Recent progress on ducks

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Indexing terms: *Waves, Energy conversion*

Abstract: Until the end of 1978, the team at Edinburgh put most of its effort into small-scale laboratory tests with increasing levels of hydrodynamic realism. Some test results show that the requirement for rigidity of the duck backbone is much lower than was predicted, and that in waves with lengths of twenty diameters and more, there are considerable benefits to be derived from controlled backbone movements. Since the beginning of 1979, we have been working on the problems of full-scale design. The CEGB-preferred schemes use low-pressure air turbines, asynchronous generation, d.c. transmission, serial connections and simple designs with reliability achieved by easy access and maintenance. We prefer high-pressure oil hydraulics, synchronous generation, a.c. transmission and parallel connections. We exploit the techniques of modern electronics and plan to achieve reliability by total hermetic sealing of the power conversion mechanism which will exclude all organisms, both marine and human. I shall attempt to justify our approach.

1 Moving-axis results

Several authors (Count,[1] Glendenning,[2] Mei[3] and Standing[4]) have published theoretical efficiency curves for ducks and suggest that movement of the mounting reduces the performance.

It is true that the early ducks were intended to operate on a relatively stable reference frame, and movement was regarded as necessary for survival. But it is not true to suppose that there is a continuous degradation of performance from absolute rigidity to total freedom. The experimental evidence is more complicated, more interesting and more promising than the straight degradation theory would suggest.

Our mounting allows separate control of the rigidity of the model in heave and surge. We made efficiency measurements at a number of frequencies and a number of combinations of rigidity. A typical result is shown in Fig. 1.

The horizontal axis is compliance in the surge direction and the vertical is compliance in the heave direction. The wavelength corresponds to 22 duck diameters and the curves show contours of efficiency. There are two separate regions of good performance, and neither occurs at the point of absolute rigidity. They are separated by a region of extraordinarily low efficiency. We call this area 'death valley'. It is a feature of tests at all frequencies, and may provide a useful operating condition for survival.

At high frequencies, with wavelength, say, eight times the diameter of the duck, then the pattern of the contours moves closer to the point of total rigidity, and at longer waves the contours move towards the more compliant area. This leads to significant improvements in efficiency in very long wave conditions. Fig. 2 shows the effect of varying heave compliance with the surge axis at a constant compliance of $4{\cdot}5 \times 10^{-6}$ m^2/N. The wave period is 12 s.

It is fortunate that ducks prefer rigid mountings for short wavelengths and floppy ones for long ones. Although our knowledge of the angular distribution of wave energy is scanty, it is reasonable to expect that, on average, short wavelength waves will have short crest lengths and so make short intercepts on the backbone and feel that it is rigid. However, longer waves will make longer intercepts and will feel that the backbone is more compliant. It is fortunate, too, that the values of rigidity required are within the range that may be achieved with concrete construction for crest lengths up to about 40 backbone diameters. Narrow-tank tests show that the shapes and positions of the contours of high efficiency can be manipulated over a wide range by subtle choice of duck-shape, ballasting and power take-off. We are learning how to work ducks with acceptable efficiencies on mountings with much greater compliance than that of Fig. 1.

It may be possible to explain the phenomena as follows. In short waves, the water movement is concentrated near the surface and very little energy passes beneath the duck so that it operates in the way expected. But in longer waves, there is movement well below the duck's draught

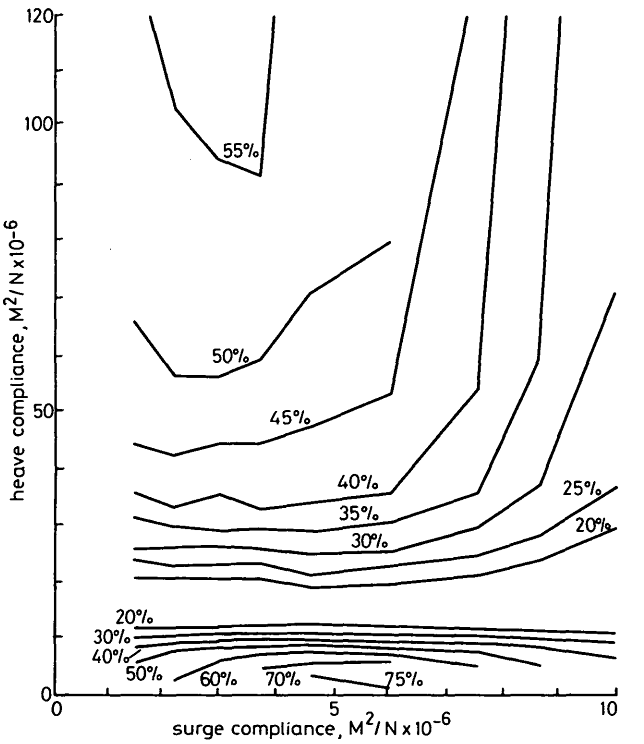

Fig. 1 *Departures from rigidity are desirable particularly in longer waves*

Duck efficiency on compliant axis
Monochromatic sea: $T = 12$ s. $L/d = 22$
Data rescaled to represent 10 m diameter duck

Paper 761A, first received 27th March and in revised form 21st April 1980

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and it becomes more difficult to attract it to the duck instead of to the water to leeward. But if the stern of the duck can be moved in such a way as to generate waves which are the inverse of those propagating below, then the water must be calm. The back of the duck is behaving like an Evans cylinder[5] and, indeed, it was the work of David Evans which led to the discovery of this effect. If the techniques of reactive power take-off and compliant mounting are exploited, it is possible to achieve, with a 10 m diameter duck, the efficiency curve of Fig. 3.

We find that the frequency axis normally used for the presentation of mathematical results is too cramped at the low-frequency end and masks commercially important improvements. In Fig. 3, we have used a period axis but varied the intervals between periods in proportion to the power content of the South Uist wave climate below one metre r.m.s. This corresponds to a reasonable guess at the sensible power limit.

2 The bearing between duck and backbone

2.1 *Forces*

Our test rigs are fitted with strain-gauge sensors, from which are calculated the surge and heave forces acting on the models. A great deal of the tank work has been concerned with the measurements of these forces in both ducks and cylinders.

The established theory for fluid loading was developed by Morison[6] and intended for use on vertical submerged cylinders. Our models are horizontal and pierce the surface. It has not been easy to relate the results to theory. Some unusual effects are caused by the interplay of buoyancy and the vertical inertial force which results in downward forces during both the crest and the trough of the wave. A reconciliation has been achieved by Dixon.[7] We have found that the forces on ducks can be reasonably predicted on the assumption that they are proportional to wave height and duck diameter, the density of water, the acceleration of gravity and a single all-embracing force coefficient measured for the wave condition concerned. The variation of force coefficient with wave length and steepness is not large. A typical value for surge is 0·4 and for heave 0·25 with a tendency to fall in larger waves. The use of a force coefficient brushes aside many of the niceties of fluid-loading theory, but it gets reasonably accurate answers with very little algebra.

If we attempted to hold a duck rigid, the extreme waves in the open Atlantic would raise forces between 150 and 200 tons for every metre of duck width. But the attempt to hold anything rigid against these waves is an exacting engineering exercise. We conducted a series of experiments on the moving rig in which we maintain rigidity up to a limit of force and then allow the rig to be deflected at a constant force. It seems that the economically justified limit for inshore equipment is about 10 t/m. But if the duck is allowed to yield to forces higher than 10 t/m then it is still necessary to provide a force to accelerate the backbone. Until we have data from a model string with the correct values of compliance, we are unsure of the value of this acceleration. The narrow-tank results suggest that it will lie between 0·5 and 1·0 g. As the backbone will weigh 75 t/m, the forces needed to accelerate it are up to one half of those needed to hold it stationary. For a long time we thought that this acceleration requirement raised difficulties in the design of the bearing surface between duck and backbone. Fortunately, the reasoning was fallacious.

The duck's backbone is neutrally buoyant and floats inside it. Its weight is balanced by its buoyancy. The buoyancy force is the backbone volume times the density of water times the acceleration of gravity. Changing the acceleration of gravity would not induce a force between duck and backbone, which would still float as before with backbone weight and buoyancy force in balance. The action of waves produces temporary alterations in the apparent direction and value of gravitational acceleration and the water between backbone and duck produces temporary alterations in the direction and value of the buoyancy forces to maintain the balance as before. The water wants to be left behind just as much as the backbone. It would be foolish to try to keep the space between duck and backbone dry.

The effect may be demonstrated by shaking a jar of pickled onions, but it is also interesting to consider an experiment on an object with excess buoyancy, such as a hydrogen balloon in a car. If the car is accelerated, the balloon moves to the front because the heavier air claims the right to be left behind. When the car is braked, the balloon moves to the back.

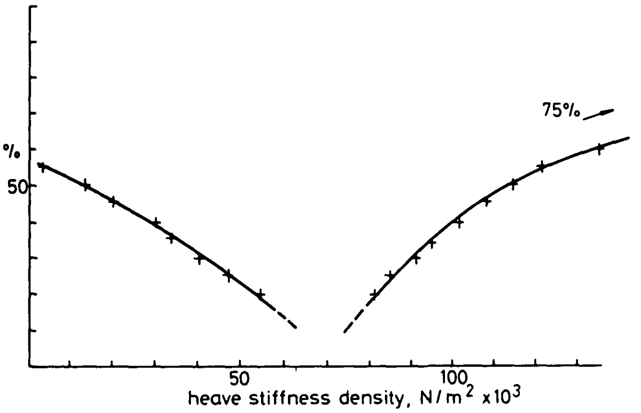

Fig. 2 *Death valley*

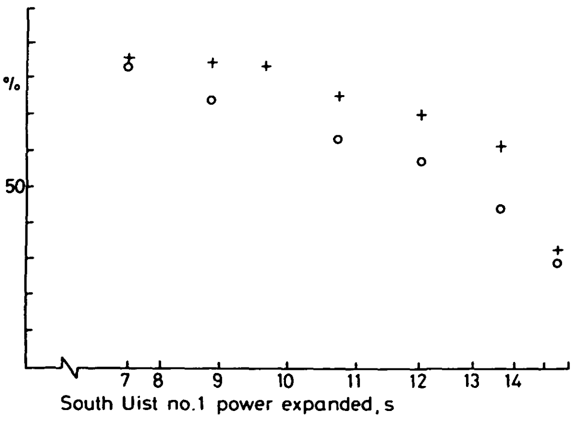

Fig. 3 *Regular and irregular efficiency curves for 1979*

Intervals of time axis are proportional to annual energy at South Uist in sea states below 1 m r.m.s. for each interval of period
+ optimised regular
o p.m. spectrum

2.2 *Bearing requirements*

Although it may be prudent to provide for occasional overload forces of, say, 100 t/m, caused by handling and collision, there is no need for the normal operating forces to exceed about 10 t/m. Despite the fact that loads are not shared evenly over the projected area of a bearing, the value of force over projected area is a convenient parameter for comparison. With 10 t over $9{\cdot}8\,m^2$, we have about $10^4\,N/m^2$ or less than $1{\cdot}5\,lb/in^2$. This is an extraordinarily low value. It is difficult to find examples in bearing technology which are as low. The maximum angular velocity between duck and backbone will be less than 5 m/s (11 m.p.h.), and for most of the time only one tenth of that.

So far, the specification is by no means very demanding but it turns out that the fraction of power lost by the bearing is about double the value of the coefficient of friction. This means that we have to achieve much lower friction than is usually expected of plain bearings. Furthermore, friction and wear are associated. All the economic considerations force the design towards much longer service intervals than would be expected for land-based plant.

Some very successful water-lubricated rubber bearings are made by BTR. They are known as Silverline and are shown in Fig. 4. The shape of the rubber entraps a water film, and so the coefficients of friction and rates of wear are exceptionally low. They are widely used for the stern-tube bearings for ships. Unfortunately, the hydrodynamic effects work only at peripheral speeds which are too high for our application. When the duck motion reverses, the film is lost and the friction rises.

The section moment of a neutrally buoyant tube depends on the fourth power of its diameter. This means that the structural demands of the backbone need every millimeter of diameter. We cannot spare the radial distance needed for a rolling-type bearing, although the dark, wet, slow, cool conditions and absence of braking and cornering forces provide ideal conditions for rubber tyres. Another difficulty is that we cannot expect backbones to be exactly round or parallel. Errors of ± 15 mm would be likely even with the very best civil engineering.

To summarise: we need to design a radially thin bearing for low loads, moderate but reversing velocities, very low friction and wear, which can work in sea water and can tolerate large geometrical errors.

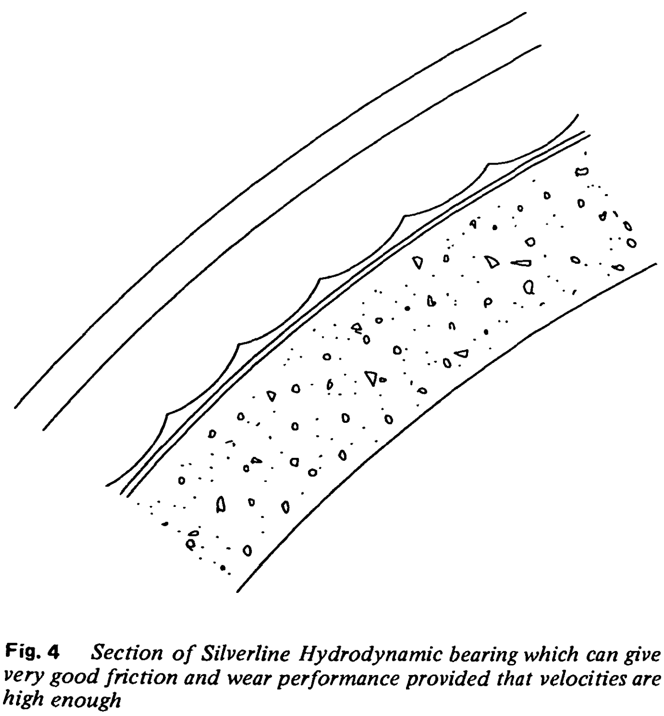

Fig. 4 *Section of Silverline Hydrodynamic bearing which can give very good friction and wear performance provided that velocities are high enough*

2.3 Slubber

Slubber is the general name for an elastomeric body which, when subjected to external pressure, exudes a liquid. One way in which this can be achieved is shown in Fig. 5.

One side of the bearing is made from concrete clad with a thin skin of cupro-nickel, one of the few materials which resists marine fouling. This skin provides a smooth, if inaccurate, surface. Riding on top of the cupro-nickel is a circular pad like a hovercraft. But instead of having an air supply from a turbine, the pad is fed with water from a second chamber with corrugated bellows for walls. If this second chamber has a diameter less than the hovercraft pad, the pressure inside will be greater, and a restrictor between the bellows chamber and the pad will, in combination with the impedance of the gap between the pad and the cupro-nickel, govern the rate of flow of water out of the bellows chamber. The hovercraft will fly over the surface of the cupro-nickel with negligible friction and wear.

An alternative scheme uses a bellows chamber marginally greater in diameter than the pad. This means that a small fraction of the load is taken by the pad lands but the leakage rate is much reduced. This arrangement could be regarded as a friction-coefficient attenuator rather than a true hydrostatic bearing.

Either type will work only for loads which alternate. The bellows chamber must be recharged during the portion of the cycle when the load is removed. But this alternating load is exactly what happens with waves. The choice between the zero-contact and the friction-attenuating type will depend on how well we can make the pad lands conform to the contour of the skin.

Slubber bands will run round the annular space between duck and backbone. Slubber could be mistaken for the skin from the tentacles of a giant octopus suffering from a mutation which causes it to blow rather than suck.

3 Full-scale backbones

The function of a duck's backbone is to provide a reference by connecting it to waves of opposite phases. This is a more difficult task than to go down to inertia in calm water like the oscillating water column or to go backwards like the rafts. But the mechanism used to connect to the reference is itself a useful energy-producing plant, and we judge the attempt to be worth making. It allows us to concentrate arbitrarily large amounts of power and avoids side-to-side collision difficulties.

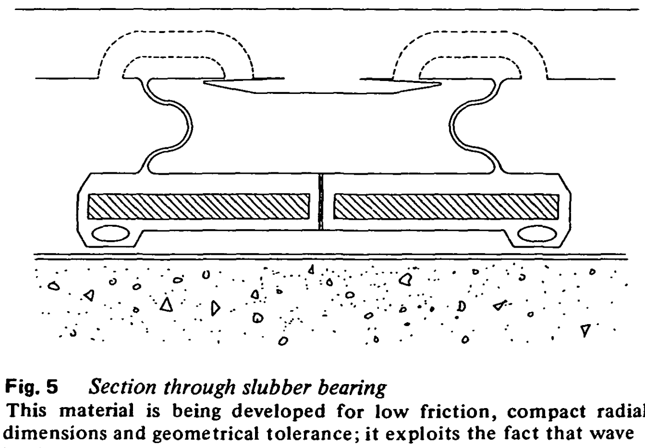

Fig. 5 *Section through slubber bearing*
This material is being developed for low friction, compact radial dimensions and geometrical tolerance; it exploits the fact that wave loads are removed within a few seconds

We realised very early that bending moments and shear forces would dominate the design of the backbone. Whatever rules for fluid loading are used, brute strength offers no solution. Experiments on very long compliant pipes suggested that the bending moments were lower in the central regions of the backbone, and so we are designing for strings several kilometers in length. The only reason for a break in the string is to allow the passage of ships.

It is necessary to let the backbone deflect so as to yield to waves with amplitudes greater than the ducks wish to absorb. Our present design uses pairs of ducks on 60 m lengths of concrete backbone jointed to the neighbours with Hooke's joints.[8] The motion of the joints is controlled by double-acting hydraulic rams arranged around the circumference of the backbone section. The maximum angle is $\pm 12^{\circ}$. Sets of hydraulic rams on opposite sides of the backbone are cross-connected and drive a variable displacement axial piston swash-plate motor/pump of the same type as is used in the power take-off. The hydraulic motors drive synchronous motor/generators running in phase with the rest of the electrical network.

I believe that proper control of the joints will prove an extremely valuable technique. The joints can act as a power-generating mechanism in their own right. They can be used to determine the compliance of the backbone with different values for different frequencies. They can let the backbone yield to extreme waves but restore the midposition afterwards. They can damp out flexure oscillations. It is not stretching imagination too far to propose a sequential sinuous series of torque commands which give sections of duck string their own eel-like mobility without tugs.

If we use the naval architect's rules of thumb for drag estimates,[9] it seems that speeds of one or two knots require remarkably small amounts of drive power. Duck strings can be self-propelled from wave energy while duck-masters sit comfortably at home, watching data relayed from satellites. All these benefits can be realised by the control of the angle of a swash-plate motor. An interesting box of electronics will be needed to work out the angles, but modern electronics is very good at making interesting boxes.

A large area of uncertainty surrounds the choice of joint angle. We know much less about the crest length of waves than about their amplitudes and periods. We know very little about the behaviour of nonlinear beams subjected to wave loading. Testing our new model will keep us fully occupied.

I believe that we will not find problems with the design of the full-scale joint bearings. Glacier DX linings promise excellent pressure/velocity properties and the reversing direction of the loading is a valuable asset allowing continued regreasing. However, it looks as though presently available ram technology will not offer lifetimes comparable with the concrete and power take-off mechanisms. Long-life rams may need further development.

4 Power conversion

4.1 Working fluid

As soon as the wave-power investigator begins to test in the realistic spectra of irregular waves, he is dismayed to discover the very wide range of instantaneous power densities. If we are to make electricity, this spiky flow of dispersed energy must be concentrated, rectified, regulated and transformed to a high-velocity low-torque unidirectional motion. This series of processes is best performed in a fluid and the majority of proposals for wave-energy devices use a fluid interstage.

We can choose to use air, water or oil. Many people in the British wave-energy programme are keen on air systems. They argue that despite the low pressures involved, it is possible to achieve high velocities and even rectification in a single leap. They argue further that the air turbines are 'simple' with only a single moving part. I have been under some compulsion to adapt ducks to use a low-pressure air system. I have resisted for the following reasons:

(*a*) Mechanisms which use air in free exchange with the atmosphere are not using a clean, dry mixture of oxygen and nitrogen. There will be water vapour, rain and spray. There will be crystals of salt and ice. There will be all the flotsam of the sea including weed, beer cans, logs and carelessly low-flying seagulls. We have to avoid obstructions in the air passages, but be very sure that solid chunks of water are kept away from the rapidly moving turbine blades. The open air and natural sea water are ill-defined working media with complicated chemical and biological properties.

(*b*) It is possible to design devices with their own private system of recirculating air from which some of these hazards are excluded. The French air-bag is an excellent example. The air is contained by a flexible skin, but this puts an immense responsibility on to the designer of the fabric. Any failure can result in a sudden loss of buoyancy. Reliable damage control systems, which may need mechanical moving parts, must be provided.

(*c*) The efficient operating band of air turbines so far invented is narrow and the efficiency falls at the low power end when we need it most. It would be possible to improve things by means of variable pitch blades and moving guide vanes, but these would at once lose the advantages of simplicity claimed for the single moving part.

(*d*) It is, in any case, not true that an air turbine has only a single moving part; an apparent simplicity masks a dangerous complexity. Each blade of the turbine can have many modes of vibration, each coupled to one another and the surrounding structure. Low drag demands thin foil sections, whereas rigidity demands fat ones. The blades are moving at high speed in a fluttery medium with very little damping. Changes of speed and air velocity will search out every possible mode of vibration. Curing one will induce another. It is my belief that complexities should be apparent and working conditions clearly defined. We want the chance to exploit clever control, with a working fluid of the most benign properties. That fluid is clean, high-pressure oil.

4.2 Problems for ducks

The problem of converting the motion of a duck into electricity has proved an interesting challenge. The most obvious course would be to build a pumping system driven by the relative motion between duck and backbone. The lack of geometrical precision would make the use of gear-driven pumps difficult, but tapes and toothed belts could stand the necessary tension and allow for generous tolerances. But because of the Archimedes bearing force argument, it would be foolish to keep the space between the duck and backbone dry, and so any pump design would require a seal to keep out sea water. The life of this seal has proved the stumbling block. We could only expect a few years of operation, and this is not acceptable.

4.3 *The power take-off that came in from the wet*

It has proved possible to design a power take-off system which works in completely sealed conditions using a scheme which provides many other advantages. Our main problem has been that physical arguments and mathematics need to be augmented by the feel of a working model to convince critics of the viability of our proposals. The scheme exploits the behaviour of the gyro. I am indebted to Professor E.R. Laithwaite for the suggestion. Thinking about gyros is difficult for two reasons. The first is that we need to imagine torques and angular velocities which are harder to visualise than their linear counterparts. The second is that we have to think in three dimensions.

The axis of spin of a gyro defines one direction. We can consider the two other axes perpendicular to the spin axis as the 'ports' of a transforming device which converts torque into angular velocity. The gyro does not know which port we are labelling as an input, and the exchange of torque for angular velocity works both ways. Input torques produce output angular velocities and output torques require input angular velocities. If the port we choose to label an output is moving without opposition, then there will be torque but no angular velocity at the input. Now a device that has a torque in one place and an angular velocity at another could also be described as a gear box with an infinite ratio of speed increase. Speed-increasing gear boxes, especially efficient ones with high ratios, are very difficult things to make and are just what wave energy needs.

Our gyros are fitted in a canister in the beak of the duck. Its diameter can be half the backbone diameter. We use pairs spinning in opposite directions about axes initially perpendicular to the duck's axis of nod. When the waves apply a torque about the duck's axis of nod, the gyros will precess about the third axis perpendicular to both the axes of spin and nod. This precession movement can be used to do work if it is opposed by a torque. It is necessary for there to be a torque on the output if the duck is to move, and this output torque will determine the angular velocity of nod. Because a pair of gyros spin in opposite directions, they will also precess in opposite directions and so the two output torques will be in opposite directions and can be cancelled by stresses in the duck without bothering the backbone.

The 'gyroness' of a gyro is determined by the two parameters $I\omega$ which remain together in all the equations. I is the moment of inertia of the disc and ω is its angular velocity. If ϕ is the angle to which the gyro has precessed, the torques and velocities are related as follows:

$$T_{out} = I\omega\,\Omega_{in}\cos\phi$$

$$T_{in} = I\omega\,\Omega_{out}\cos\phi$$

Therefore,

$$\frac{T_{in}}{\Omega_{in}} = I^2\omega^2\cos^2\phi\,\frac{\Omega_{out}}{T_{out}}$$

But T_{in}/Ω_{in} is the hydrodynamic damping coefficient of the duck. Let us call it κ. We must get it about right if the duck is to behave correctly in the water.

Everything is inverted when seen through a gyro. Low damping coefficients on the output look like high damping coefficients on the input. Captivity looks like freedom. Similarly, spring and inertia are interchanged, with big inertias looking like low rate springs. Death valley looks like the garden of peace.

The cos ϕ term reduces the usefulness of the gyro by less than one at first expects. We require the greatest amount of torque and power at the central position of the system. It would be sensible to arrange that the biggest waves in a sea state drove the gyros through nearly the maximum angle of 180°. We can arrange this by altering the speed of gyro spin. This will set a number r which is rather like a gear ratio.

$$r = \frac{\Omega_{out}}{\Omega_{in}} = \frac{T_{in}}{T_{out}}$$

and as

$$I^2\omega^2\cos^2\phi = \frac{T_{in}}{\Omega_{in}}\,\frac{T_{out}}{\Omega_{out}} = \frac{\kappa^2}{r^2}$$

therefore,

$$I\omega = \frac{\kappa}{r\cos\phi}$$

As the angle of precession increases, the value of r rises.

If we reduce the speed of gyro rotation we increase the value of the gear ratio. In calm conditions, we expect to run at a midpoint gear ratio of about six, and near power limit at a ratio of about three. But we are still at an early stage in the optimising process and many factors influence the decision. I believe that wave-energy devices with larger damping coefficients could make use of even higher gear ratios. (We have carried out tank tests with ratios of up to twenty.)

4.4 *From precession into oil*

The energy in the precession movement of the gyro is turned into oil pressure by a modified version of the ring cam pump. These units are made by MacTaggart-Scott, Poclain and Häggelund. Fig. 6 shows the shape of a conventional ring cam with inward-facing lobes driving rollers in radial directions.

The cost of such a unit depends on the sum of the cost of the lobes and the followers, and so ought to rise with

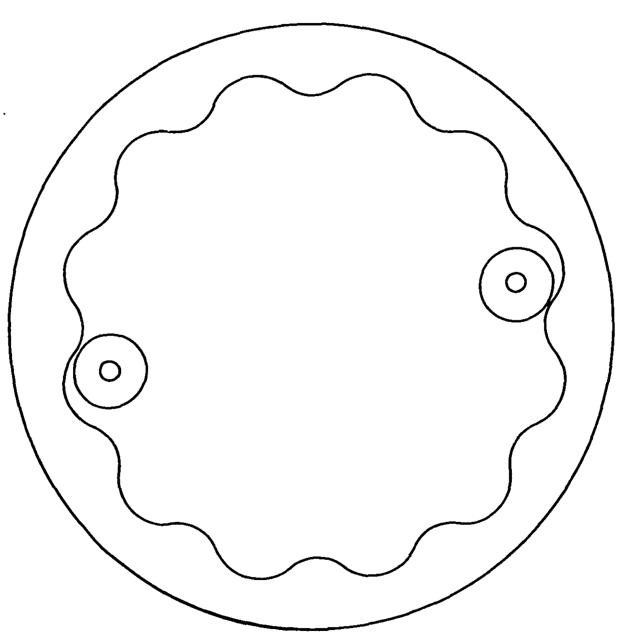

Fig. 6 *Conventional ring cam*

diameter. However, the volume pumped depends on the product of the number of lobes and cams and so rises with the square of diameter. We should, therefore, go toward the largest diameter that can fit inside the power canister. But as the diameter rises, the force from each roller has to travel further round the rim before it can be balanced by another force on the opposite side. For very large diameters, this becomes less attractive and so we are using pairs of rollers working against each other through the thickness of the ring, as in Fig. 7.

To make best use of the space inside, we twist the ring so that the rollers no longer move in its plane. It is now more like a belt than a shape cut from plate (see Fig. 8).

If we think of the gyro assembly as occupying a sphere like the earth with its precession bearings at the north and south poles, then the largest possible ring cam would be the equator. But the equator is a busy place. We have to crowd

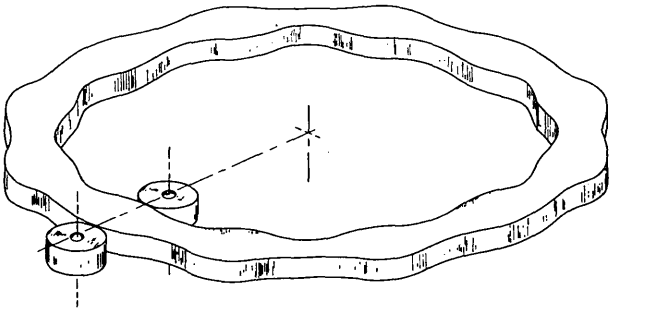

Fig. 7 *Enlarged ring cam with duplex rollers which avoid the need for forces to be balanced across diameter*

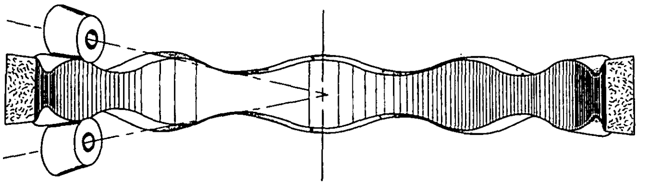

Fig. 8 *Twisted belt form of ring cam with conical rollers which make better use of space inside power canister*

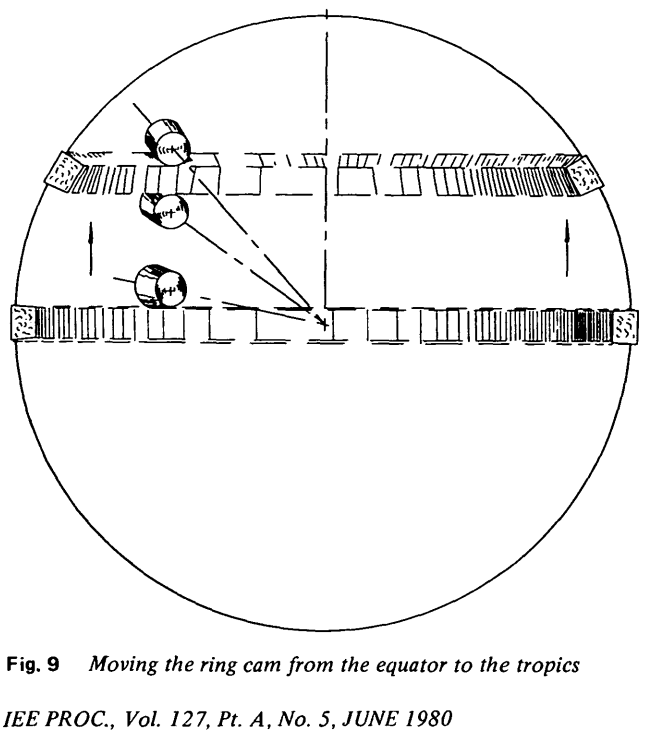

Fig. 9 *Moving the ring cam from the equator to the tropics*

it with the high-speed gyro bearings and their motors. So, in Fig. 9, we have moved the ring cams to the tropics, and have the chance to fit two to each gyro.

At first, we were daunted by the prospect of machining cam lobes on a conical surface, but it turns out that the specialised machine tool required will be cheaper than that needed for a flat plate cam of comparable diameter. It can be built with rotating bearings only and needs no slideways.

The forces on the ring cams are largely compressive, so that we can make them from flame-hardened cast iron. The line of rolling between cam and follower is the only place in the entire system where moving metal touches metal with a force across the contact. It is the only place in the power chain with a finite life. Fortunately, the mechanics and metallurgy of rolling contacts have been the subject of intensive study by the rolling bearing industry. We know how to calculate the fraction of a population which will survive particular loads and can make this fraction acceptably high over the design life of 25 years. Some rollers or cam lobes will fail and we shall need to provide for their early retirement.

Each roller must be coupled to a piston. This is done by a link shown in Fig. 10 which gives yet another speed increase and provides restraint against unwanted roller movements. It is very important that the force should be evenly distributed along the line of contact between the roller and the cam. Imperfections in the geometry are accommodated by a spherical bearing at the pivot point of the link and a ball-ended connecting rod between link and piston. The rollers are kept in line on the cam by small flanges. These dominate, and everything else must be free to comply. The ratio of roller circumference to cam lobe wavelength is chosen to spread the wear patches evenly round the roller circumference.

4.5 Cam profile

The sinusoid is not the best form for the profile of our cam lobes. It has sharp curvature at the crest which leads to higher compressive stresses. It has high accelerations at the crest which must on no account overcome the boost pressure and let the following rollers lift. Although the profiles chosen for most conventional motors are designed to give constancy of torque with acceptable life, we have so many lobes and followers that smoothness of output is not a problem. Instead, we shall design for maximum life with a sharp concave curvature at the trough (nearly the same as the roller diameter) and a blunt curvature at the crest.

It may prove desirable to use a form with some asymmetry about the vertical to the cam surface. This is because the rollers on trailing links have a slightly easier time than those on leading links and we are trying to spread the duty as evenly as possible. The rollers are thick-walled cylinders and run on hydrostatic bearings which support them evenly behind their contact area. Lubrication for the

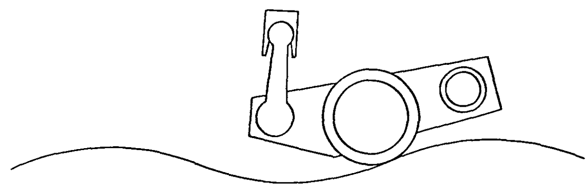

Fig. 10 *Roller follower and link which drives low speed pump*

There are 384 units in each duck

roller and pivot bearing is supplied from the working cylinder via a hole through the middle of the connecting rod. This means that the pressures are always appropriate whether the cylinder is operating or idling.

It is important that the torque which resists precession should be reflected through the gyro to give the duck the correct behaviour. This precession torque is set by the number of pistons allowed to pump. Each cylinder has an electronically controlled poppet valve which is held open by the field of a permanent magnet. Boost pressure in the low pressure manifold drives oil into the cylinder during the down-stroke of the piston. If the inlet valve remains open, then the up-stroke of the piston will return this oil to the low-pressure manifold and no work will be done. But if an electrical pulse is applied to the coil controlling the inlet valve then the cylinder is enabled, the inlet valve is allowed to close and the oil raised to a high pressure (20×10^6 N/m^2). It passes through the outlet poppet into the high-pressure manifold. If this pressure is constant (and I shall show later why it is constant) the torque will depend on the number of valves enabled. The total number is large enough for the gradations of torque to be essentially smooth.

The decision about which valves are to work is made by a computer to which is fed information about the duck's angle, velocity, pressure field, recent history of power levels, gyro disc speed, gymbal frame angle, condition of the mooring system, deflection of the backbone, demand on the grid, Rotterdam spot prices, yellowcake futures, news of unrest in Namibia, and whatever other pieces of hydrodynamic or economic data may turn out to be relevant in the future.

An interesting example of the way in which microprocessor technology can revolutionise mechanical design is the use of the computing system to nurse weak points on the ring cam pump. Rolling contacts have a finite probability of failure. The earliest sign will be a change in the noise made by the rollers. A number of piezo-electric transducers around the ring cam will sense the time of arrival of fault noises and so pinpoint their source. If the blemish is on a roller, then that roller can be avoided in the future. If it is on the ring cam, then rollers will be offloaded as they are due to pass over it. The maximum torque of the ring cam pump will decline with age, but it can start with a generous margin of spare capacity.

4.6 *Keeping the pressure constant*

The hydraulic circuit is shown in Fig. 11. The oil from the ring cams will drive a swash-plate motor at each end of a gyro shaft in parallel with another driving an electrical generator. If the pressure tends to rise as a result of a burst of wave energy, the angular deflection of the swash-plates will increase so as to allow the extra energy to speed up the gyro flywheel. If the flow of energy from the ring cam is at exactly its mean value the gyro swash-plate motors will move to their zero displacement angle, and all the oil will flow to the motors driving the electrical generator. If, during a lull, there is less oil from the ring cam pump than is necessary for the generator drive, then the swash plates on the gyro motors will swing over so that they pump and draw the energy deficit from the gyro disc. The gyro-drive swash plates will move at about twice wave frequency.

The moment of inertia of the flywheel is such that only very small speed variations are necessary to stabilise the generator output. At the nominal speed, each duck stores about half a megawatt hour. (This may be doubled if necessary).

The swash plate of the motor driving the generator is controlled so as to keep the phase of the generator correct. The phase angle determines whether it motors or generates. The inertia of the swash plate is so small and the forces available to control its angle are so large that its frequency response is very high. The generators can respond to power variations in a few milliseconds. The reserves of energy instantly available from the flywheels make for better control characteristics than any land-based system, whether steam or hydro. The entire duck string consitutes a spinning reserve capable of stabilising the grid rather than causing it problems. Whenever a pumped storage scheme is proposed, the generating boards emphasise the value of spinning reserve which is said to be worth hundreds of millions of pounds a year, even if never used. The ducks are claiming those millions for being a short-term but instantaneously responsive storage scheme. They will leave pumped storage to do its proper job of overnight working.

The vast amount of flywheel storage means that every piece of electrical equipment from the shaft of the generator to the land connector can now be rated at its mean rather than its peak rating. We can deliver some of the electricity when the consumer wants it rather than when the waves provide it. There is an enormous difference between the value of a peak and a base kilowatt hour.

4.7 *Swash-plate motors*

The scheme described in Section 4.6 demands exemplary performance from the swash-plate motors. The design selected for ducks was developed by Clerk for flywheel energy storage applications. It is well described in Reference 10 but its design is so remarkable that some of its features should be mentioned in this paper.

Most hydraulic pumps and motors run with their casings full of oil. Although this serves to lubricate any areas overlooked by the designer, churning losses limit the speed to about 2000 rev/min. The Clerk motor runs with its casing evacuated. This has several advantages. Churning is avoided. Vacuum-stripped oil gives fewer cavitation problems. There is no degradation of the oil from oxidation. There is no problem about sealing the drive against the vacuum of the duck flywheel enclosure. Several designs of swash-plate motors allow side forces from the plate to spoil

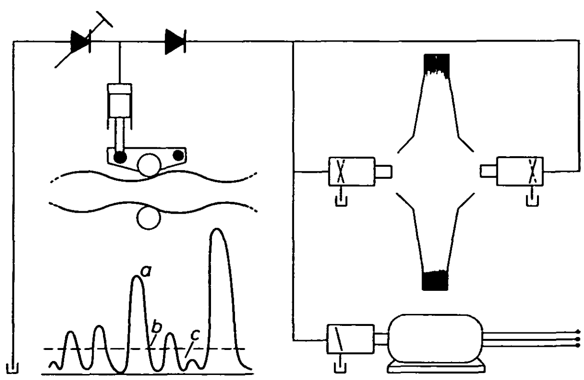

Fig. 11 *Hydraulic circuit and short section of power record*

At point *a*, power is being fed into the gyros. At point *b*, the incoming power is exactly right and the swash plates of the gyro drive motors go to their middle position. At point *c*, energy is flowing out of the gyros to maintain the output steady during the lull the contact between cylinder and piston. The Clerk design provides ball-ended connecting rods so that no side loads develop.

Most designs have cylinders which increase in diameter under applied pressure. Clerk has sealed his cylinder liners to their block at the outermost end and applies pressure to the outer as well as the inner wall. The result is compression instead of tension, resulting in less fatigue and the chance to use ceramic liners. There is reduced rather than increased clearance so that leakage is lower in the power stroke and viscous losses are lower on the return.

Swash-plate motors induce side loads into their shafts which upset the alignment of the main bearings. Clerk uses spherical bearings. Every loaded moving surface runs on a hydrostatic pad. At 1500 rev/min and 20×10^6 N/m^2, a unit weighing one ton develops 1·125 MW.

5 Gyro discs

5.1 *Materials*

The requirements for energy storage differ from those of precession. The energy people want $I\omega^2$ while the gyro applications need only $I\omega$. This means that, instead of materials with high ratios of strength to density, such as the glass, kevlar or carbon-fibre composites, we require a material which has a high product of strength and density. If we want to find the best value for strength times density divided by cost, then it is extremely difficult to beat steel. The ordinary mild steels and the low to medium carbon steels with their properties improved by cold working are strong, heavy and cheap. Furthermore, engineers have been using them for so long that we can be confident about their behaviour.

5.2 *Disc shape*

The newcomer to flywheel technology learns very soon to conceal his surprise that the biggest stresses are at the hub of the flywheel and that even the smallest hole there has a devastating effect on all the stress calculations. Super-efficient energy-storing flywheels have obese hubs and knife edge rims and profiles which are cunningly calculated curves.

Again, the gyro requirement is different. The most efficient design from the $I\omega$ point of view is a ring wound from piano wire. This offers a very high tensile strength-to-cost ratio (1400 MN/m^2 and £600 per ton, 1979). But although it makes an excellent gyro, it is not easy to connect it to the spindle through which large torque must pass and about which large moments must be applied. Solid forgings have been used for flywheels and give good material properties. They are excellent for small components but, as the weight of the part rises, forging becomes more and more expensive.

The disc design which will be used for duck power take-off is that invented by Robert Clerk.[11] It is not the lightest possible, because the gyration forces are so much larger than the gravitational ones that weight is of little consequence. Its shape is chosen from the point of view of manufacturing convenience rather than mathematical perfection. It uses very nearly the cheapest steel available (£222 per ton, 1979). It will be operated at speeds which keep the stresses well below the danger level. It needs no machining, balancing or even painting. We must forget the image of fantastic precision achieved by navigation gyros and think about plain lumps of steel spinning at a moderate speed.

The disc is assembled from a stack of laminations as shown in Fig. 12. Each lamination is pressed into a double cone shape. The lamination thickness is 8 mm, a compromise between the costs of rolling thin sheets and those of pressing the conical shape into thick ones. The ring nuts which clamp the laminations to the gyro spindle induce in them large precompression forces. These are reduced by the centrifugal forces so that we end up at the working speeds with an acceptable level of tensile stress. Our present design for a duck of 10 m diameter and 24 m width uses four gyros of 17 tonnes each. The total weight of gyro disc is comparable to the weight of pumps saved by the increase of angular velocity.

Applications of the gyro at sea are not new. The first proposal known to me is from Sir Henry Bessemer, better known for his metallurgy. Sir Henry wished to improve the comfort of cross-Channel steamers but he was not well advised. He believed that gyroscopic effects only worked over small angles. He planned to use them as roll sensors which would control large rams to stabilise the first class saloon suspended in a rotating gymbal. It is an interesting insight of the class attitudes of the day that this design would have induced larger rolling amplitudes for the second-class passengers and crew.

Much sounder grasp of gyro technology is shown by Otto Schlick[12] whose work resulted in the most effective stabilisation of naval vessels. An MTB displacing 56 tons was stabilised by a gyro disc of 500 kg. Gyro stabilisation was used successfully for over forty ships, the largest being Conte di Savoia of 41 000 tons which used three gyros of 100 tons weight each.[13]

Let me summarise what I see as the advantages of the gyrated power take-off approach:

(*a*) There is a substantial increase of angular velocity and corresponding reduction in torque.

(*b*) No torque is developed in the backbone.

(*c*) Oil can be centrifuged to an extraordinarily clean state with no pressure drop.

(*d*) Enough energy storage is provided to run all the later stages at the mean rating rather than at the peak and to let the generators run synchronously.

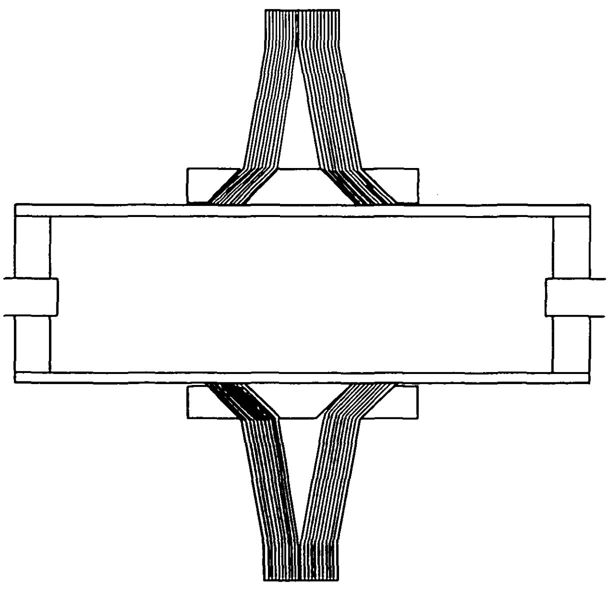

Fig. 12 *Gyro disc is attached to spindle by inward pulling cones which prestress hub*

(*e*) Most important of all, we have got away from the indefinite dangers of sea water and its biology into the best working conditions that the mechanical engineer has ever enjoyed.

6 Reliability, simplicity and maintenance

The Future Energy Concepts Conference,[14] organised by the IEE in January 1979, contained several papers on wave energy. Some of them contained statements with which I find it difficult to agree. For example, Glendenning writes (p.112): *'A reliable system is one which is simple, contains few, robust components and is readily maintained.'* and Bellamy writes (p.168): *'The lesson appears to be that the engineering design of wave power devices should provide for large factors of safety with simplicity as the over-riding design principle.'* There are no recognised units for the measurement of simplicity, and the various national standards institutions have not suggested approved levels for the engineering profession. But we can turn to 'The Oxford English Dictionary' and find twelve columns devoted to the concept. They begin:

'Free from duplicity, dissimulation or guile, innocent and harmless, undesigning, honest, open, straightforward... Free from, devoid of, pride, ostentation or display... Humble, unpretentious... Free from elaboration or artificiality, artless, unaffected, plain, unadorned... Free from over-refinement, unsophisticated, unspoilt... Poor or humble in condition, of low rank or position, undistinguished in office or rank... Not marked by any elegance or grandeur, very plain or homely... Small, insignificant, slight, of little account or value, also weak or feeble... Low, poor, wretched, pitiful, dismal... Deficient in knowledge or learning, characterised by a certain lack of acuteness or quick apprehension... Lacking in ordinary sense or intelligence, more or less foolish, silly or stupid, mentally deficient, halfwitted... With nothing added, mere, pure, bare, single... unlearned, ignorant, easily misled, unsuspecting...

The definitions continue with simple heads, simplistic, and so on.

It seems to me that 'simple' is not a simple word. Although I do not argue that simplicity is for simpletons, I believe that it is an irrelevant factor. I want to get things right whether rightness comes from simplicity or complexity. The history of technology has many examples of designs which were 'right'. Very often, these 'right' designs are *elegant.*

The simplest car engine design uses side valves. The most complicated ones use overhead cam shafts. How many side valve engined cars are sold today?

Glendenning[14] wants the number of components in a system to be small but does not discuss the manner in which they are arranged. I would argue that the distinction between serial and parallel arrangements is of some importance. With parallel connections, the overall reliability of the system *increases* with number. The human brain, after the age of about thirty, loses one of its 10^{10} cells per second and can survive major surgical interference with effects that are hard to detect. I conclude that it is the manner of interconnection of components and their individual suitability which determine success.

The importance of ease of access for maintenance is also frequently urged. For land-based plant, this is an important feature. It is easier to design things with a short life than for a long one, and often more profitable to replace components than to make them last. But if we take the practices of land-based plant and try to apply them at sea we have to expect that the maintenance people will have problems getting to work. The more favourable the wave climate, the harder they will find it. When I discovered the costs of an offshore man-hour, they seemed so high as to be effectively infinite. When I tried to imagine the quality of work carried out by a man who is cold, sick and frightened, it seemed better to manage without it. Our design group has decided that it should be against our religion to assume that any maintenance will be possible. This concentrates the mind and we have tried to apply this principle to every stage of the design. I believe that we can succeed at least as far as the power conversion systems are concerned. Failures will occur and the power canisters must be recovered for inspection so that modifications can be made for future units. The surviving fraction of the population must generate enough electricity to pay for the dropouts.

Bellamy[14] wants large factors of safety. But factors of safety are really factors of ignorance and lead directly to factors of waste. Although there must be some band of uncertainty about the magnitude of a force or the strength of a component, there is no reason for this band to be very wide. We have to balance the cost of reducing the band against the cost of extra, unnecessary material or the costs of failure.

It is also important that we design for the best level of failure. In the field of wave energy, this need not necessarily be zero. After thousands of years of evolution, the naval architects, ship owners and underwriters have settled on practices which result in between one and two per cent of all ships being lost each year.* They have had enough time to get it right. Their factor of safety is less than one.

My own belief is that the correct level of reliability is achieved if:

(i) The engineer has an accurate understanding of the working conditions for which he has to design.

(ii) The engineer has an accurate understanding of the behaviour of the materials and components at his disposal.

7 Getting the power from the duck to the backbone

The precessing gyros and the ring cam pumps work into a constant pressure manifold and produce a flow which follows the spiky energy flux of the random sea. The pressure is regulated by the variable displacement axial piston motors which pass energy into or out of the gyro flywheels. We have the options of sending power to the backbone as a constant pressure, constant flow of hydraulic oil or as electricity. The oil route would allow the possibility of economically large electrical generators, but we reluctantly rejected this solution in favour of the on-duck generator for the following reasons:

(*a*) For comparable weight of metal, the flow of hydraulic oil in a pipe is less efficient than the flow of electrical current in a wire.

(*b*) The flexibility of electric cable is much greater than that of hydraulic hose.

(*c*) We are anxious to prevent cross-contamination between hydraulic units.

(*d*) Reciprocating axial piston motors are more efficient and more easily controlled than turbines but have not been developed for the sizes necessary for exploitation of the big

*M.J. Tucker, Personal communication

generator option. The on-duck generator can use exactly the same units as are used for the gyro drive and joints.

The generator voltage is a compromise between robust windings and efficient transmission over a distance. The choice of 3·3 kV fits in well with the rule of a kilovolt per mile and a standard range of generator. This sets the current rating of the conductor at a little over 200A.

A quick estimate of the flow velocities round the duck in extreme conditions (20 m/s) reduces the attraction of loose flying leads. We insist that everything should be inside the duck outline Although typical duck angles of nod are 30° to 60° the cable run must allow for angular movements of 270°.

The outline drawing of a duck shows that there is room for a spool of about one quarter of a duck diameter in addition to the main power take-off canister. We propose to wrap several turns of a flat multistrand belt cable round the spool under and then round to the top of the backbone, as shown in Fig. 13. If the ratio of diameters is one to four, then the spool will need at least three layers of cable.

When a belt is wrapped around a drum, the strain is the ratio of belt thickness to drum diameter. A solid conductor 2·5 mm in diameter would only suffer one thousand microstrain and a seven-strand core only 330 microstrain. These strains would allow infinite life for copper or even aluminium conductors. However, our suppliers, W.L. Gore and Associates, prefer to use conductors wound from 37/0·4 mm for which the strain will be only 160×10^{-6}. They propose an insulation thickness of 1·15 mm of CR teflon followed by 0·4 mm of polyurethane for abrasion resistance. Eighteen of these strands can lie in a pressurised oil cavity inside 2·5 mm of polyurethane. Polyurethane-clad steel strands can be included for strength. An outside layer of polyurethane felt may be added for antifouling protection. Fibre optics may also be included to carry data to and from the duck computers.

The CR teflon insulation is extraordinarily effective for its thickness. In air, this cable would be rated for 14 kV a.c. and 31·5 kV d.c., with a 30°C rise at 37·5 A per core. The belt thickness is 8·64 mm, so that the strain in the outer layer is only 0·35% This value is extremely small for elastomers. The rotating spool must be connected to the stationary power canister without slip rings or rotating shaft seals. We can twist it evenly along a 20 m length parallel to the power canister. The cable will need a sealed entry to power canister and backbone. Neither need be mated wet.

When the duck is moving upwards, the torque necessary to accelerate the spool can be provided by belt tension. But it is necessary for the belt to be recoiled on the spool on the return stroke. If ever the torque needed to overcome the spool friction and inertia failed, then there would be an ugly tangle of live 3·3 kV conductor.

Model tests suggest that the maximum angular acceleration of the duck is 0·9 rad/s^2. With a 3:1 ratio between duck and spool, the spool acceleration will be 2·7 rad/s^2. A provisional estimate for the amount of inertia of the spool plus the conductor wrapped on it is 450 kg m^2. This means that we need a torque of about 1200 Nm.

The life time requirement is about 10^8 operations, which makes it difficult to use mechanisms like the torsion bar or the tensator, even though we could make them serve double duty as electrical conductors. The cheapest and most reliable mechanism we can devise is a second belt wrapped round the conductor spool as in Fig. 14. The extensions of the second belt are small because the amount of conductor wound off the spool will be very nearly the same as the amount of tension belt wound on. The only source of difference is the thickness of the layers. Nevertheless, we should choose a material giving a reasonably low spring rate with a steady tension of about 1000 N. It should have a thin section, so that the bending fatigue is kept low.

This tensioned spool design offers the advantages that:

(i) The dangerous path between the safety of the power canister and the duck backbone is protected by the hard duck skin.

(ii) There are no seals or slip rings.

(iii) The amounts of strain are very small and accurately defined.

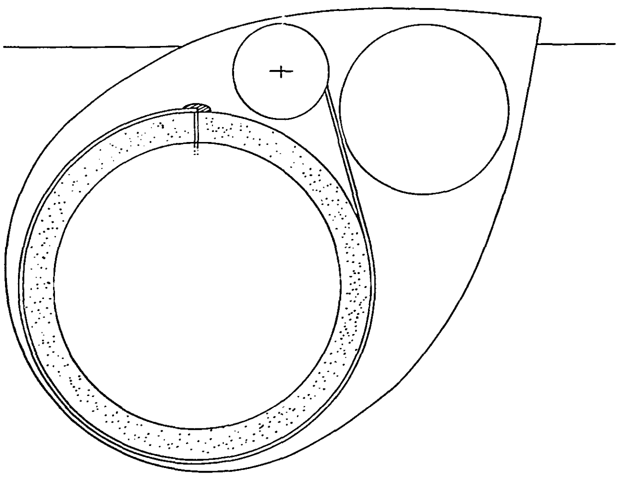

Fig. 13 *Cable spool can be about one quarter of duck diameter*

Although the typical movements are about 60°, we are allowing for a maximum of 270°

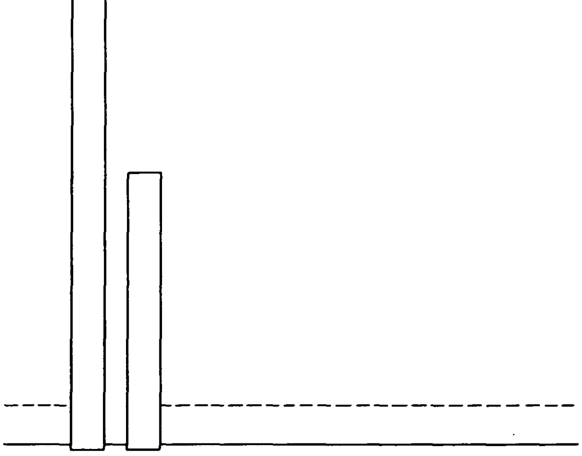

Fig. 14 *If full width of duck is used, twisting of ribbon cable will not induce excessive strain*

8 Moorings

8.1 Forces

Every wave-energy designer must be familiar with the key equation of mooring developed by Longuet-Higgins and Stewart.[15] This holds for any device. They showed that, quite apart from the effects of wind or currents, there is a unidirectional leeward force F determined by the amplitudes A of the incident, reflected and transmitted waves:

$$F = \frac{1}{4} \rho g \left[A^2_{incident} + A^2_{reflected} - A^2_{transmitted} \right]$$

If we are to minimise the mooring force, then it is clear that we must reduce reflections and begin to transmit as soon as the power limit has been reached.

With small to moderate amplitudes, we find that the mean horizontal force on a duck is about 13 000 N/m width for a 10 m duck. But we do not find that the mean horizontal force rises with larger wave amplitudes. Indeed, the tendency is for the values to fall and some negative values have been measured in extreme irregular waves. The movement of the duck mounting and the waves created to leeward by large duck angular movements are dumping energy into the water astern. This is the very best place for it to go.

The negative values of mooring force are caused by the nonlinear behaviour of the waves going over the top of the duck. They are broken up into components with higher frequencies and, as the energy is the same and the period less, the transmitted amplitudes must be higher. This effect is particularly marked with plain circular cylinder models floating nearly awash which can sometimes migrate towards the wavemaker. Longuet-Higgins gives a full account in Reference 16.

8.2 Practical moorings

As duck strings present a small profile to long shore currents and the back-bone avoids problems with side-to-side location, our main concern is to resist thrust from the wave direction towards the beach. We are advised by British Ropes that variation of tension is much more damaging to a rope than a steady large value of tension. Everything must be designed to keep the tension in the cables constant rather than low. But the safety of the backbone and duck bearings demands that the duck string must be allowed to yield to any wave with energy more than can be absorbed. If we want lots of movement with the least variation of tension then we must go for a low spring rate.

But low spring rates combined with the inertia of the device will make the natural period of surge movement long, probably several minutes at full scale. This movement can be excited by the envelope of wave groups and, although the exciting forces are low, it is unfortunately the case that the damping at these long periods is also very small. The models go back to the end of their tethers and then come surging forwards towards the waves. The lower we make the spring rate, the further they move, causing great difficulties with electrical connections.

The solution of the problem is to find a way of increasing the value of damping. There are many ways in which the duck string can move. One is surging as a single, unbending body. Another is yawing. The others are separate movements of different sections forming the various modes of vibration. These latter can be damped by the rams in the joints. But another method can be used which will also damp movements of the string as a whole. Damping is force opposing velocity. If we can sense forward velocities of the string and then modify the duck's power take-off so as to increase the amount of reflected energy then the mean horizontal force will be temporarily increased. When the forward velocity ceases, we revert to the previous low reflection condition. I hope to show that quite small amounts of damping applied in this way will prevent the growth of long period oscillations. Doppler sonar methods will give cheap, reliable velocity indication.

If we design a system of sinking weights and rising buoys which gives a spring rate of 100 N/m^2 for each metre of duck width and then assume that the ducks follow every wave of the South Uist climate, the mean tension (90 t per duck pair) and the variations on the mean are of no concern. British Ropes suggest that we can expect a life of 25 years. They warn, however, that the wires must not be allowed to corrode or bend. Cladding a grease-impregnated cable with a plastic sheath will delay corrosion and we can afford exotic metals to prevent it. The difficult problem is the design of a shackle termination to protect the wires from bending.

To summarise:

(*a*) Never reflect unless you are trying to damp.
(*b*) Transmit when you have absorbed your power limit.
(*c*) Use a low-rate mooring.
(*d*) Find a way of damping the long period oscillations.
(*e*) Beware of the terminations.

9 Transmission

Inventors of wave-energy devices concentrate their efforts on the first stages of energy conversion and have a tendency to assume that the later details of connection to the network are somebody else's problem. Up to 1978, our main effort went into the testing of small scale models and the development of the new tank facilities. The 1978 reference designs, which revealed such very high costs, used a single scheme of electrical connection for all floating devices. The scheme employed an individual 12 km length of flexible cable from each single device to a group of collecting towers where the output was rectified and added in series. This means that in a 2 GW installation there would have been enough cable to reach Australia laid initially in lines 30 m apart, but moved by the sea into a less regular pattern which might make identification of a particular length awkward.

To calculate the rate of failure of this arrangement its designers drew on the experience of the North of Scotland Hydro-Electric Board who operate nearly 80 underwater links. They used the figure of 2 faults per hundred kilometre year (the actual figure is 1·37) and concluded that there would be a failure every thirty hours. Knowing the mean time to repair faults and the lengths of the weather windows it was possible to calculate the number of repair ships necessary and the amount of energy that would be lost. The answer was very high; high enough to stop wave energy.

If we examine the raw data on which these calculations were based, we realise that the present cable runs are usually across short gaps of water where currents and marine traffic are both high. We also find that a large proportion of the faults are caused by ships' anchors (10/45) and by clam dredging (8/45). Clams, it seems, like to be near electric cables and grow fatter as a result. This fact is well known to clam dredgers whose gear is well adapted to inadvertent cable recovery and whose respect for restricted areas is not marked. It also turns out that the faults are not evenly distributed but are concentrated among a few rogue runs. In particular the cable to Jura suffered faults far above normal (7/45) because of particularly severe geological conditions unknown at the time the cables were laid. 50 of the 78 cables, some of them going back to 1933, and many to the '50s, have never failed. If we base our statistics on the practices which were used for the most reliable two-thirds of the cables, then the conclusion would be that undersea transmission of electricity is totally reliable.

The 1979 scheme for transmission for ducks is the result of work by Cure and Sullivan of the Scottish Offshore Partnership.[17] Their guiding principle has been to reduce the amount of submarine cable and to put a large effort into route surveying. To do this, they have exploited to the full the synchronous generation provided by the gyros and the chance to send electricity along the protected path of the ducks' backbone. This route can use parallel connections with air-spaced 3·3 kV cable. They conclude that the best balance of costs between cable in the backbone and cable to shore is achieved by having about seventy ducks collected together giving power levels above 100 MW. Each connection feeds an individual winding of a step-up transformer, fitted inside the backbone of the central duck pair, which combines the outputs and raises the voltage to 132 kV. Protection is by replaceable fuse. Groups of three transformers are parallel-connected for the 40 km run to shore.

Sullivan and Cure are specifying low dielectric constant crosslinked polyethylene cable which has a lower capacitance per unit length than oil and paper designs and could be used for distances up to 100 km. They are fitting compensating shunt reactors at each end of the run. The length of cable is one twentieth of the 1978 reference design. The cost of the transmission is about £100/kW to connect to the grid.

The link between the moving duck string and the rigid cable on the sea bed is one of the more difficult problems because no flexible marine cables at this high voltage are yet available. Perhaps this is because there has not been any previous demand. As we need so few lengths and as the power level is substantial, we can afford an expensive design.

To summarise:

(*a*) We should minimise the lengths of submarine cable.

(*b*) We should survey the routes carefully and use the best practices developed by the Hydro-Board.

(*c*) We should discourage ships' anchors and clam dredgers.

10 Conclusions

Many people have suggested useful methods of device classification in the field of wave energy. Ducks could be described as asymmetric, nearly submerged, surface piercing, deep-water, low-Q resonant, crest-spanning, hard-skinned, close-packed, end-stop free, capsizeable, overload-shedding, rotary terminators on active mountings. Their maximum swept volume is about twice their displacement. Their mean power limit is 7·5 kW/m^2 of projected area, and about 0·6 kW/t. Conversion efficiency is about 0·8, and mean output, including conversion losses and an allowance for equipment failure, is about 13 kW/m. The high-pressure oil hydraulic power take-off is hermetically sealed, energy storing, designed for zero maintenance and electronic control. Generation is synchronous parallel-connected a.c. with surface accumulation. Ducks use bulk high-voltage a.c. transmission. Costs of electricity generated by the 1979 reference design are in the range 5·6 – 7·9 p/kWh.

11 Acknowledgments

Work on ducks is carried out by the Edinburgh Scopa Laing Wave Energy Group. Working alongside engineers from industry has proved particularly stimulating. The project is supported by the Department of Energy. This paper is an amended version of one given at the 1st Symposium on Wave Energy Utilisation, Gothenburg, November 1979. Acknowledgment is made to the Oxford University Press for permission to publish the extract from 'The Oxford English Dictionary'.

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