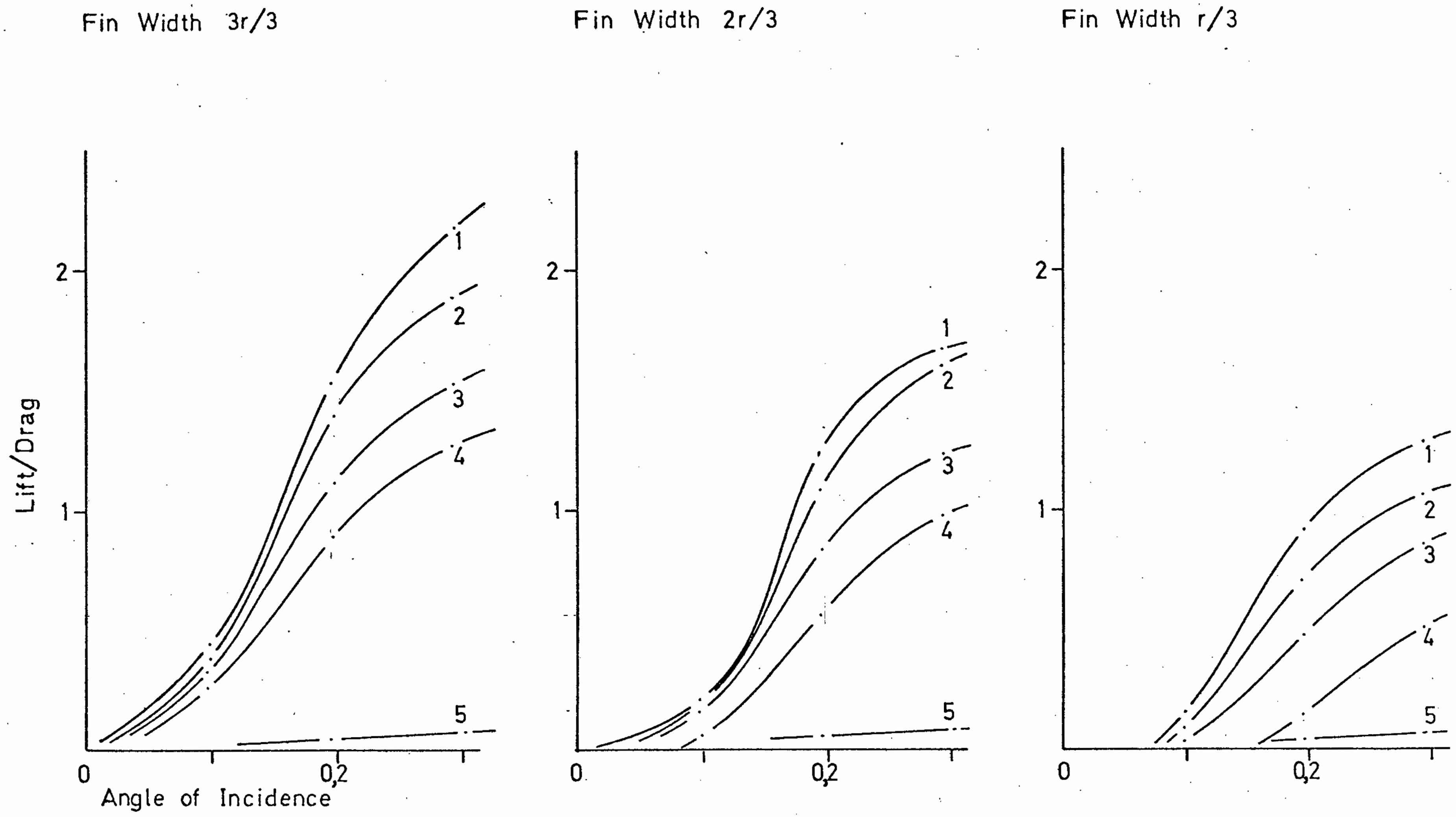


FIG C.6 LIFT/DRAG RATIO FOR DESCENT

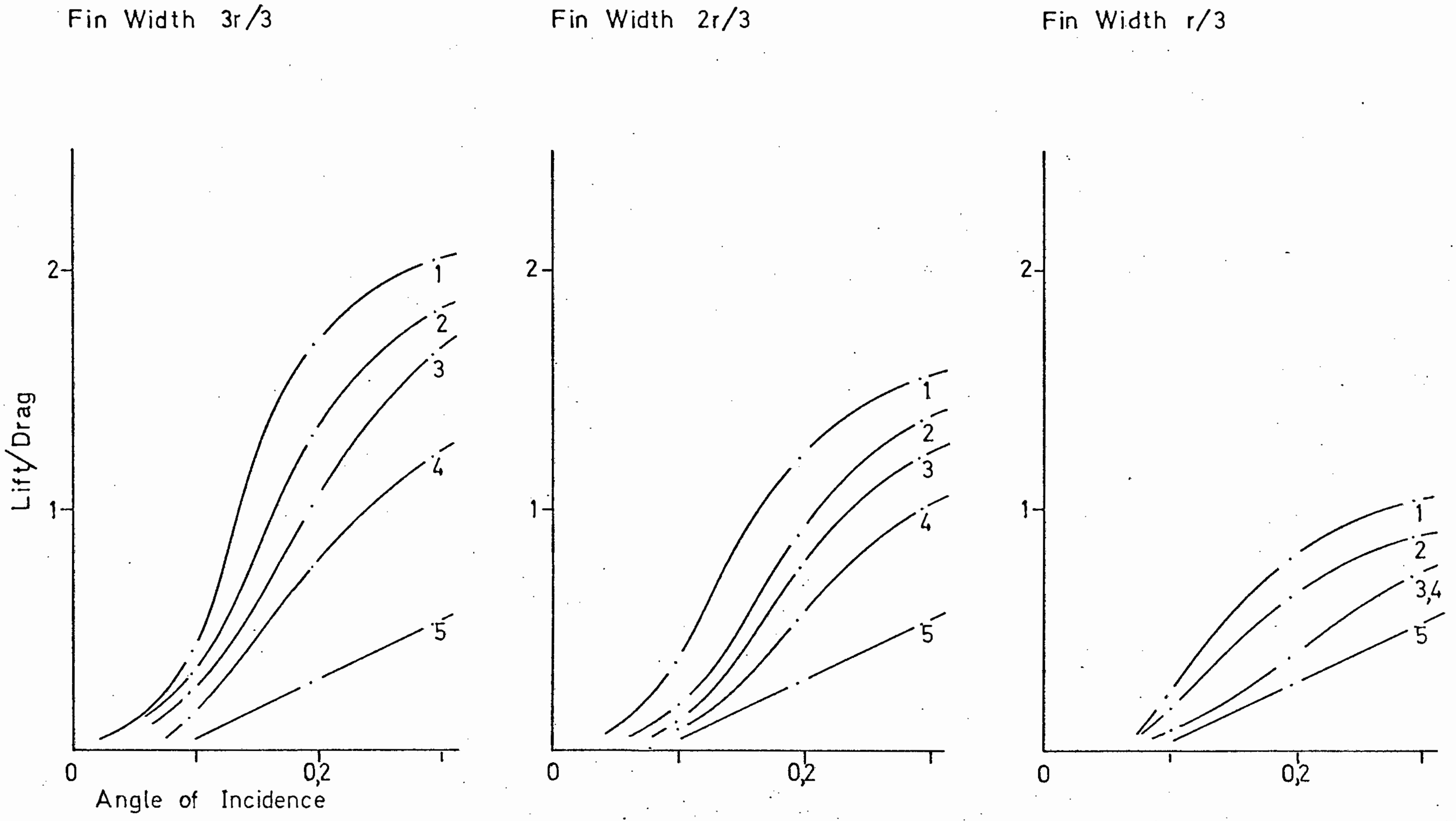


FIG C.8 LIFT/DRAG RATIO FOR ASCENT

Appendix D. Operating Instructions.

This series of programmed instructions should be read in conjunction with Section 5, which describes the testing and operating procedures and the reasons for the steps described.

Start

Fully fit out

1. Test in static tank

Is the vehicle laterally balanced?
If yes continue
If no balance and go to 1

2. Is the towing position correct?
If yes continue
If no correct and go to 2

Does the vehicle assume the appropriate positions at the switching levels?
If yes continue
If no balance longitudinally and go to 2

3. Are the nose joints watertight?
If yes continue
If no repair joints and go to 3

Weigh

Remove from tank

4. Are the fins the correct shape?
If yes continue
If no reshape the fins and go to 4

Take to the ship

5. Are the air bottles full?
If yes continue
If no fill the bottles and go to 5

6. Are the switching levels correct?
If yes continue
If no adjust the levels and go to 6

7. Are the batteries fully charged?

    If yes continue
    If no charge batteries and go to 7

Does the forward valve open when the circuit is closed on land?

    If yes continue
    If no check circuitry and go to 7

Check recorder tape, wind and start

Remove the side panel
Bolt recorder into place
Replace panel

Attach cable through swivel to towing eye
Tie warp to stern handle

Connect circuit
Launch

Stage tests

Recover
Switch off circuit
Wash down with fresh water

Remove side panel
Remove recorder
Replace panel

Are the tests ended?

    If yes continue
    If no go to 5

Lash vehicle to deck
Remove cable, swivel and warp

Stop

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Acknowledgements.

I would like to thank the many people who, in one way or another, have helped me in the course of this thesis. My thanks are especially due to Mr. J. Busbridge and Mr. D. Findlayson of the Engineering Staff whose encouragement and advice were always at hand in moments of crisis, and to my supervisor, Dr. W.P. Boyle. Thank you too Mr. A.R. Sass. Without your green draughting tape none of this would have been possible.