(Paper given at Third European Wave Energy Conference, 30th September - 2nd October 1998, Patras Greece)

Proposal for a Large, Vertical-Axis Tidal-Stream Generator with Ring-Cam Hydraulics

S H Salter

Department of Mechanical Engineering, Mayfield Road, University of Edinburgh EH9 3JL, Scotland

ABSTRACT

This paper is based on work carried out for the European-Chinese Synergy programme. It shows how ideas for electronically-controlled ring-cam pumps with very large torques and variable displacement, which were originally developed for wave energy, can be used in a tidal-stream generator with a vertical-axis rotor. The arrangement allows all the generation plant to be at the surface in an accessible, sealed compartment at atmospheric pressure. The proposed rotor uses variable-pitch blades where pitch is set by control of the moment about the pitch axis. Hydrostatic bearings use a set of compliant master-slave pads to allow large geometrical distortion. Self-propulsion is possible. Some design parameters are presented.

1.0 WIND-WATER COMPARISON

Although there are some similarities between wind-turbines and tidal-stream ones, it is important to understand several differences. The velocities in the best tidal streams are about one fifth of typical wind cut-out speeds, but over the time-scale of tens of minutes they are much steadier and also accurately predictable. There is less need of storage and easier integration with network planning. Blades in water can suffer cavitation damage which severely limits tip-speed ratios. Water rotors will therefore have higher solidity than modern wind-turbine designs. The lower rotor-to-current speed ratios enforce the need for both variable-pitch and variable-speed if stall is to be avoided. Gravity loads are a major problem for wind but in water they can be accurately opposed by buoyancy.

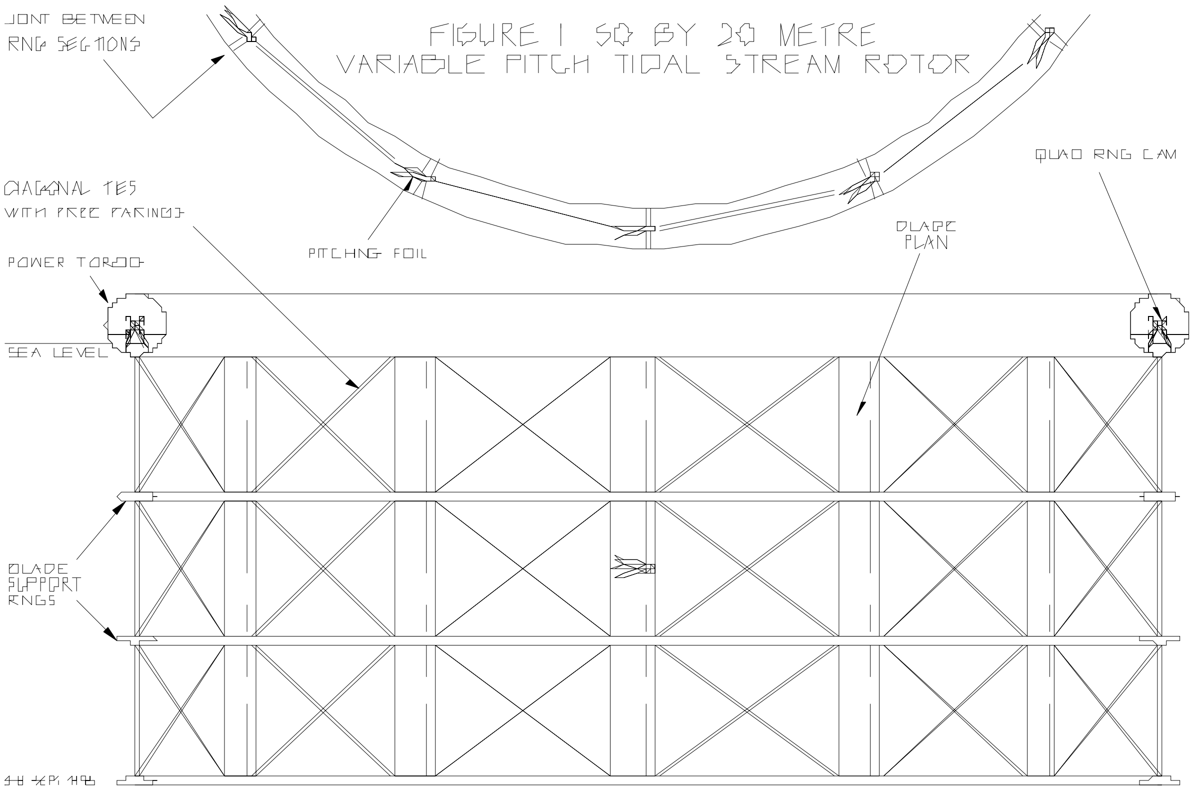
The transport of parts of the largest wind turbines on land is a major problem but floating objects of any likely size can be moved very cheaply. Indeed the proposed rotor design bears such a strong resemblance to the Voith-Schneider propeller that we should seriously consider making them self-propelled, with fuel tanks to give trans-ocean capability. Centrifugal effects are important in wind energy but are negligible for tidal streams. Wind plant in good coastal European sites will be subject to rain and salt spray and so corrosion problems will be similar. Indeed sea spray which has partly evaporated will have a higher salt concentration than the sea water from which it came. However bio-fouling of water turbine blades will be serious. The slower speeds mean that enormous torques, of the order of 108 Nm, and very high bearing loads are needed.

2.0 THE ROTOR

A possible design for the complete system is shown in figure 1. It consists of sets of symmetrical vertical blades supported at both ends by bearings in horizontal rings. The rings have streamlined elliptical sections with a chord-to-thickness ratio of five. This gives a large reduction in bending moments and bearing loads relative to a cantilevered support and allows a much greater total depth. For convenient transport in sea containers, a single blade should not exceed 10 metres length and 2.4meters chord. The present choice is 3 banks of 6.7 metre blades with chord of 1.9 metres.

Initial hydrodynamic analysis shows that variable pitch is essential. Fixed-pitch blades would spend too much time stalled at the necessarily low tip-speed ratio and would have a performance coefficient of only 0.2 whereas variable pitch can achieve over 0.4, which is comparable with vertical axis wind machines. We achieve variable pitch by rotating the blades about spars which join the horizontal rings.

The pitch bearings could use self-aligning hydrostatic master-and-slave pads fed with filtered seawater, like those proposed for ram guides of the Budal and Falnes buoy (Salter 1993). These bearings would be placed at about 0.05 of chord forward of the centre-of-pressure so that they would need a small but positive moment to move away from the local angle of incidence. If the bearing surfaces are made from the marine version of Glacier DU they can operate for a few weeks with no feed.



Pitch actuation is much easier if we do not attempt to fix the angle but instead fix the moment on the blades. The blades will then take up whatever angle gives the right moment and we do not have to be concerned about the change of direction of current flow. At lower angles of incidence the blade chords would be tangential but once the angle of incidence exceeds the value of maximum lift-to-drag ratio (about 9 degrees) the blades should track the angle of incidence to keep it at the optimum value. Each blade would need one hydraulic change-over valve with a central locked position. Two common fluid lines can control all the blades. The maximum pitch angle during normal operation is unlikely to exceed 30 degrees. However after any damage to moorings it might be convenient if the actuator allowed angles up to 90 degrees either side of the rotor tangent line to reduce drag.

The rotor rings are braced between the spars by diagonal ties of high-tensile steel cable fitted with rotating fairings to minimise drag. Without these fairings the drag of the ties would be enormous, wasting about one third of the power. For the very largest rotor sizes it may be necessary to cross-brace the rings with ties running from each spar attachment to a point on the opposite side ofthe rotor so as to improve elastic stability on the upstream arc of the rotor.

The upper part of the rotor drives a ring-cam of the same diameter running through a floating torus shown in figures 2 and 3. This has to be at least 3.4 metres in diameter if maintenance staff are to stand comfortably. The lower part of the torus is divided into many separate water-tight compartments. We cannot make the cam pass through the generators so these will have to be in enlarged sections of the power torus. These enlargements would make convenient attachment points for the moorings.

Many small, identical components can be assembled to make different rotor diameters. All items, including the half-sections of the power torus, will fit in a sea container. None weighs more that 20tonnes.

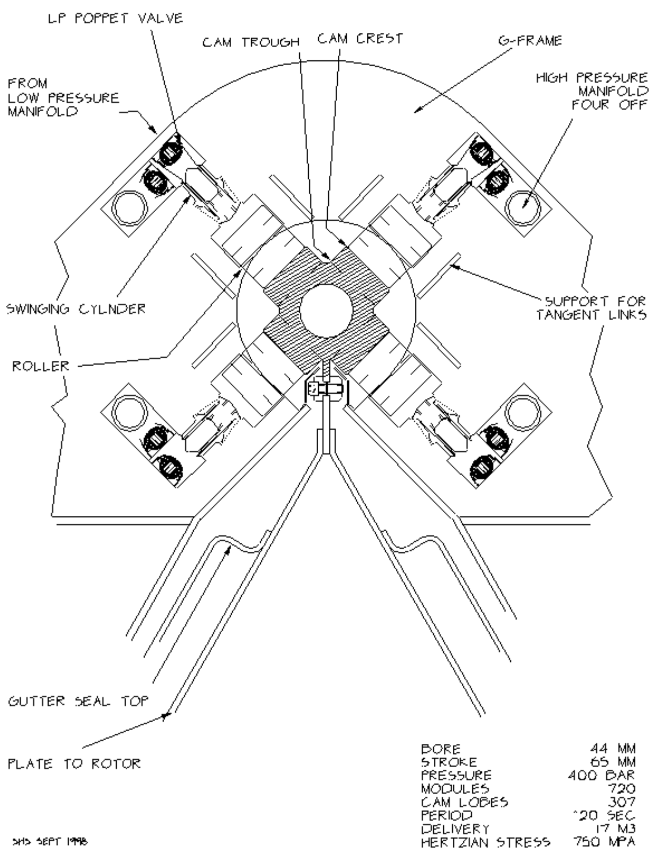


FIGURE 2. A SECTIONAL VIEW THROUGH THE CRUCIFORM CAM

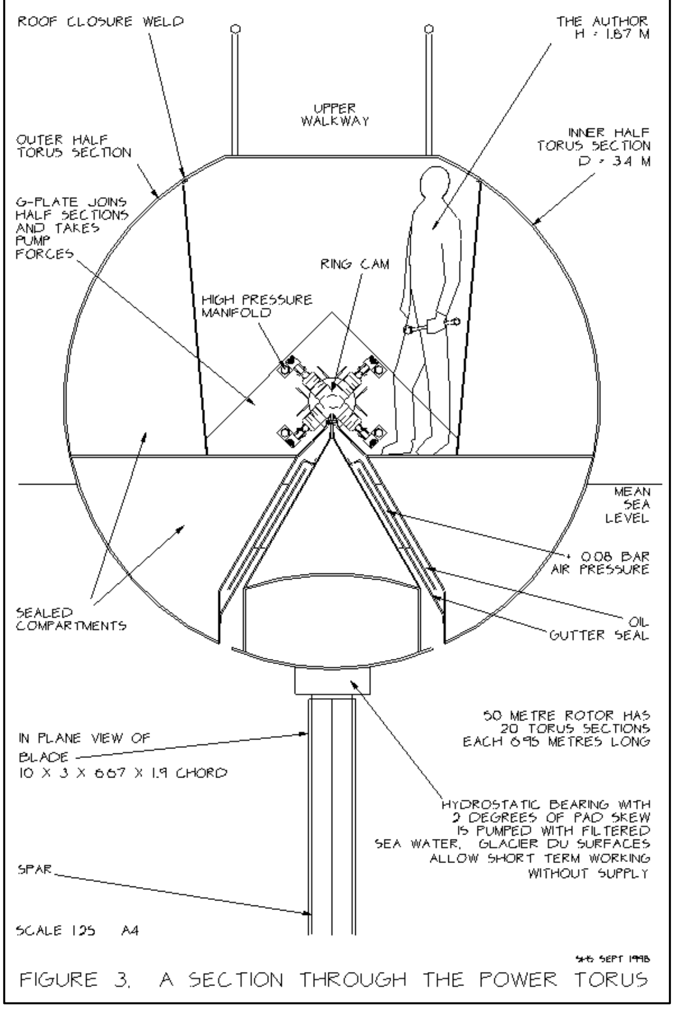


FIGURE 3. A SECTION THROUGH THE POWER TORUS

3.0 HYDROFOIL SECTIONS.

The blade thickness section is decided by the need for it to act as an efficient beam for the high distributed loading and also for it to have near neutrally buoyancy. If we restrict the maximum stress in the steel to 180 MPa and keep to one thickness of steel plate over the whole blade surface then the section thickness will need to be 18% of chord. The preference is for a new section designed by Rodewald (1998) which has been optimised for a smaller negative pressure coefficient.

4.0 EXISTING RING CAM MACHINES

Fixed-displacement ring-cam motors with a single ring are made by several firms and have for many years provided slow, accurate drives in marine uses. Their casings have inward-facing lobes that impart a radial movement to a number of roller cam-followers driving pistons. Hydraulic connections are made with rotating kidney-port valves, close to the axis, which act like the commutator of an electric motor. In each rotation of the machine, every cam-follower is driven by every cam lobe. The cost depends on the sum of lobes and followers but the output depends on their **product**. Thus machines give better value for money as size increases.

The tidal stream application needs a ring-cam diameter the same as the rotor, variable-displacement and the ability to take bearing loads. At these much larger diameters it is better to react the forces on the cam through its thickness rather than round its circumference and to split the ring into many separate lobes with scarf joints at the troughs. The separate lobes would be pulled together by a cable running through their centre like beads in a necklace. The final cam surface would be ground after assembly in the power torus. Each lobe section weighs only 45 kg.

A cruciform combination of 4 cam tracks as shown in figure 2 can also provide the functions of geometrically-tolerant radial and thrust bearings which are not available as catalogue items at 50meters diameter. Extra pumping stroke can accommodate thermal expansion. The cam is contained in a power torus with dry manned access. The gap between the power torus and the rotor has flow-restricting labyrinth flaps forming a gutter seal as shown in figures 2 and 3.

Although all the present ring-cam machines use fixed displacement, it is possible to achieve variable displacement by disabling a selected proportion of pumping modules by holding open some intake valves. The modules are enabled by signals from a computer to a magnetic coil on each valve. The technique (Salter 1993) has been described in previous work for the JOULE programme.

4.1 Comparison of ring-cams with gears

The simplest spur gear pair puts all the power through one line of contact. In an epicyclic gear-train the power goes equally through three lines. Some very special epicyclic machines with controlled compliance spindles can use five. But in the proposed very large ring-cam machine the power goes through many hundreds, even thousands, in parallel.

Gear teeth have to combine hardness and tensile strength but cams need only hardness

Gears must mesh accurately with correct centre distances and tooth profiles, even under heavy load. Ring-cams need smooth surfaces but can have tens of millimetres of initial geometrical error and distortion under load, provided only that Hertzian fatigue limits are not exceeded.

High torques twist long gears thereby changing line contacts to points but do no harm to ring cams.

A noise sensor can detect early damage to rollers and cam lobes in a ring-cam machine. The control-computer can arrange that followers can skip damaged lobes. However a single damaged gear tooth will rapidly infect all the others.

Gears have fixed velocity ratios. Ring-cam pumps can give the infinitely-variable ratios which are needed to connect variable water velocities to synchronous generators.

Selective use of the pumping modules can allow a ring cam to provide the function of the main rotor bearing and still be tolerant of gross changes of dimensions.
These factors allow ring-cams to achieve the very high torques (well over 108 Nm) with surprisingly

low weights of machinery. A 50m diameter pump suitable for use with a 20m deep rotor and able to act as a bearing would weigh only 52 tonnes, of which much is low-precision structural steel. Design calculations are available as a Mathcad document which can quickly recalculate values for any other input values.

Torque can be calculated from the product of pressure, number of cam lobes, number of pumping modules, piston area and piston stroke divided by 2.

Pumping modules are disabled by holding open plastic poppet-valves made from carbon-fibre filled polyether-ether-ketone which we have tested to full fatigue life. They have an annular shape and very low flow losses when they are disabled, about 0.2% of the energy of a working stroke. Losses will be dominated by the shear and leakage of the hydrostatic bearing between the rollers and the pads that drive the pistons of each pumping module. The Mathcad analysis uses the standard equations for leakage and shear and shows that losses should not exceed 1% of the instantaneous power, with perhaps a further 1% if the pump is used for bearing duty.

We need quite large power-torus diameters to make the system stable in pitch and roll and to allow comfortable manned access. The initial design study suggests that cost per kilowatt falls gently with rotor increasing diameter to at least 100 metres. Provided that the system design allows the isolation of faulty pumping modules, there seems to be no limit to the size of ring cam pumps.

The limit to rotor diameter may be imposed by the elastic stability of the rotor rings which will suffer compressive forces on the upstream side. Downstream-only generation is possible and might suit very big units.

5.0 MOORINGS

Large turbines will need economical ways to resist the very large (> 10 MN) down-stream and cross-stream forces. It is possible to perform all the mooring function with tension-leg cables as shown in figures 4 and 5.

Cables must be attached to the power torus at the surface. The horizontal component of tension in the cables must equal the resultant horizontal forces on the rotor. The vertical component of the cable tension is opposed by a change of depth of immersion of the torus which must have enough plan area and freeboard. If cables were attached to the most upstream point then the entire system will tend to pitch nose down. If they were attached to the downstream point then it would pitch nose up. There must therefore be some intermediate attachment point that is neutral in pitch. It will be the point that allows the line of action of all the cables to pass through the intersection of a vertical line through the centre of buoyancy and a horizontal line through the centre of pressure of all the blades.

The plan view of the cables must be rotated from the stream direction to take account of the crosscurrent force. Some slack in the downstream lines (which will tauten when the flow direction reverses) allows the torus to follow the rise and fall of the tide. This is likely to be quite small at a point of specially high stream velocity because this point should be at a tide node. In some sites it may be possible to make a cable attachment to rock anchors on land which would have many attractions compared with sea bed ones. However this would remove the vertical component of cable tension and might require careful attention to pitch stability.

A stack of parallel triangular steel plates spaced apart by about 30 mm will be lowered into it and some steel post-tensioning wire lowered through the gaps between the plates to reach the bottom of the hole. The bottom three-quarters of the hole will be filled with grout. Concrete will then be poured round the steel plates to bed them into the crater. After the concrete has reached full strength, the wires will be tensioned up to their constant working stress and so will suffer no fatigue. An increase of mooring force will change the compressive rock pre-load.

The stream velocities at a good site are above the safe levels for divers except for a few minutes either side of slack water. The sediment entrained in high velocity streams may well reduce visibility to zero. The drilling and placement of charges and installing of the pile hardware will therefore have to be done from a jack-up platform or a purpose-designed, remotely operated vehicle.

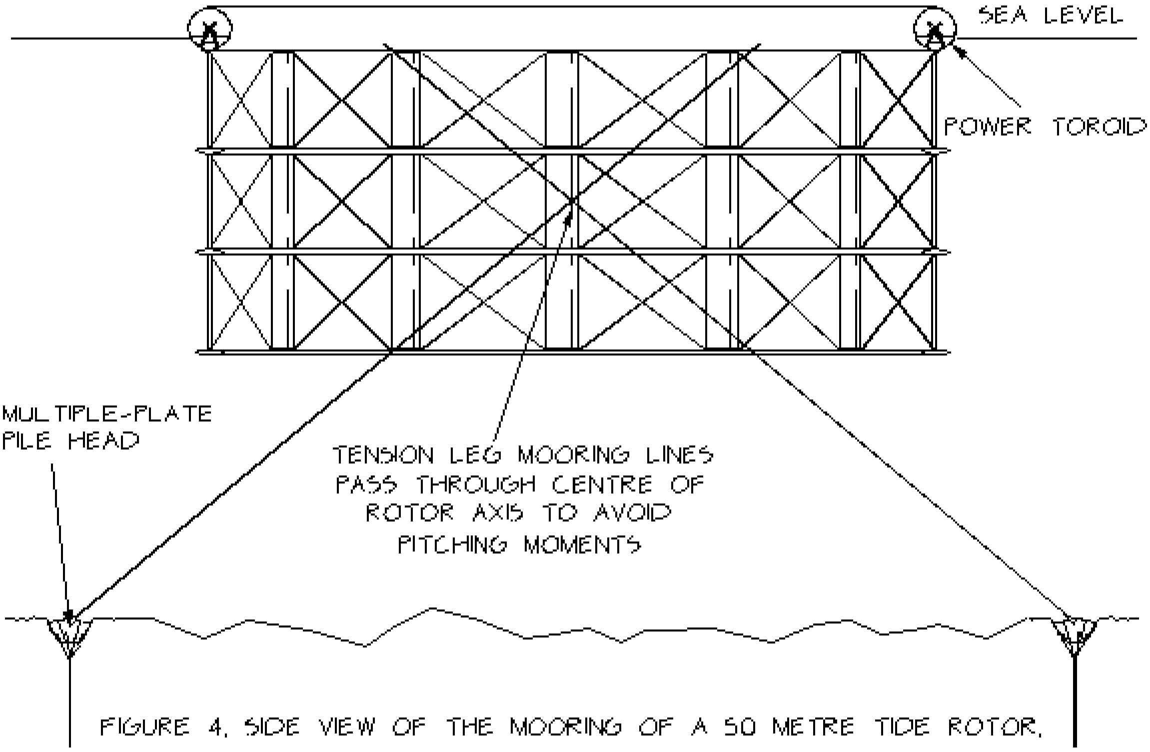
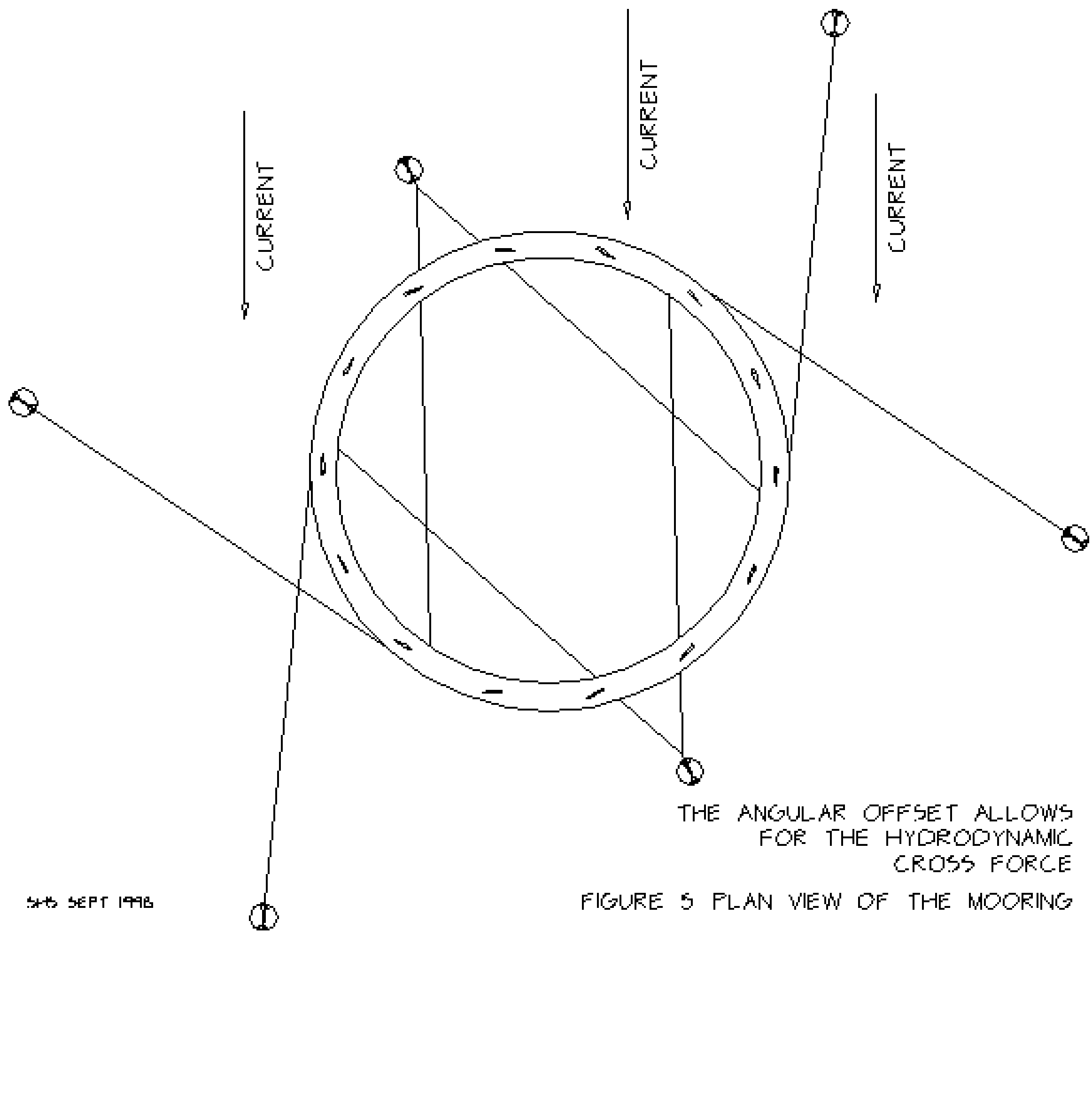


FIGURE 4. SIDE VIEW OF THE MOORING OF A 50 METRE TIDE ROTOR.



SHS SEPT 1998

FIGURE 5 PLAN VIEW OF THE MOORING

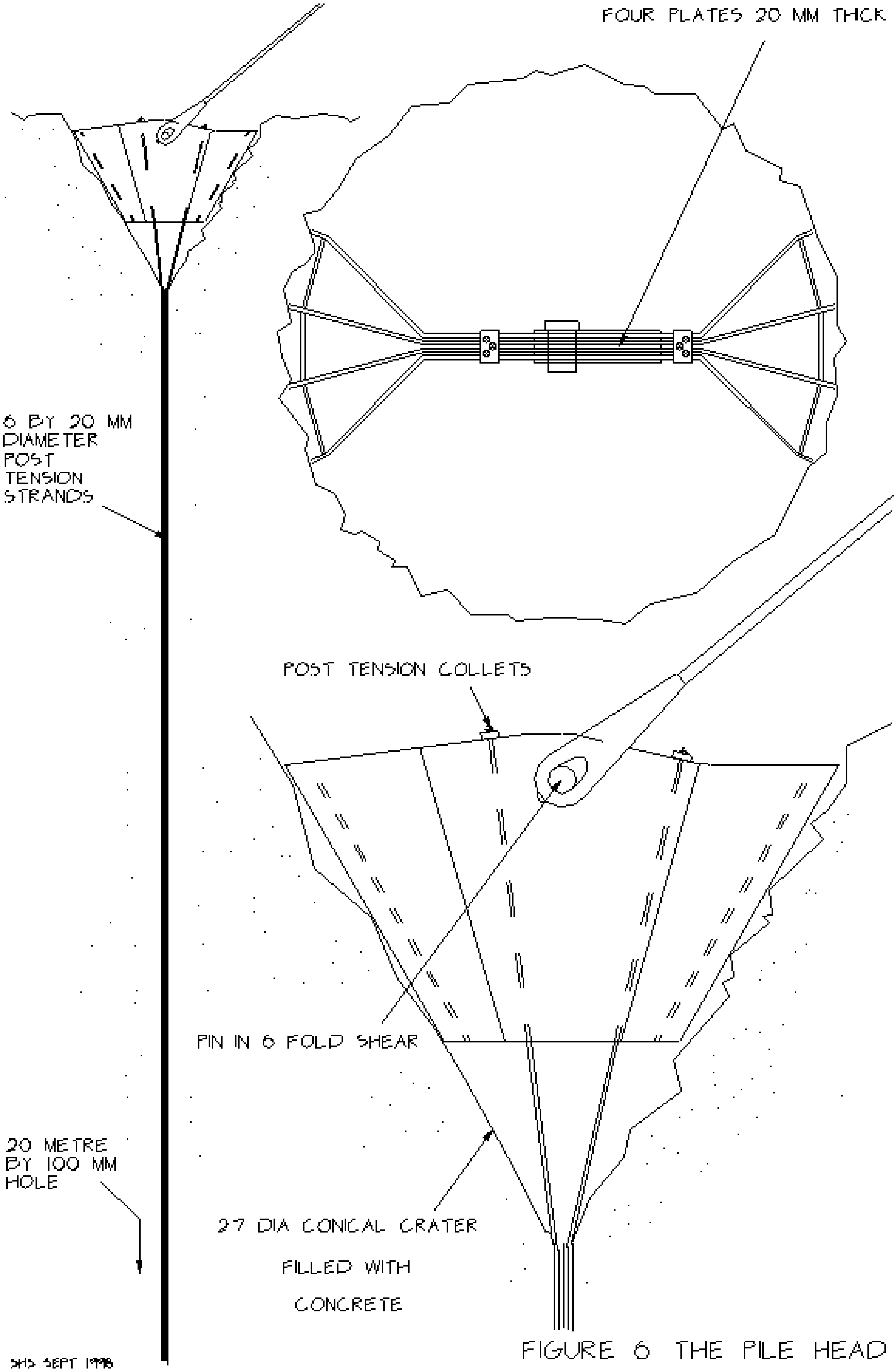


FIGURE 6 THE PILE HEAD

6.0 CHANNEL HYDRODYNAMICS

Clusters of individual tidal stream generators which approach the size of the channels in which they are installed will behave differently and potentially better than those in an open flow field. It gets harder for water to flow around the rotors and so they will behave more like a ducted turbine, unrestricted by the Betz limit. We need to understand the 'output impedance' of the channel and the forcing function of its excitation in order to make the best match with the rotor. We also need to be sure that the higher water level upstream of the rotor does not cause flooding. The rectangular window of the vertical axis configuration allows it to have a higher packing fraction than the horizontal one. The ideal arrangement might be to use two contra-rotating vertical-axis rotors side by side to cancel each other's torque, down-stream vorticity and side thrust.

It is possible that the combined resonance of the channel and the ocean driving it can be enhanced by changing the phase of generation in the manner of the latching and reactive loading being developed for wave energy.

Good tidal stream sites are likely to be in the channels between a large mainland and a smaller island. If there are alternative routes for shipping it may be possible to add a flat upper surface to the power torus and use it as a roadway.

7.0 USEFUL DESIGN FIGURES

The following design parameters have been calculated from a Mathcad worksheet for an initial reference design.

Diameter:	50 m
Maximum stream velocity:	4 m/sec
Maximum design stress:	180 MPa
Tip-speed ratio:	2.67
Rotor tip speed maximum:	10.7 m/sec
Solidity:	0.121
Blade number:	10
Blade span:	3 x 6.67 m
Chord:	1.9 m
Foil section:	Rodewald 0018
Optimum incidence angle:	9 deg.
Pitch change angle:	11 deg.
Theoretical efficiency:	40%
Mooring force:	6.5 MN
Mooring cables:	8 by 64 mm
Torus minor diameter:	3.4 m
Hertzian cam stress:	750 MPa
Roller diameter:	98 mm
Roller width:	100 mm
Lobe number:	307
Follower number:	720
Pump bore:	44 mm
Pump stroke:	65 mm
Pressure:	400 bar
Delivery per rev:	17 m3
Pump strokes per rev:	221,040
Computer decision time:	88 E-6 sec
Maximum torque*:	139 E6 Nm
Total pump weight:	52 tonne
Torque per	£: 273 Nm
Torque per kg pump:	2162
Cost per kg pump:	£7.90
Cost per total kg:	£6.97
Total weight:	640 tonne
Power at 4 m/sec:	12 MW
Harwell parametric cost:	£4.5 m
Cost per peak kW:	£375

** Useful torque would be reduced if some modules are used for bearing duty.*

8.0 ASSEMBLY

Particular care has been taken to make sure that all the parts of the rotor will fit into a sea container so that they can be moved by road or sea to any site. Assembly can take place in a sheltered bay which can be given additional protection from a ship (perhaps one about to be scrapped)moored across the bay entrance. The ship can store sub-assemblies and house the workforce.

The starting point would be to add flotation bags to each the section of the lower ring, float them into place and bolt them together with the shorter bearing-sections between them. Roundness can beset with floating cables running across the diameter. When the ring is sufficiently true the sections can be welded together and a hoop-wire tensioned round the ring to bias the welds into compression. The lower ends of the diagonal ties would be attached to the rings with their upper ends tied to floats.

The floating ring can be rotated to bring each part in turn under lifting gear on the ship or gantries sitting on the sea bed so that the arrangement is effectively a circular production line moving past aligning, welding, painting and inspection points.

The lower hydrostatic bearings will be attached to the bearing sections and the lower set of spars and blades lowered into place. The bearings can have up to 2 degrees of misalignment and this allows the spars to be deliberately splayed outwards from the ring like a crown but retained at the outward tilt by guy ropes tied to points further round the ring. Although 2 degrees does not sound much, a 1.75 metre high wave would be needed to tilt a 50 metre rotor through this angle.

Air would then be released from the ring flotation bags and the first stage lowered until flotation bags at the top of the blades reach the water. The next set of bearings would be added and the next set of ring sections assembled as before. When the upper ends of the diagonal ties are connected we will have a strong and sufficiently stiff structure which can be handled with less delicacy. It can be raised to any height at any time for painting or anti-fouling by means of the flotation bags. These could so useful that it may be worth building them in to the lower ring.

The halves of a section of the power torus can be floated into place inside and outside the upper ring, bolted together with temporary jigs and then joined together with the G-frames which will hold the pumping modules.

Circularity can be set for all 20 sections with the radial floating cables aided by laser measurements from a central datum point. They will then welded together. The power torus will then be bolted to the upper ring of the rotor with removable attachments.

At this point the structure will still have an open roof making it easy to lower parts into the powertorus but not yet giving the clean working conditions desirable for hydraulic assembly.

The cam lobes will be lowered into their correct places which will be defined by rolling bearings with Vee pulleys which locate machined surfaces between the cam humps. Groups of about 20 lobes will be threaded over the internal tensioning wires and tension set on the entire circle. We now have a fairly flexible cam ring running on temporary Vee rollers but with unfinished lobe surfaces.

The cam will be rotated past an optical scanning head which will record the variations in the cam surfaces and so allow the most efficient grinding to be planned. The grinding heads will be assembled in the space later to be used for the generators. The cam will be drawn past them and ground until all the lobe surfaces have been covered. Note that is not necessary for the lobes to have the same height or wave length so that the precision is far less daunting than in making big gears.

At this point the top roof will be welded on to give the torus its full strength. The inside will be washed down with flushing oil passed through progressively finer filters so that the inside becomes clean enough for hydraulic assembly.

9.0 CONCLUSIONS

- The tidal stream resource is not as large as the deep-sea wave resource on the western side of a continent but it is still quite substantial and the output is totally predictable.

- Although the efficiency of turbines in an open tidal stream will be about half that of ones in a barrage the technique avoids the financial hazards of a very large investment and a long construction period. The design can evolve making use of experience from early prototypes rated at a few megawatts.
- The proposed vertical-axis configuration allows convenient entry to an atmospheric generation chamber and the use of zero-pressure gutter seals. It also allows turbines to be packed closely so as to block a large fraction of some channels and so improve on the Betz efficiency limit.
- Cavitation imposes lower tip-speed ratios and so higher solidities than for wind turbines. It may be possible to design hydrofoil shapes such as the Rodewald foil which have a more even pressure distribution with lower negative pressure coefficients.
- The low tip speed enforced by cavitation implies very large torques which would distort the geometry of conventional gearing.
- Low tip-speeds also impose the need for variable pitch if we are to avoid stall at the most useful part of the rotation. This can conveniently be achieved by controlling the pitching-moment rather than the angle of the blade. Blades can be feathered in emergency.
- Ring-cam pumps can provide the very large torques with greatly relaxed tolerances. This increases the maximum possible diameter of the rotor allowing generation at several tens of megawatts and making it more stable in tilt.
- Disabling poppet-valves gives ring-cam pumps a continuously-variable 'gear-ratio' so that a rotor moving at the correct tip-speed for any phase of the tidal cycle can drive a true synchronous generator.
- Energy losses of the ring-cam are dominated by the hydrostatic pads that support the cam rollers. They can be calculated from standard equations for leakage and shear and seem to be less than 1% of output.
- The use of a segmented cruciform cam with force reacted through the thickness gives great economy of pump weight and allows it to act as a geometrically tolerant bearing.
- Buoyancy can remove many of the structural problems imposed by gravity which are suffered by wind turbines. If blades are to be neutrally buoyant they must be quite short with thickness of at least18% of chord.
- Installation problems may be eased if we can use pairs of turbines in reverse in the manner of a Voith-Schneider propeller.
- Maintenance and anti-fouling can be eased if a air bags built into the lower ring can be used to lift the entire structure clear of the water without loss of tilt stability.
- The proposed tension-leg location avoids bending moments associated with tower mountings. Sea-bed attachments at good tidal stream sites will be much harder than in the open sea because of zero visibility and very short diver access time.

10.0 REFERENCES

Salter SH. Power Take-off Systems. Section B3 pp 101-151 DGXII Preliminary Actions in Wave Energy R&D. CEC August 1993.

Rodewald A. Optimisation of a Hydrofoil for use in a Tidal Stream Project. Erasmus studentship project report. Edinburgh University. April 1998.

Salter SH, Rampen WHS. The wedding cake Multi-eccentric Radial Piston Hydraulic Machine with Direct Computer Control of Displacement. pp 47-64. 10th International Conference on Fluid Power. Brugge April 1993. Mechanical Engineering Publications London 1993.