

THE SOLO DUCK

Introduction

One of the criteria used to screen the many proposals for new sources of energy which were made in 1973 was the size of the ultimate resource. Any idea which could supply less than a few per cent of national consumption was penalised in consequence. A central feature of duck philosophy has always been that the use of the sea-space must be maximised. The entire emphasis was on U.K. mainland use and the spine-mounted terminator was the result.

This note describes a device which is intended as a precursor of the ultimate duck string. It may be used for island sites at an early date. It will give us experience of gyro operation and sea-bed attachments and attack the markets at present supplied by diesel generation.

Reference Frames

The very first ducks needed a mounting with stiffness and strength in both torsion and translation. The requirement for torsion was removed by the introduction of gyros. The requirement for translation support was modified when it was discovered that moderate compliance helped performance by cancellation of the backward wave. This was achieved in later ducks by the deliberate generation of an 'un-wave' through controlled movements of the stern. We now understand the need for both asymmetry and the additional degrees of freedom.

The new solo duck can retain gyros for its torsion reaction but we are also considering an inferior non-gyro solo unit for use in the event that our zero maintenance philosophy is thought to be impossible. The entire gyro design stays exactly as in the 1981 reference design. However the translation reaction is obtained through tension leg rodes

attached to piles in the sea-bed. These rodes are in the form of flat tapes wrapped around bosses on the axial ends of the duck. (See Fig 1.) The duck and its section of spine are fused together in a single body. Electrical ribbon cables run along one (or two) of the tapes to rigid cable on the sea-bed. The buoyancy margin of the duck induces a very large pretension in the tapes. On no account must this tension ever go to zero and so its value must be very carefully considered.

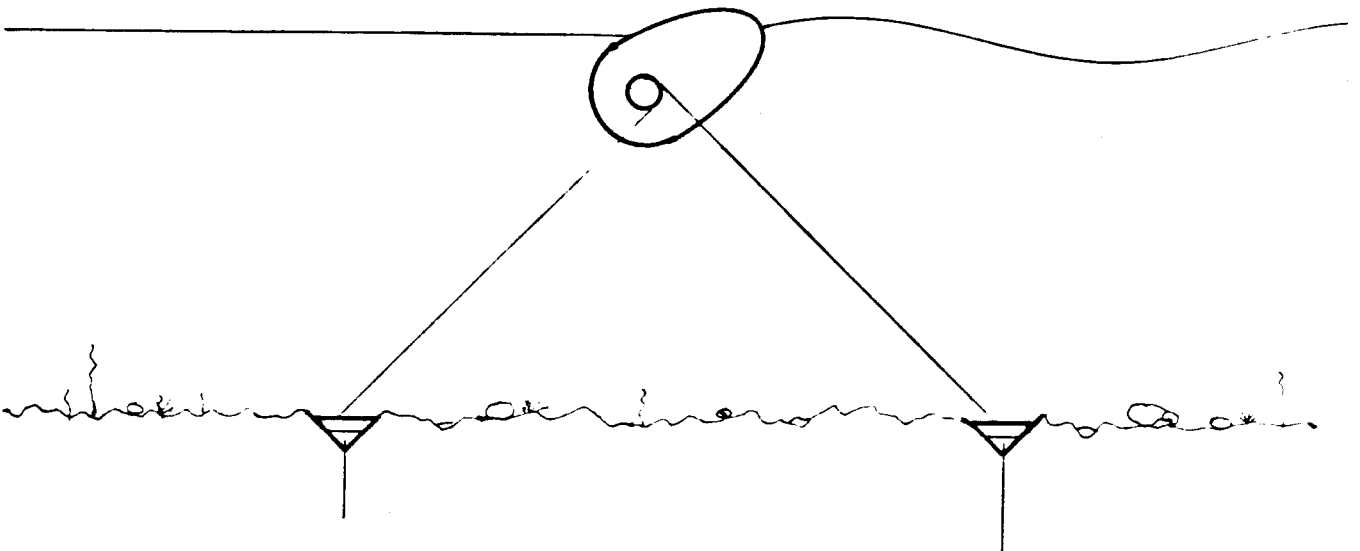


Fig 1. A solo duck with tension legs.

We can adjust the hub depth to suit tidal variation or different wave spectra by gradual rotation of the boss. Although this will require an extremely high torque (2400 ton metres) it is a very slow movement and so its sealing requirements should be easier than those for a power take-off shaft. This mechanism could also ease installation and even allow an emergency overload release for very rare, once-in-a-lifetime use.

The way in which the tapes are wrapped around the boss gives the designer a powerful control over the translations which result from rotation.

If, as in Fig 2, we have a single tape wrapped round a boss and impose a particular direction of rotation, we will induce translation along the line of the tape in either direction according to the polarity of the wrap relative to the boss rotation.

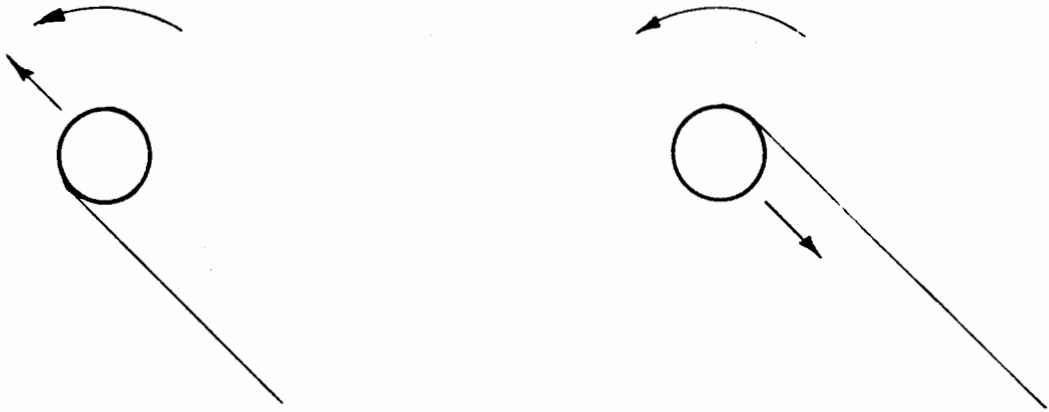


Fig 2. Rotation induced translations.

Translations at right angles to the tape are not of course defined. If, as in Fig 3, a second tape is added, this degree of freedom is removed. There are four possible ways for the body to translate.

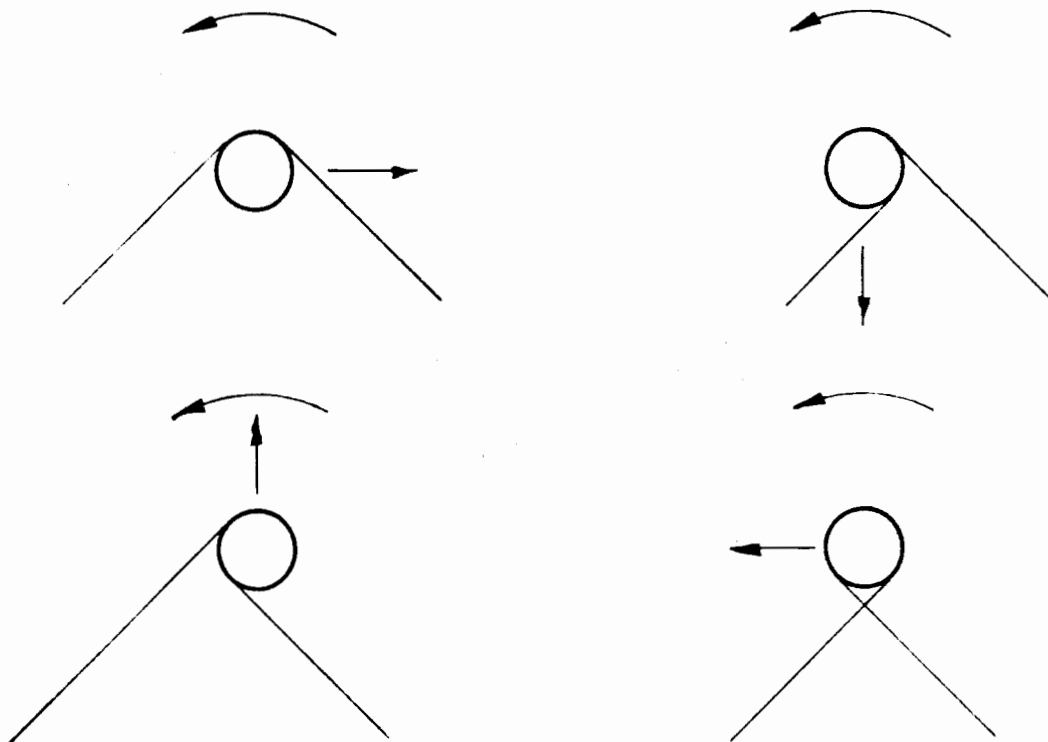


Fig 3. The four possible movements.

The magnitude of the translation is equal to the product of nod angle and boss radius. It follows that by proper choice of boss diameter and wrap polarity we can choose both the magnitude and direction of the body translation induced by rotation. We can use this to reduce (or increase) the effect of rode extension. The bosses for the aft rodes do not have to be the same diameter as those for the forward ones.

We do not need to have the bosses concentric with curvature of the stern. Nor does the stern have to be exactly round. There are therefore several possible 'un-wave' generating mechanisms. Furthermore, there is a particularly fascinating possibility of the exploitation of bosses. We can make a solo duck behave as if its mounting showed the phenomenon of *negative* compliance. Some typical efficiency contours of a compliance diagram (see pp 4.12 of our 1978 report) are shown in Fig 4.

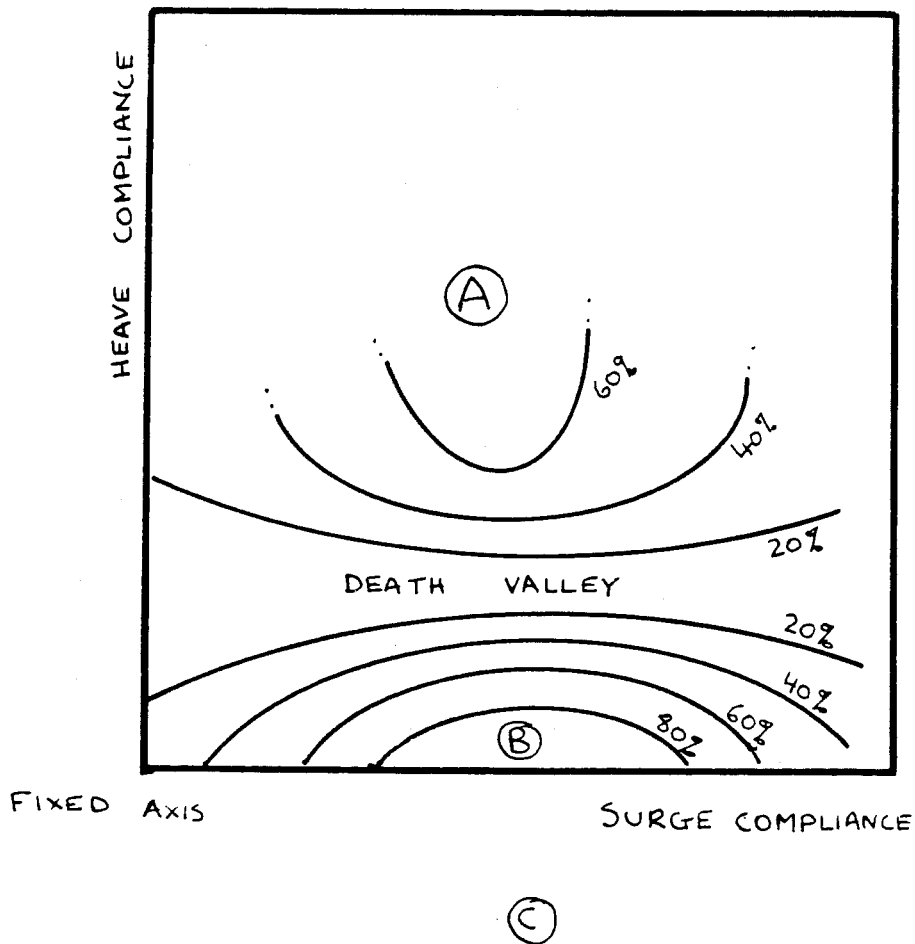


Fig 4. Efficiency contours in the compliance plane.

The point A is of high compliance in the heave direction. At point B the heave compliance is close to zero. With boss-induced translations we can also consider working at point C where the duck translations are in the opposite direction to the applied force. It is reasonable to suppose that the contours of efficiency may be something like those sketched in Fig 5.

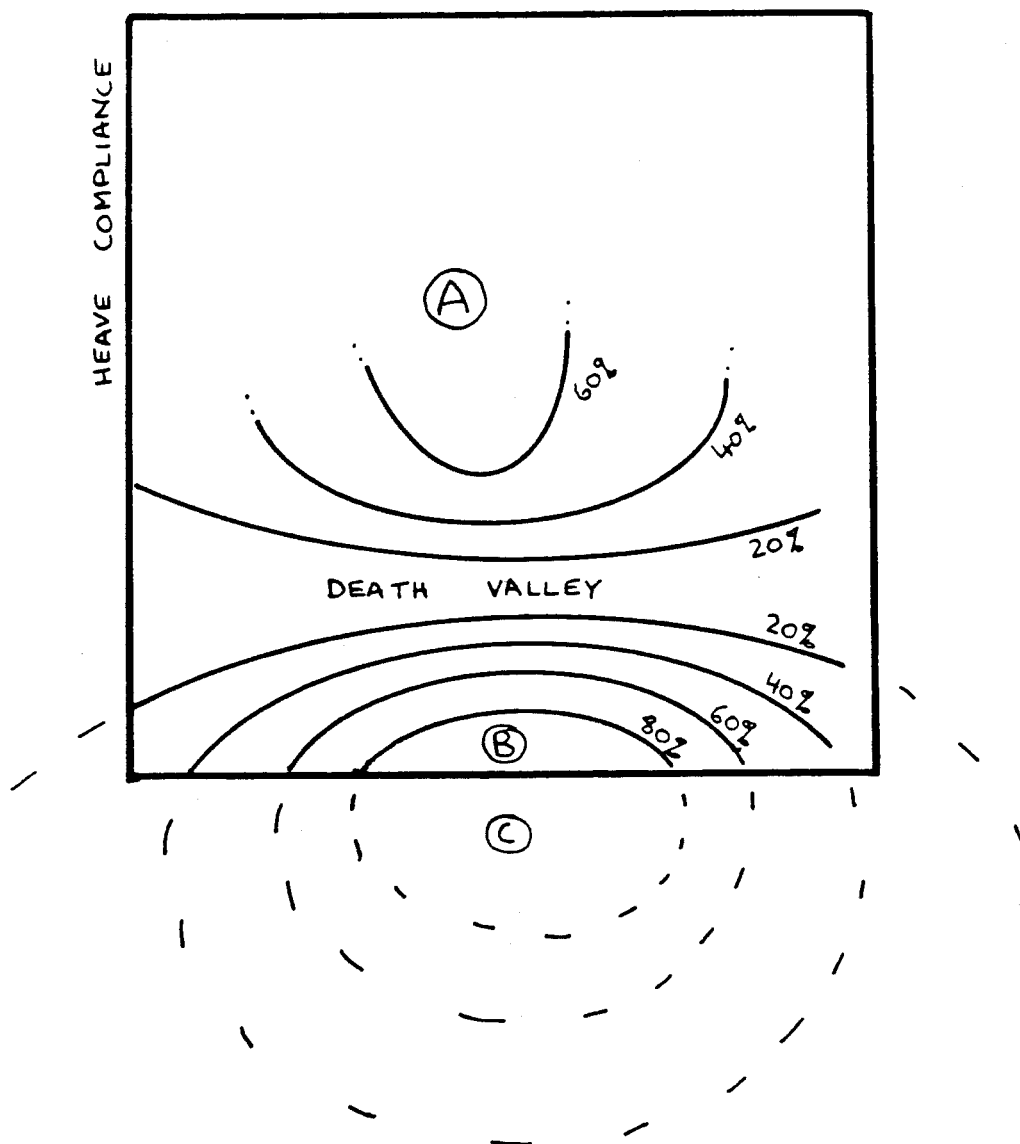


Fig 5. A possible map of the uncharted compliance plane.

Alternatively, if there is a sufficient buoyancy margin to maintain an upward bias we can move even further beyond point A. Some careful experimental work will be needed to decide which course is best. But clearly we should be quite ashamed if we cannot use those powerful design tools to generate the correct unwave for the central parts of the spectrum and thereby equal the performance of spine-mounted ducks.

*

There is some experimental evidence from the work of Bisceglia at M.I.T. that single ducks in wide tanks can achieve capture widths up to 1.85. This is well supported by the theoretical work of David Evans on point absorbers. It would not be surprising if solo ducks can improve on the output of spine-mounted versions by about 50%. Nevertheless I propose that our first cost estimates should be based on the assumption that they only equal the performance of the current 14 metre design.

Water Depth

Solo ducks have to pay a higher proportion of their cost for rodes and power cable than do long spine-mounted ones. This suggests that we should move into shallower water. The provisional figure is 50 metres so that 45° rodes would be about 70 metres long. Sites with deep water close to shore are particularly suitable.

Hydrodynamic Requirements

We need a shape which has moderate-to-large nod torque excitation without excessive force coefficients in the other directions. The wealth of theoretical and experimental data available suggests that an approximate ellipsoid with the waterline length of a 14 metre duck but the stern diameter of a 12 metre one will be correct for N.E. Atlantic conditions.

* Bisceglia, S. V., M.Sc. dissertation, M.I.T., June 1978.
"Economic Feasibility of Cam-Type Wave Power Generators".

Civil engineers do not like shuttering with double curvature and so the ellipsoid shape is approximated by circular cylindrical sections.

We are using the bulbous front and high freeboard of D0027 which maintains linearity up to larger waves than D0019. The very soft corners of the ellipsoid should reduce fluid loading in extreme conditions.

The first design value of the width inside bosses is 37 metres. However the experience of McAlpine with the Evans' cylinder suggests that perhaps this should be increased. The decision depends on whether the extra fluid loading can be taken by the rodes or reduced by the soft corners. I am sure that as our confidence in tape performance increases we will move towards wider units.

The effects of increasing width will be:-

1. Power will increase at slightly less than the increase in width because point-absorber effects are slightly reduced.
2. Rode tensions will increase directly.
3. Sway motions will reduce, probably directly, because the same or slightly reduced fluid loading will be acting on a larger inertia.
4. Structure costs will rise less rapidly than the increase in length because the middle is cheaper than the ends. However . . .
5. Bending moments in the central section will become larger. We have a superb shape for resisting them and we should be using this fact to a comfortable limit.
6. Reliability will improve because more gyro canisters can be added.
7. Performance in short-crested seas will fall as coherence across the front is reduced. We probably have some way to go before this is serious.

All these effects require careful harmonising of data on wave climate, tank behaviour and civil construction costs.

Civil Construction

The bosses will be made from steel cylinders which can provide admirable attachment points for post-tensioning wires. These should be in two layers with opposed helical rotation running along geodesic lines from one boss to the other inside the concrete skin. This puts the entire shell into biaxial compression and provides an extremely efficient structure. A strong man can not break a hen's egg by palm pressure along the major axis.*

The Rodes

We have enough test data on conventional ducks and the Evans Cylinder to give a good estimate of what the rode forces will be. In comparison to the mooring requirements of very long spines these forces are enormous. But the work of the McAlpine team and Du Pont's development of Kevlar have given me much more confidence in the engineering of high load connections to the sea-bed. The initial estimate is that a mean tension for each rode will be 600 tons with an ultimate, once in 50 year, duty of 1600 tons. There are a number of mechanisms which could allow this peak value to be reduced but it is not clear whether they will be cheaper than putting in more Kevlar.

The tape must be wrapped and unwrapped many times round a boss without being damaged by bending fatigue. This is why we use a thin, flat tape rather than a round rope. The requirement is difficult but the specification is exactly defined and rig testing of short, narrow samples not very expensive.

Quite apart from its remarkable tensile properties (U.T.S. is $2.65 \times 10^9 \text{ N/m}^2$), Kevlar is unusual in that its price has *fallen* by 20% over the last five years when nearly everything else has risen by a factor of two. As Du Pont recover more of their research investment and as their patents reach the end of their life we can expect this trend to accelerate. The cost of large Kevlar rodes was quoted on p. A-11 of WESC 78 MA P14 as £0.3 per tonne metre. This is confirmed by a check on the 1982 retail price

* uncracked, free-range eggs only

of a single 25 tonne crane sling at £0.50 per tonne metre. Polyester parafil is quoted at £0.17 - just over half the Kevlar cost. Kevlar is probably superior for bending and we can expect the ratio to close in future.

Steel wires cost only £0.06 per tonne metre but could not withstand the bending around the drum. We have to decide whether or not to take the tapes all the way to the sea bed or risk the unreliability of an extra splice to a steel wire. The inferior performance of steel in fatigue suggests that we should pay for Kevlar all the way and hope that its cost reductions continue.

Tape Specifications

| | Forward | Aft |
|---|------------|-----------|
| Slope | 45° | 45° |
| Length | 70 m | 70 m |
| Width)) of Kevlar | 1500 mm | 850 mm |
| Thickness) | 8 mm | 8 mm |
| Boss diameter | 4000 mm | 4000 mm |
| Mean tension | 600 tonne | 600 tonne |
| Ultimate (20m wave) | 1600 tonne | 900 tonne |
| Total tonne metre product (including side rodes) | 350,000 | |

Terminations

I am advised that in all the experience of the oil industry in the North Sea none of the many mooring failures has occurred by the breaking of a rode. The problems are always at the terminations. The properties of Kevlar are so remarkable that achieving a satisfactory termination will be

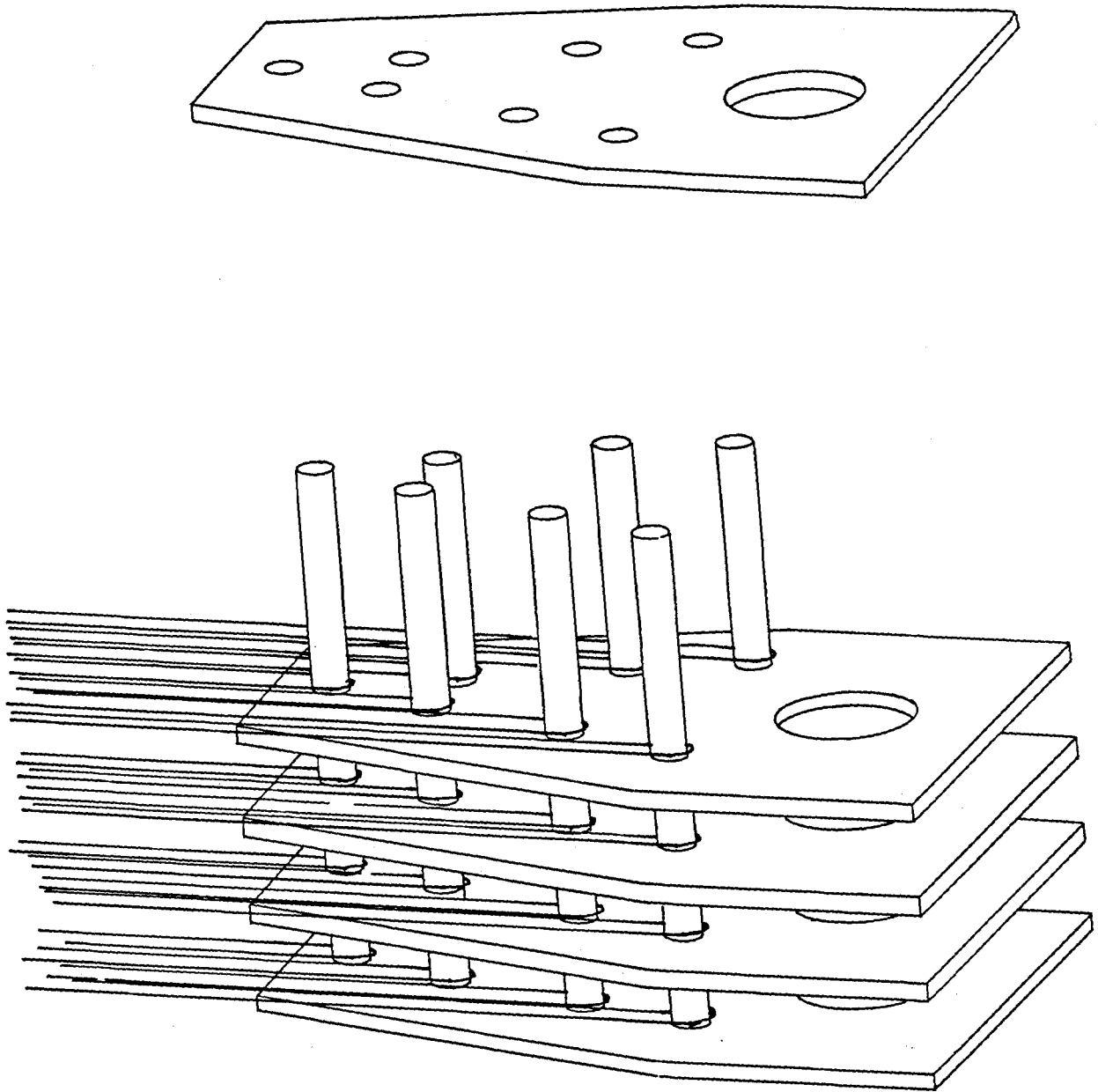


Fig 6. The lower rode termination during assembly before encapsulation in polyurethane.

even more difficult than previously. One thing is certain: we cannot snip a length of rope from a coil and hope to staple, crimp or glue some kind of end piece to it.

It would be very nice not to have terminations. There are two ways in which this may be achieved. Both involve dedicated manufacture of a particular length in the factory. The first is to make a double length of endless belt. However we must beware of inter-layer abrasion where the two belts are wrapped around the drum.

In a second, preferred, approach, the yarn is arranged in a boustrophedral constant tension lay many times back and forth around a pattern of end pins passing through terminating plates. Material from fresh bobbins is progressively twisted in with those bobbins which are partly used. The entire assembly is sandwiched inside layers of polyurethane. Fig 6 shows end details.

Both methods demand reasonably accurate knowledge of the intended water depth.

Pile Design

The use of a non-yielding attachment to the sea bed runs counter to all previous duck philosophy. If we are to break with tradition we must do so with extreme caution.

The proposed pile design exploits the excellent compressive strength of Lewissian Gneiss. Alternative designs for sand, gravel and mud beds are being considered. The values quoted in the following description are for the forward pile rated at 1600 tons.

A single vertical hole 200 mm diameter by 25 metres deep is drilled for each pile. An explosive charge is lowered 6 metres down the hole and detonated to produce a conical crater just over 7 metres in diameter. The fragments of debris remaining are air filtered away.

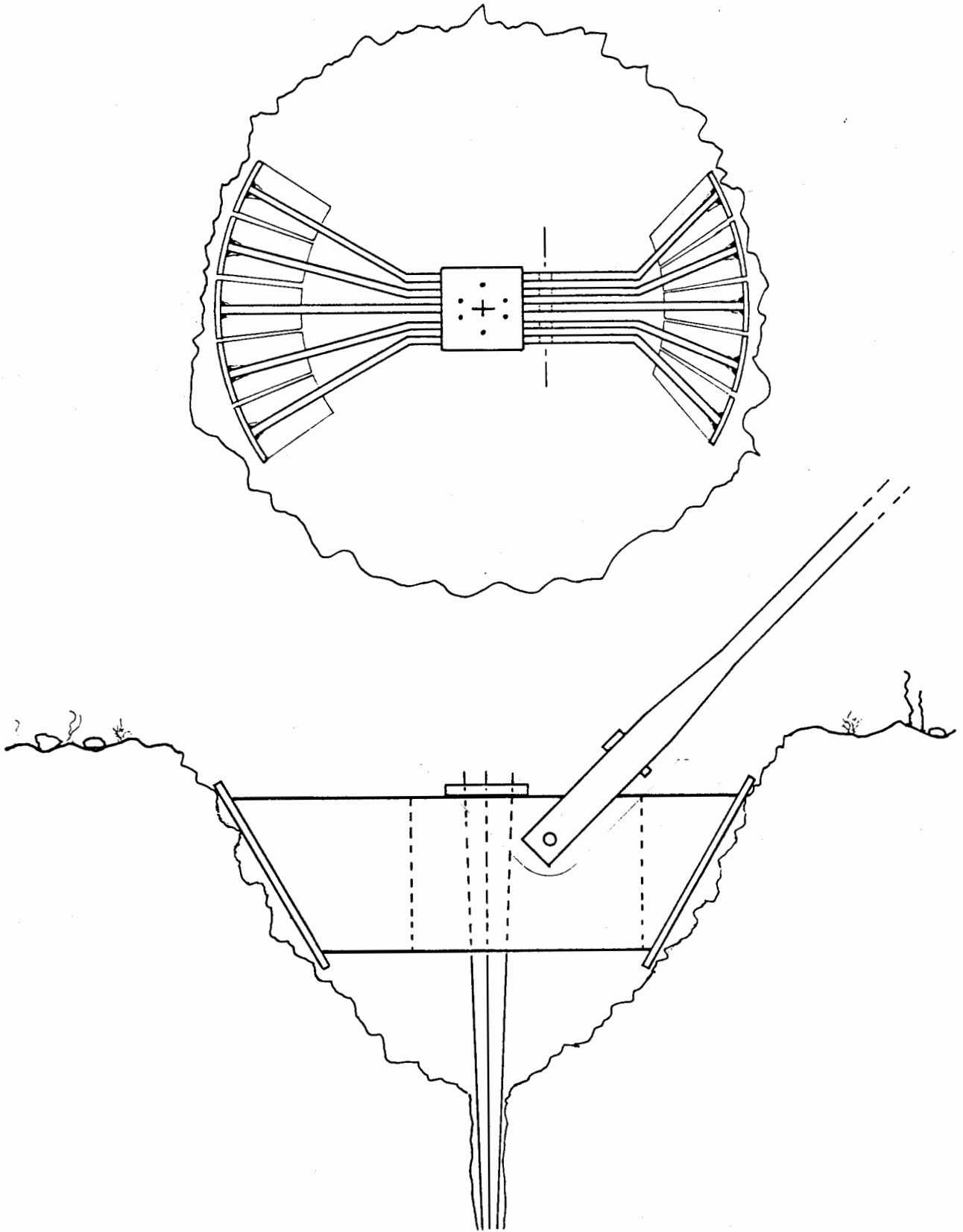


Fig 7. The pile head

The pile-head consists of a stack of 5 trapezoidal sheets of steel with interleaved spacing plates. They are bent and welded as shown in Fig 7 to produce a fraction of a cone with internal radial webs and a series of holes for a shackle pin offset a little from the centre.

Six 65 mm diameter reinforcing bars are passed between the plates and the assembly is placed in the crater. Grout is pumped to the bottom of the hole so as to rise to the lip of the crater but rubber sleeving prevents the grout engaging the bars for the upper $\frac{1}{4}$ of their length. The vicinity of the shackle pin and the post-tensioning threads is also protected. The protection device can provide moulding for reference points for any handling aids. The most obvious of these is a "vee block" trough in line with the shackle-pin holes. After the grout has cured, the bars are tensioned to 200 tons each. This tension will be constant so that fatigue will not occur. The steel tendons are protected from corrosion by the alkaline grout.

The individual tendons of the Kevlar are separated into five groups which are terminated between six steel plates. These interleave with the five plates of the pile-head and are secured by the pin through the entire stack. Because we have a tenfold shear the pin diameter is quite moderate - about 120 mm. Both it and the post-tensioning nuts can be fully immersed in high-density grease.

This pile-head design uses about 9 tons of steel plate and 4 tons of post-tensioning bar. At the standard rates of the Consultants' Working Paper No 36 this amounts to £16,500 for the two forward pile-heads.

The reduction in costs relative to those of WESC 82 MA 135 is the result of the application of post-tensioning techniques. Similar ratios can be achieved where post-tensioning replaces reinforcement in concrete. Even though the force rating is four times greater the weight of steel is reduced by a factor of nearly three resulting in proportional savings in craneage and vessel costs. Note too that all welds are in compression and all the steel is working comfortably hard. Only a single rock hole is drilled instead of the fourteen holes of the Wimpey design.

Summary of Pros and Cons Relative to the Long-Spine Ducks

| Item | Remarks |
|------------------------------|--|
| Separate Spine | Removed |
| Spine generation | Removed |
| Spine power collection | Removed |
| Radial duck-to-spine bearing | Removed |
| Axial duck-to-spine bearing | Removed |
| Joints | Removed |
| Joint rams | Removed |
| Skin | Totally convex - good for concrete shells resisting external pressures. cf. eggs? |
| Ballasting | Large buoyancy surplus. Greater freedom to adjust values of inertia and centre of gravity. |
| Usable internal volume | Increased by at least two |
| Weight | Reduced because of tension legs - saving is ~ 1700 tons |
| Surface area and shuttering | Nearly halved |
| Gyro canister | Unchanged |
| Electrical cable spools | Removed |
| Size | Could be made smaller without risk of weight penalty |

| Item | Remarks |
|---|---|
| Damping coefficient | Reduced because of point absorber effects |
| Torque limit | Unchanged |
| Work limit | Unchanged |
| Power limit (for the same torque and work limits) | Increased or left the same with better duty factor |
| Water depth | Reduced |
| Distance from shore | Reduced |
| Section moment | Increased |
| Concrete stresses | Lower than old spines, higher than old ducks |
| Mooring forces | Much bigger and more variable |
| Use of sea front | Much worse |
| Hydrodynamic design | More interesting |
| Efficiency | Possibly 50% higher because of point absorber effects |
| 132 kV distribution | Unnecessary for local use |
| Power collection costs | Higher |
| | |
| | |
| | |

The Snags

There is a well-known phenomenon in wave energy known as first-sight infatuation which makes the newest device, about which least is known, appear the most attractive. (It should not be for me to suggest that the oldest device about which most is known should be our real true-love.) The snags do not appear until the models are in the tank and RPT begin their work. From then on it is a race between RPT thinking of nasty problems and the device teams thinking of solutions. I will now pause to give the reader the chance to identify the first major problem.

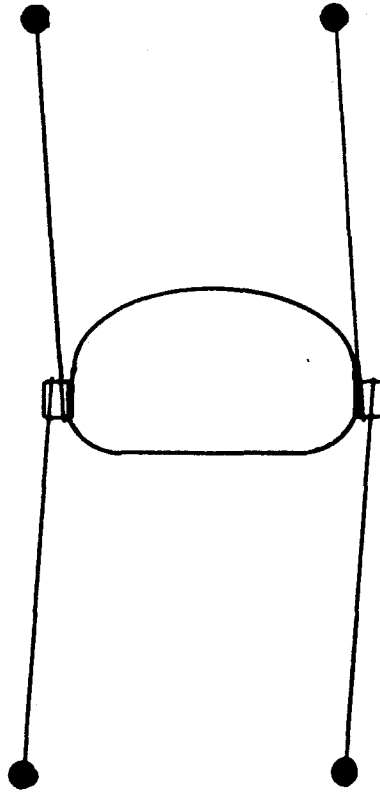
Snag 1

Fig 8. Says it all.

The tapes will work splendidly in a narrow tank. Small models will look convincing in a multi-directional mixed sea. But the gentle scuffing of a cotton thread on the flanges of its reel becomes a major problem when scale is increased and the tension is a thousand tons. We have to work out what happens when the solo duck sways.

The first question to ask is how big the amounts will be. The news from McAlpines is encouraging. Their cylinders do not sway noticeably in ordinary multi-directional seas. When very large seas approach from large angles to the normal they predict a maximum excursion of 5 metres. If our

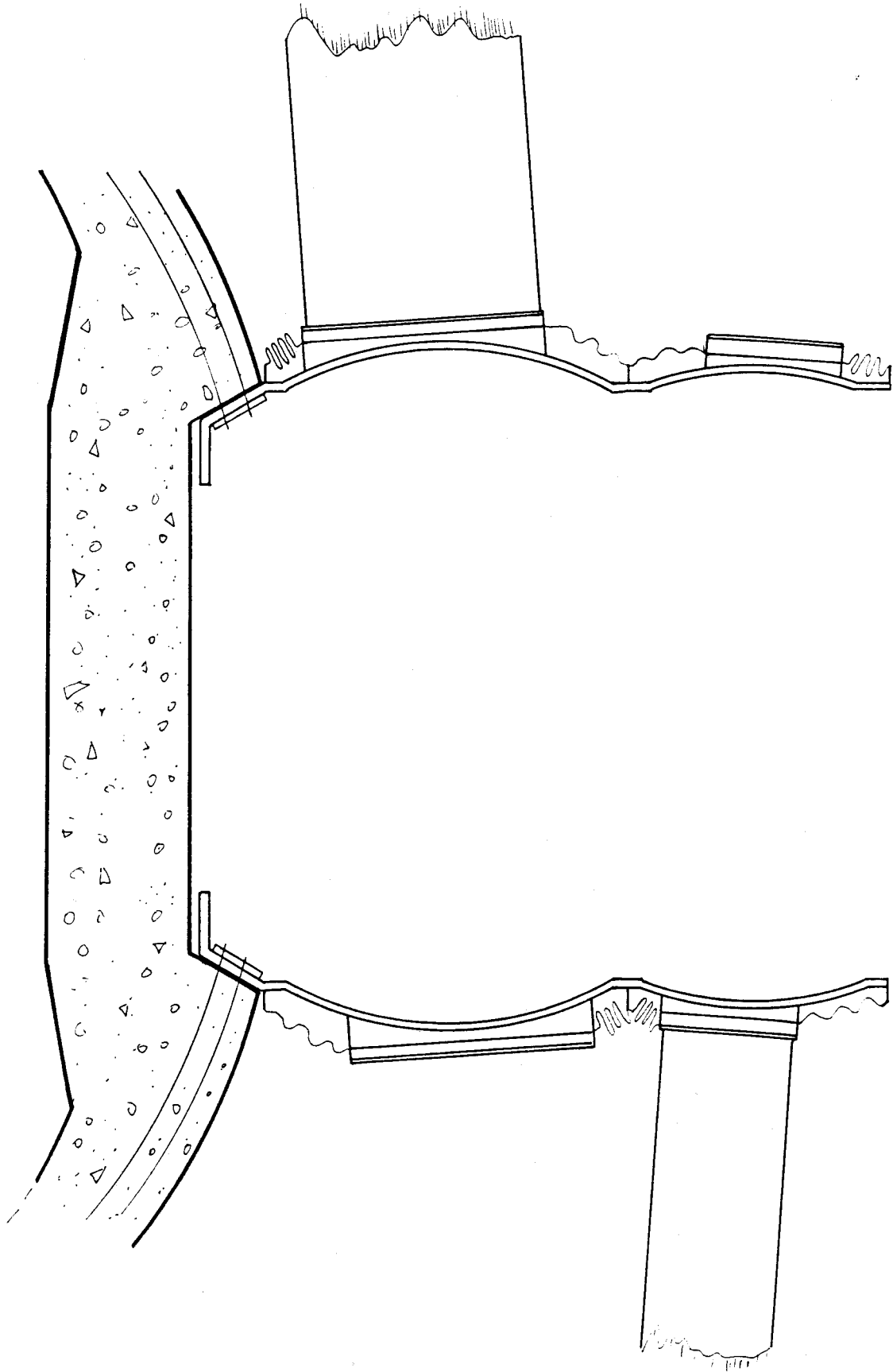


Fig 9. Spherical bosses to prevent damage due to sway

rodes are 70 metres long, the tapes will have to move through an angle of 4.1° . Although this seems a small angle, and side rodes could reduce it, we must not allow any chafing at the junction of boss and tape.

The only certain escape is to provide a separate tape drum with spherical "swash" freedom but torsional restraint on the outside of the boss. This spherical motion could be provided by elastomeric bearings in shear. But as the motion allows full sealing with gaiter bellows, and as we are keen to respond to the very lowest perturbing force, it seems that hydrostatic bearings will be more suitable. At 1600 tons the pocket pressure will be less than 400 psi. It is important to remember that the amount of motion is very small.

Although it is tempting to use the high-pressure oil supplies inside the power canister and although the spherical bearing can operate in vacuum-stripped conditions we believe that the sanctity of the power canister should not be risked. A separate oil system with high-viscosity oil powered from the electrical outlet should be used. A section of the assembly drawn at its maximum offset is shown in Fig 9.

I am sure that it is right to spend a great deal of design effort and money on nursing the tape-to-drum interface. A satisfactory mechanism to limit sway, or to allow sway to occur without tape damage, will allow us to reduce water depth. This will save a great deal of money on Kevlar, rodes and electric cable. Side rodes seem to be the most obvious choice and their costs are allowed for in the estimates. Fortunately refraction in shallow water will make the waves approach from angles closer to the front and thereby reduce the sway-exciting forces.

We need not be so concerned about yaw and roll because these motions are resisted by the rodes. Furthermore it is possible to arrange that both these modes are damped (and that energy is extracted by gyro action).

Snag 2

We should also devote careful thought to the risks of tapes becoming slack and unwound during the extreme trough. The largest force on a duck

is downwards and forwards in such a way as to reduce the tension in the forward rode. A rigidly-mounted duck would be subjected to extreme forces very close to the value of pretension. But the density of Kevlar is 1400 kg/m^3 . At full scale the weight of the tapes, even in water, is substantial. We do not expect downward acceleration greater than one third g, so that the tapes' own weight should maintain sufficient tension to prevent unwrapping. Furthermore, any fluid loading on the tape itself can only increase this tension. If model testing reveals a problem the tape weight can be increased and the value of pretension increased. Although this costs more in Kevlar it saves money on concrete. The design can stand a substantial increase in pretension if this is necessary.

Snag 3

To be identified.

Output

The consultants' 1982 estimate for the output of a string-mounted duck in the South Uist wave climate was a mean delivery of 595 kilowatts. This included an assumption about the wave climate for 100 metre water depth which we believe to be incorrect but which would be about right for 50 metre conditions. It also included a target availability figure of 80%. A perfectly reliable device would therefore produce mean output of 740 kilowatts. Arguments about the delights of working conditions in the gyro canister continue, with a factor of 300 between conflicting opinions. We believe that 90% availability can be achieved.

Capital Costs for 2.4MW Generator Rating

(where possible given in consultants' working paper # 36)

| | £,000s |
|---|------------|
| Concrete 4500 tonne at £58 /m ³ = £24.2/tonne | 109 |
| Form work 2500 m ² at £12.8 /m ² | 32 |
| Reinforcement 90 tonnes at £543 /tonne | 49 |
| <hr/> | |
| Post tension wire 50 tonnes at £1480 /tonne | 74 |
| Power canisters complete assembled | 562 |
| Rodes 350,000 tonne metre at £0.3 /tonne m | 105 |
| <hr/> | |
| Piles: 2 off at £16,500 each | 33 |
| 2 off at 9,500 | 19 |
| † 2 off at 6,000 | 12 |
| 6 holes at 8,000 | 48 |
| <hr/> | |
| Bosses + bearings 2 off 30 tonne each at £2,000/tonne | 120 |
| Cable to sea bed including plug | 10 |
| * Cable to shore say 3 km at £90/m shared between 6 ducks | 45 |
| Tow out, install, explosives, grout etc. | 100 |
| | <hr/> |
| | £1,318,000 |
| | <hr/> |

During the nominal 30 year life of the scheme we expect the chief maintenance work will be to change the rodes at 10 and 20 years. If it costs £25,000 for sea operations the total cost will be £130,000. However these are future expenses. Assuming a 5% discount rate they will have discounted values of £79,800 and £49,000 today. The present capital allocation is therefore:

| |
|-------------|
| 1,318,000 |
| 79,800 |
| 49,000 |
| <hr/> |
| £ 1,446,800 |
| <hr/> |

Credits for gyro storage are not included.

† A group of solo devices can share side rode pile heads.

* Cable costs to shore depend on range and number of devices deployed.

Electricity Costs

In addition to the repayments of capital we have to allow for costs of maintenance. An annual figure of 3% of the original capital cost has been suggested from experience with fossil-fuel stations. However a much lower figure is achieved in land-based hydro schemes. Large wind turbines generally use figures of 1% or 2%. We expect that our chief maintenance activity will be anti-fouling work on the rodes and suggest the 2% will be about right.

The costs of a kilowatt-hour of electricity from a duck with capital cost as given on page 20, a two year construction period, a discount rate of 5% over 30 years, and the consultants' productivity at a typical South Uist site are given for various availabilities and maintenance charges as follows:

| | | Maintenance charge % of original capital | | |
|--------------|------|---|------|------|
| | | 1 | 2 | 3 |
| Availability | 100% | 1.67 | 1.89 | 2.11 |
| | 90 | 1.85 | 2.10 | 2.35 |
| | 80 | 2.09 | 2.36 | 2.64 |

If point absorber expectations are fulfilled these figures would be reduced in direct proportion. We may expect an increase of output to about 900 kW mean (at 100% availability) and cost reductions of about 25% on the figure above.

Output in other wave climates cannot be calculated without access to the appropriate wave data. A climate with lower raw power input but lower extremes could lead to reduced force ratings. A consistent input like that of the trade-wind region could lead to very economical designs with lower stresses and power ratings but higher duty factors.

Summary

The solo duck is a hard-skinned, factory-built, bottom-referenced, asymmetric, single-body, terminating point-absorber with controlled mounting translations and the option of exploiting negative compliance. It uses hermetically sealed, vacuum-stripped, high-pressure oil for intelligent, continuous, non-linear, gyro power take-off, with bulk, front-end storage and synchronous, mean-rated generation. It makes heavy demands on the properties of Kevlar for its tension legs, is prodigal in its use of sea space and must be carefully designed to avoid sway. It may achieve higher efficiencies than those of ducks in a string, for which it will be an excellent test-bed.

S. H. Salter

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