Power conversion mechanisms for wave energy

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Abstract: It is easy to make a device that will respond vigorously to the action of sea waves. Indeed, it is quite hard to make one that will not. However, the conversion of the slow, random, reversing energy flows with very high extreme values into phase-locked synchronous electricity with power quality acceptable to a utility network is very much harder. This paper describes a range of different control strategies of varying degrees of sophistication and then describes possible conversion equipment for high-pressure oil and water and low-pressure air. Like many renewable energy sources, waves would benefit from some form of energy storage, particularly if they are to be used in weak island networks. Flywheels, gas accumulators, submerged oil/vacuum accumulators, thermal stores and reversible fuel cells are discussed, with emphasis on the coupling hardware. This leads on to a description of a new type of hydraulic machine with digital control which has been specially designed for high efficiency and flexible control of multiple erratic sources. An explanation of the rather low performance of air turbines in the random air flows caused by sea waves suggests the need for a variable-pitch version of the Wells turbine and the design of a reciprocating-flow wind-tunnel with recirculating energy recovery in which it could be tested.

Keywords: wave energy, reactive loading, complex-conjugate control, rolling seal, vacuum expansion storage, ring-cam, poppet valve motor, fast hydraulic machines, pressure accumulator, high-speed stop valve, variable-pitch turbine, reciprocating-flow wind-tunnel

NOTATION

B	bulk modulus of oil
C_D	drag coefficient
C_L	lift coefficient
E, ΔE	energy, energy change
F_D	aerofoil drag force
F_L	aerofoil lift force
pH	acidity
P, P_1, P_2	pressure, start, end
V, V_1, V_2	volume, start, end
α	angle of incidence to an airfoil
γ	ratio of specific heats of a gas
ρ	density of flywheel, accumulator
σ	safe stress of flywheel or accumulator

1 INTRODUCTION

With the exception of overtopping devices such as the Norwegian Tapchan, which has defied many attempts at classification, most designs of electricity-producing wave plant can be divided into the following subsystems:

(a) a slow primary displacing element, such as a float, flap or column of water, which will have interesting hydrodynamic interactions with the waves as it sweeps an area through a distance that may be larger than the wave amplitude;
(b) a connecting 'linkage' which must transmit the required forces and either allow a sufficient range of movement to escape from excess force or be strong enough to resist the worst the sea can do. The links may be ramrods, cables, struts, hoses or merely columns of air;
(c) means for increasing velocity and rectifying its direction, not necessarily in that order;
(d) a fast unidirectional rotating part, ideally but not always turning at synchronous speed, driving an electrical machine;
(e) means for providing an 'equal and opposite reaction', which may be attachment to the sea bed, to volumes of water, to inertial masses or to waves moving in the opposite direction;
(f) an electrical network of cables, transformers and switchgear for combining and transmitting the power to shore.

This paper deals with the parts between the linkage

This paper is adapted from an unpublished report written for the European Commission JOULE wave energy initiative with some updating to take account of recent developments. The MS was received on 7 January 2002 and was accepted after revision for publication on 15 July 2002.
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and the electrical output, with particular emphasis on the control strategies which the various types of hardware allow.

2 CONTROL STRATEGIES

Many control strategies of varying degrees of sophistication are possible. At a fixed wave amplitude and at the resonant frequency of the primary interface, there is little to choose between any of them. It is the *range* of wave input frequencies and the even wider range of wave amplitudes that give the more advanced strategies their advantage. This may amount to a factor of 10 or more in overall productivity. Alternatively, it could allow the use of smaller plant and lower wave forces for the same productivity. The strategies listed below are to some extent independent of hardware although some kinds of hardware may make the more sophisticated strategies easier to implement. Some rather unsatisfactory ones are included for completeness.

2.1 Fixed amplitude, variable force

This strategy is often the first idea produced by wave inventors who have been conditioned by early exposure to the crank mechanism. A moving element drives a crankshaft via a connecting rod. The amplitude of the motion is fixed by the throw of the crank. The shaft speed is locked to the wave period. The conversion mechanism has no way to yield to forces above its design rating so that, in extreme waves, these could be dangerously large. Some form of energy store, usually a flywheel, is needed to keep the mechanism moving past top-dead-centre. If this has enough kinetic energy to keep the shaft rotating during calm periods, the system will then become a wave *maker* rather than an absorber. The fixed-amplitude strategy could be made to work if waves were always the same size. Sadly they are not.

2.2 Fixed force, variable stroke

In this strategy the primary interface can deliver work through a rectifying mechanism such as a non-return valve if the force on it exceeds some constant threshold. Small waves, which produce force less than the threshold, will deliver no work and will be reflected. A vertical flap will produce an antinode while a horizontal raft will produce a node. Larger waves, which might have been able to deliver forces above the threshold, will not be able to deliver their full energy so that output tends towards the first power of wave height rather than the square. Because the reflected amplitudes are set by the velocity mismatch but the reflected energy is set by their square, a carefully chosen fixed force can perform fairly well for one wave amplitude and the method does not waste energy in calm conditions. Power conversion mechanisms which have force or torque limits (essential in any economically designed system) will behave like fixed-force ones after their maximum values are exceeded, so this is the high power mode for many designs. The primary interfaces of wave converters usually have enough inertia and added hydrodynamic inertia to smooth out the jerkiness which would otherwise be present. Fixed-force power conversion has one attraction in that many separate converters can feed a common energy path so that aggregation and larger generator sizes are possible. For this reason it has been called manifolding. Analysis by Count [**1**] shows that, in the Gaussian distribution of real sea wave amplitudes, manifolding produces about half the output of a plain, variable-force strategy.

2.3 Fixed velocity, variable rectified force

This employs a mechanical rectifying device like a freewheel ratchet, or the coupling of one or more displacer elements to a common rotating shaft. If the waves drive the displacers more slowly than the shaft rotation then the rectifying mechanism is disengaged and transmits no force. At larger wave amplitudes the rectifiers engage but displacers have to move at whatever the shaft velocity happens to be, often a velocity linked to the grid frequency. Again there will be some reflection when the force is not ideal. It could be argued that the wear and idle losses of a disengaged mechanical rectifier are more serious than those in a non-return fluid valve.

2.4 Independently dictated fixed force

This strategy could result from an attempt to improve the performance of the fixed-force strategy by changing the force threshold to suit different sea states. If the force level is changed slowly, say on an hour-by-hour basis, there would be a significant improvement. However, if the force is dictated by the content of a rather small energy store which changed over the period of a wave group (such as an undersized pressure accumulator), then small waves following a period of high activity would be locked out while large ones following a lull would be underused.

2.5 Force proportional to velocity

In this strategy each wave will feel a variable force which will depend on the instantaneous velocity of the moving element. The ratio of force to velocity (or torque to angular velocity) is called the damping coefficient. Each wave will now be able to contribute its fair share of the work regardless of its size. Power output will depend on the square of wave height. If the frequency of the waves is

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the same as the natural frequency of the moving element and if the damping coefficient has the correct value, then the extraction efficiency can be very high. Values for damping coefficients can be calculated analytically for a few shapes and numerically for many more. They can be adjusted by trial-and-error with a tank model or measured by driving a model in calm water and observing the ratio of force to velocity for each frequency. Provided that particular care is taken to suppress reflections from the tank walls, this reverse operation gives very close agreement with numerical techniques. Unfortunately, the value of the ideal damping coefficient is not necessarily constant with frequency. Furthermore, the frequency of the waves does not always match the natural frequency of the moving element.

There may also be reasons why the operator of a wave plant would choose to work with a damping coefficient different from the theoretically ideal value. For example, the range of movement of some designs, air-bag devices in particular, is limited; in this case it might be better to set a higher damping coefficient so as to defer the moment when the maximum range of movement is reached. A factor of two deviations from the ideal coefficient loses only about 15 per cent. Similarly, the air turbines of oscillating water columns may not be able to generate enough pressure to match the higher waves, and so the column may be under-damped.

2.6 Reactive loading or phase control

This strategy can be used to widen the efficiency band of wave plant to either side of the resonant period. To understand this, the resonance of the primary moving part needs to be known. The moving elements of all wave plant, even air bags, have inertia. This consists of the obvious inertia of steel, concrete or water column plus an extra, less obvious, term which is the inertia of the adjacent water. This extra term is often similar in magnitude. Inertia is what resists acceleration. It depends on the device dimensions but is also increased if the movements of the water around the device are different from those in a propagating wave, because water has to be given an acceleration to get into the correct pattern of exponential decay with water depth. Unlike the inertia of an object being accelerated in air, it need not be constant with frequency.

Many but not all wave devices also have a 'spring' term. 'Spring' is what resists deflection. For example, pushing a float below the surface requires a vertical force which is proportional to the area of the water plane and the distance pushed. If the force is removed, buoyancy will move the float back to its original position, just like a spring.

When a system consisting of a spring and an inertia but no damping is vibrating at its natural frequency, the amount of potential energy stored in the spring at its maximum deflection is exactly the same as the kinetic energy of the inertia when it is at its maximum velocity. The vibration involves a transfer of energy between the two forms. If there is absolutely no damping, the vibrations would continue indefinitely. The value of the natural frequency in radians per second can be calculated by dividing the spring by the total inertia and taking the square root.

When the incoming waves have the same frequency as the resonant frequency of the device, its behaviour depends only on the damping and the force is in phase with velocity. If this has the right value then efficiency is at its maximum. However, wave energy comes at different frequencies spread over a range of about two. At each side of the resonant period the efficiency is reduced because some of the wave force goes into deflecting the 'spring' or accelerating the inertia. It is stored temporarily as either kinetic or potential energy and then returned to radiate as a wave reflection. The forces associated with damping should be arranged so that they dominate those associated with spring and inertia. The ratio of damping to spring force (which has the dimension of time) and damping to inertial force (which has the dimension of frequency) are interesting parameters for device comparison.

Reactive loading introduces a phase shift into the power take-off force so as to cancel some of the undesirable spring or inertia and keep the device looking like a pure damper. A simple analogue network in the control system can introduce negative spring, which helps the device to work better in long waves. So far it seems harder to use negative inertia to help in short waves.

Reactive loading is better applied to improve a good device which is already resonant at nearly the correct part of the spectrum rather than to rescue a bad design. One problem for devices with highly reactive characteristics is that the energy needed to deflect the spring may be many times higher than the useful energy going into the damper. If the transfer of energy into the spring is not very efficient, an unacceptable fraction will be lost during the two passages in and out of the spring element. The comparison with the losses of inductors and capacitors in electronics is directly relevant. Designers are always trying to reduce the size of wave absorbers which usually (but not always) makes them resonant at frequencies that are higher than optimum. Reactive loading can then be used to restore resonance, but transfer losses will limit how far this can be taken.

Consideration of the phases of force and velocity shows that perfect reactive loading in an off-resonance device requires some reverse operation of the power conversion mechanism so that power is coming out for part of the cycle, with motors working as pumps and generators working as motors. This investment should be returned with interest, but this may not be possible with some kinds of powertrain. The Edinburgh Group built electronics to prevent this reverse power flow in a model

duck. It was found that there was not a large reduction in total output. Most of the advantages can be obtained by phase-shifting the positive parts of the cycle and, for the duck at least, not using the negative ones.

Full reactive control with reverse power flow is easy to apply with variable-displacement hydraulics if pumps can behave as motors. Small amounts could also be applied to a variable-pitch Wells turbine, like the air equivalent of a Kaplan machine. It would be harder, but still perhaps possible, to vary the damping coefficient of a very low-inertia, fixed-pitch turbine through the wave cycle and so simulate phase-shifting by rapidly changing the rotation speed. A lower turbine speed means a lower damping coefficient. Making it low at the start of a power cycle and high at the end would look like negative spring and so would lower the resonant frequency. Some Wells turbine users claim that the inertia of their generators provides a useful amount of kinetic energy storage, and so it is difficult for a turbine-plus-generator combination to make a large change of rotation speed within a fraction of a wave period. The moment of inertia of a hydraulic link between the turbine and generator is very much lower and so might allow approximate reactive loading by variation of turbine speed. Hydraulic coupling might also provide easy access to an energy store and would allow several turbines to drive a larger generator.

2.7 Latching

This strategy was developed by the Norwegians Budal and Falnes [**2**] to improve the performance of their heaving buoy. Their device was very small in comparison with many others, and so its natural frequency was higher than desirable. They found that performance could be improved by latching the buoy so as to delay any movement until the wave force had built up to a sufficient level. The buoy would then be released so that its velocity would be more nearly in phase with the exciting force of the wave. The Norwegians developed methods of predicting the correct moment of release in a mixed sea. A latch has to be operated very rapidly to reduce the time for which partly engaged brake pads are slipping with a large force. This difficulty would vanish immediately if the power take-off used a ram feeding appropriate hydraulics. Friction brakes would be even worse for slowing a wave device already in motion, but this should never be needed.

Latching of an oscillating water column has also been proposed by Salter and Taylor [**3**] and could provide the further advantage of protecting the air turbine from water impact. The valve has to offer no pressure drop when open, be strong enough to stand up to water impact when closed and operate in a fraction of a second. This rapid movement is important because of the losses of a partly open choke valve. The scheme used a textile-reinforced membrane [**4**], a little lighter than a car tyre, which could be inflated to a diameter of 1.7 m to block the turbine passage or deflated in order to cling to the inner wall of the fairing in the turbine hub. The actuation came from internal pressure or vacuum generated by a piston. It was held at either end of its stroke by a hydraulically operated brake (built by Richard Yemm) which could operate in 1.8 ms with the entire valve movement taking only 80 ms. Operation of the brake at the instant of zero piston velocity allows nearly all the actuation energy, more than 10^5 J, to be recycled from the previous stroke, with the losses being made up by pumping air from the lower to the higher pressure compartment. This means that peak actuation power of over a megawatt can be provided from a mean power of only a few hundred watts. The rest of the system is being built by Luis Gato for installation in the Pico oscillating water column [**5**].

2.8 Unlatching

This method is the dual of the previous one, but is used for the less common case of a device like the mace which has a natural period *longer* than the useful part of the spectrum and a range of movement unlimited by end-stops. The primary moving element is allowed to move freely for part of the cycle and then the power take-off mechanism is engaged at the desired velocity. The problems of sudden engagement of a friction clutch are just as unpleasant as those of the brake but, again, hydraulics can provide a satisfactory answer.

Unlatching is also being considered by Justino and Falcao [**6**] for the Portuguese oscillating water column in Pico where a parallel valve would be used to vent the air flow from larger waves so as to prevent the blades of the fixed-pitch turbine from going into stall. This is much easier to arrange than the series stop valve, but may lead to rather large column excursions which are a major concern to oscillating water columns that have been well designed. With a low-impedance bypass valve there may be very little loss of energy, because some can be stored as kinetic energy of the column and reclaimed later in the wave cycle. However, this will produce an impedance mismatch and so lose energy from wave reflection, increasing with the square of the reflected amplitude. Unlatching is used very effectively in the IPS buoy to limit the range of stroke of the power-conversion rams to an economic value.

2.9 Full complex-conjugate control

This is reactive control taken to the limit. All the spring and all the inertia at all the useful frequencies are cancelled and the damping coefficient is adjusted to the correct value for each of the frequency components of the incident spectrum. The control task is difficult and

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it is known that a perfect closed-loop controller would be unstable. However, Nebel [**7**, **8**] has used an open-loop control which had advance information about the phase, frequency and amplitude of the input to simulate complex-conjugate control. He was able to get an almost flat 100 per cent efficiency curve for a duck in a narrow tank in waves small enough not to exceed power or linearity limits. The availability of digital signal-processors may open many new ways of manipulating amplitude and phase which are not at present possible with analogue networks. Mathematical methods exist [**9**] that allow reasonable predictions of the near future of a wave sequence. The results could be used to calculate the correct phases and amplitudes of the responses which could be derived from a look-up table having any level of complexity that can be devised.

There is no need for complex-conjugate control, or the closest achievable imitation, to be limited to a single mode of movement. In the general case of an arbitrary body shape, wave excitation in one mode will cause movements in other modes so that, for example, surge forces might induce pitch velocities. The perfect multi-axis controller would take account of all these cross terms.

2.10 Porcupine control

This mode of control is not a practical suggestion but is included to help the understanding of power take-off and to deflate any aura of extreme difficulty. The porcupine controller was built for the narrow tank at Edinburgh University to test arbitrary control methods for regular wave tests with a duck. The cycle of the wave period was broken into a set of 16 intervals. An individual power take-off force or power take-off damping coefficient could then be set for each 22.5° phase interval. An XY display of duck torque against duck angle (an oblique ellipse analogous to the indicator diagram of a steam engine) could be made to look like the best reactive control or could be grossly distorted so as to justify the name porcupine.

It was a surprise to discover how little effect extreme distortions had on overall efficiency, provided that the 'centre of gravity' of the energy was taken at the right phase. Even if loading was removed for only half the cycle, the productivity (if not the power quality) was acceptable. The lesson is that if you are nearly right, then the waves are very forgiving, at least to ducks. Quite a small number of step changes at the right times will do. Readers with porcupine phobia could rename the method 'variable time–amplitude pulse control'.

2.11 Pre-emptive over-damping

This strategy can provide a small but useful increase in overall productivity in larger sea states. If a way can be found to predict that an imminent peak will exceed a power, torque, force or excursion limit, then it will be better to over-damp early so that the powertrain will be brought to its maximum rating sooner and stay at that rating for longer than would be the case for maximum efficiency of plant with a higher power rating. This gives a squared-off force waveform and better plant use. In addition to the higher energy yield there is a welcome reduction in the excursion of moving elements. The short-term predictability of waves over, say, one-quarter of a wave period is reasonable, and sonar equipment or seaward wave-rider buoys that could improve accuracy are a possibility. The latter will be a likely feature of experimental installations.

2.12 Array cooperation

Work by Evans [**10**] on arrays of absorbers showed that performance can be enhanced for some combinations of wavelength and direction, and also reduced for others. However, this assumed that each absorber was oblivious of the conditions of its neighbours. With intelligent cooperation between members of an array it should be possible to reduce the counter-productive effects. A small investment in communications and computing might produce a substantial benefit. Further mathematical work should be encouraged. Plant designers should ensure that the communication facilities can be provided and that any future sophistication of control strategy can be implemented. To do otherwise is to invite obsolescence.

2.13 Stiffness modulation

During early work on duck models it was found that the position of the centre of gravity greatly affected the conversion efficiency. Placing it high produced a negative spring effect like an inverted pendulum. However, placing it *too* high prevented automatic recovery after capsize. Although the official cost assessment was made on a single ballast arrangement used for the whole year it would have been possible to move water around the duck body so as to tune it for different spectra. The most useful one would be a dangerously high one, safe only in small wave amplitudes.

A similar technique can be used with a sloped version of the IPS buoy [**11**] in which the angle of slope of an IPS buoy tunes the stiffness. This can improve the long-wave performance by a factor of four relative to the same device moving in the vertical direction.

French and Bracewell [**12**] proposed pneumatic stiffness modulation. A device with a hydraulic power conversion can be provided with additional pneumatic rams which can be connected through valves to a set of external gas vessels. The operation of the valves could select

the size of the external volume and thus the amount of pneumatic stiffness.

A related idea has been proposed for the Evans cylinder, which has extremely high conversion efficiency but no inherent spring to resonate with its inertia. The amount of gas on either side of the piston of a double-acting ram can be adjusted to change the effective ram stiffness. Neither pneumatic method can produce negative stiffness values.

3 HARDWARE

3.1 Slow linear pumps

An obvious input mechanism for taking power from the primary moving element is the conventional linear hydraulic ram. It could be used for the spherical buoy of Budal and Falnes or the heaving float of Nielsen. Groups of rams could be used for the tension legs of the Evans cylinder or the solo duck. Opposed pairs of rams could be used in the PS Frog developed by Michael French at the University of Lancaster and for the Swedish IPS buoy. A group of twelve rams arranged in a truss shaped like the edges of a coronet has been designed for the Edinburgh duck to give all six degrees of controlled freedom at the force levels needed for large spines [**13**]. Yemm *et al.* used rams for the Pelamis [**14**].

A well-designed wave device could easily demand velocities higher than are comfortable for the average hydraulic seal. However, this may reflect a lack of land-based demand for high-velocity seals rather than any insuperable technical difficulty, and some manufacturers do offer speeds of up to 30 m/s given sufficient cooling. The essential problem is the temperature rise caused by shear loss and friction at the moving interfaces. A cheap ram can suffer a friction force which is several per cent of its useful thrust, which often proves to be a predictor of wear. Things could be improved by using hydrostatic seals for the piston and the rod gland, giving them carefully chosen leakage rates by cutting a fine screw thread with a flat top on one face of the seal. The leakage from the rod gland would have to be scavenged and returned. It would be desirable to have on-line control of the leakage rate as a function of instantaneous pressure and velocity. The necessary technology could be demonstrated with small-diameter seals running at elevated pressures and sliding velocities.

There are now no great concerns about the problem of using low-velocity rams at sea because of 30 years experience with Ceramax coatings on ram rods in direct contact with sea water [**15**]. Van Dijk reports that the rods remain in an excellent condition even though the cylinders are covered with barnacles. This material will find many uses in wave energy for other seal faces. The only concern about ram rods is that very small quantities of sea water might conceivably be drawn into the ram cylinder with each retraction and gradually contaminate the oil, causing problems elsewhere in the hydraulics. The severity of the effect will depend on the pore structure of the ceramic coating and the relative affinity of the material for oil and water. If water ingress does occur there are several possible solutions.

Rods can be given additional protection with a rolling 'Belofram' seal, as shown in Fig. 1. The requirements for this are not too daunting and can be exactly specified. The material of the seal must be able to bend round the curvature defined by the gap between the cylinder and shroud. This is a bending fatigue problem. Big gaps between the piston and cylinder help.

The seal must have enough tensile strength to withstand the force of the hydrostatic pressure acting over the annular gap between the shroud and the cylinder body. There will be a wave-induced variation on top of the standing pressure of the depth of immersion. Small gaps help.

The material must have enough abrasion resistance to stand up to the small stretch that will occur as a result of pressure variation. Sliding could be reduced if the texture of the finishes on the inside of the shroud and outside of the cylinder matches that on the mating elastomer surface.

The elastomer must change diameter from that of the outside of the cylinder to that of the inside of the shroud tube. This will be a potential fatigue problem, but most elastomers can tolerate a 10 per cent strain indefinitely. If the reinforcement included to resist tension is arranged axially or with a very acute 'V' lay, the material will be

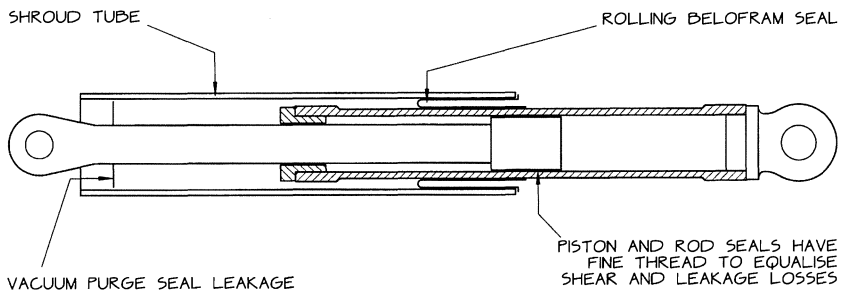

Fig. 1 A rolling seal inside a shroud tube keeps the rod of the hydraulic ram away from sea water. However, with Ceramax surfaces, unprotected rods can work in sea water for many years

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able to expand easily. Close gaps between the piston and cylinder help. The large costs of a hot-cure mould could be reduced by spraying polyurethane on to a suitable former, laying a nylon-reinforcing layer strong enough to resist the external pressure and then spraying a second layer of polyurethane. Belofram technology can find many useful roles in wave energy if units of sufficient size can be produced. Caldwell and Taylor [**4**] made 1.2 m diameter rolling seals for the Pico actuator, using a machine that could make diameters over 2 m.

Hydraulic machines, particularly ones that make extensive use of hydrostatic bearing technology, can work well on 95 per cent water/oil emulsions. Work is in progress to develop material combinations that will allow poppet valves (to be described later) to work in water, because it has many safety, environmental and bulk modulus attractions. However, if adequate life cannot be achieved with water, thermal or chemical 'oil-conditioning' could remove contamination from conventional oils.

Industry can produce rams with strokes up to 24 m and diameters up to 1 m. The piston and ends of the ram, particularly the rod guide, account for a large part of the cost, so that long rams are not proportionally more expensive than shorter ones. In very long units it is necessary to consider the rod buckling problem, particularly if side loads are present at maximum extension. However, if the ram force is proportional to velocity, its maximum value will occur at mid stroke. Long rams can suffer problems in land-based applications if they are used in a horizontal attitude because the weight of the unit causes a central bending moment which must be resisted by forces on the guide and the piston. If these get too close to one another, the forces can get dangerously high and a minimum distance is often stipulated. Vertical installations used for heaving buoys will suffer less, but the problem will have to be considered for the legs of the Evans cylinders.

The specification of extension range is difficult. One policy might be to design for the extreme trough-to-crest wave height predicted for the wave site. This might be longer than available strokes for good Atlantic sites and would mean that a large quantity of accurately machined equipment was idle for nearly the whole of its life. Another approach would be to subtract from the extreme wave height an amount corresponding to a wave load that can safely be taken by the ram attachments. Yet another would be to devise a displacer shape which shed wave loading after a chosen wave height. Tests on several kinds of model have shown that heave motions are often less than the wave height because of the conflict between inertia and buoyancy forces. This is because there can be no increase in the buoyancy force once a float is fully submerged. However, this provides no relief from horizontal forces. Furthermore, the wave climate experts are busily trying to find good wave hot-spots and the hydrodynamicists are busily trying to make buoys resonant at the most useful parts of the spectrum. It would be annoying to have chosen a value for ram extension and then find that some advance in point-absorber theory could not safely be applied. The most dangerous case would be if the resonance had been achieved without the aid of reactive loading, so that extreme amplitudes occurred without limit after a power failure. The kinetic energy that can be built up in an unrestrained wave displacer cannot be suddenly absorbed by hitting the end of the ram travel. It was this problem in the tension legs that stopped work on the solo duck which otherwise had very attractive productivity. The end-stop problem is the most serious limitation of ram technology and an inevitable outcome of the linear geometry. Only the IPS buoy has a neat way of removing the ram thrust at the end of the range. For all other ram devices, some form of independent mechanism with a travel of several metres at the maximum force rating will be necessary if rams are to be used safely.

3.2 Connections

Flexible hydraulic hoses such as those used for the brakes of road vehicles are a great attraction in low-power hydraulics. They give designers freedom to place equipment where it suits them and allow various parts to have relative movement. Unfortunately, there seem to be no sources of flexible hoses and hose unions with the diameter, pressure rating and flexure life necessary for even quite small wave energy applications. A 1 MW rating at 400 bar would need a hose with a 100 mm bore and the biggest available is only 50 mm. The problems of making such a hose are daunting and the results are likely to be suitable for only very small flexure angles.

Anecdotal observations from the hydraulics industry and evidence from the flexible cable tests performed by McConnell [**16**] show that hoses are a notorious cause of failure—much worse than flexible electrical cables. This means that things are made more difficult by using large numbers of smaller-diameter hoses in parallel.

One solution is to choose configurations in which the oil can be taken from the hydraulic cylinder to other parts of the system through rigid steel piping. This leads to the concept of a power cartridge with ram, cylinder, hydraulic motor and electrical generator in one integral package. This arrangement has been proposed for the Bristol cylinder. Another possibility is to take oil through a hollow cylinder rod and its end-swivels. It might even be possible to use rods with two coaxial paths so that oil could be taken from both sides of the piston to a destination on the rod end. Most rams are provided with swivels as standard. However, while the angular excursion of one of the degrees of freedom is generous, the second is often smaller than would be needed to allow enough freedom to suit all wave slopes. New

designs of large swivel angle will be necessary for several devices.

3.3 Slow rotary pumps

Rotating machinery avoids the end-stop problem of linear rams. Rotation can be used directly for devices like the duck. It can be used less directly via a winch drum in many other devices such as buoys, the mace and the Evans cylinder, provided that care is taken with respect to bending fatigue. The winch configuration proposed for the mace allows bidirectional operation, but more usually winches are restricted to pulling, and so are idle for half the time. They need a small motoring effort to rewind cable if used with a neutrally buoyant displacer or a very much larger motoring effort to rewind the lighter, tension leg displacers. Large winch diameters are necessary if bending fatique of the ropes is to be minimized, and it is regrettable that this means slower rotating speeds.

The largest displacement hydraulic machine available commercially is made by the Swedish company Hagglunds. Their Marathon range goes up to a displacement of 0.15 m^3 per revolution with torque of over 10^6 N/m for the MA 4000. The units have a long record in exposed marine conditions for use as motors in winch drives. The Marathon design uses a series of hard rollers running on inward-facing cam lobes. The rise and fall of the cam profile drives pistons in chambers feeding the valve ports at the centre of the machine, which are connected to the high or low pressure areas at the correct times as the machine rotates. The arrangement provides large, smooth torque at relatively low speeds—only 8 r/min or 0.84 rad/s. The differences between pumps and motors are quite subtle and there would be no problem about modifying the Hagglunds design in the pumping mode. However, large though these torques are, they are not nearly large enough for the bigger wave applications. Furthermore, the technology is much more satisfactory when it is used in the fixed-displacement mode, whereas wave energy really needs a variable-displacement operation.

The force exerted on the cam by each roller can follow two paths to find some reaction. One path is round the circumference of the ring. The other is through the cover plate. Any residual imbalance can be taken by the central bearings. If the size of the marathon design were to be further increased, the problem of providing this force reaction path would become more difficult. It would be better to find a shorter route.

Figure 2 shows a double-sided cam with the force going through the *thickness* rather than round the circumference. The rollers running on the cam would have to be conical rather than cylindrical. Any radial lines drawn on the surface of the cam would have to pass through the apex of the cone. The lightest cam would

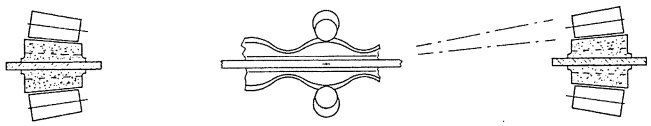

Fig. 2 A ring cam with conical rollers and reaction forces taken through the cam thickness

be obtained if there were two such apex points. While this geometry is harder to understand than that of plain cylinders, the machine tools to generate it can be surprisingly simple, with rotating rather than translating movements. The separate cam lobes can have scarfed joints so that the tickety-tack-tickety-tack noise of unwelded rails (used for over one hundred years) will not occur.

The work done by a roller depends on the force it exerts on the cam multiplied by the distance from the trough to the crest. The force is limited by the Hertzian stress between the roller and the cam surface. This depends on their curvature and material properties (see, for example, Roark and Young [**17**]). Excessive stress will eventually lead to fatigue fracture, known as spalling, at points just below the surface. The rolling bearing industry has for many years used Hertzian stress calculations for predicting the statistical life of bearings. Material combinations giving 90 per cent survival over 10^8 operations are known. The designs used here have lower contact stress values than those of railway wheels and would not have to suffer bending stress or the usual railway hazards such as rain, sand and little boys putting pennies on the line.

It is a wonderful feature of ring-cam machines that every roller can be worked by every lobe. The cost will depend on the sum of the costs of rollers and lobes, but the value will depend on their product. Thus the value-for-money becomes steadily better as the size increases. The new designs here have circumferential galleries close to the pumping modules which are connected to them through electronically controlled poppet valves. This greatly reduces the breathing losses and makes a direct interface to computers more convenient. A duplex ring-cam pump for a spine-based duck would have a displacement of 7 m^3 per revolution (which is 45 times more than the MA 4000) but it would weigh only 50 t. Figure 3 shows a section. Weight and efficiency calculations are available. Even larger units with torque up to 10^8 N/m and a quad cam are being designed for vertical-axis tidal-stream generation.

Comparisons between ring-cams and gears are informative. An ordinary spur gearbox sends all its work through the one contact line of a tooth pair while all the other teeth are idle. The common designs of epicyclic gears usually have three contact lines working in parallel. Some very advanced epicyclics with clever force sharing use five lines. The spine-based duck ring-cam uses 294 rollers driven by 109 pairs of cam lobes so that at any instant nearly 150 contact lines are active. This means

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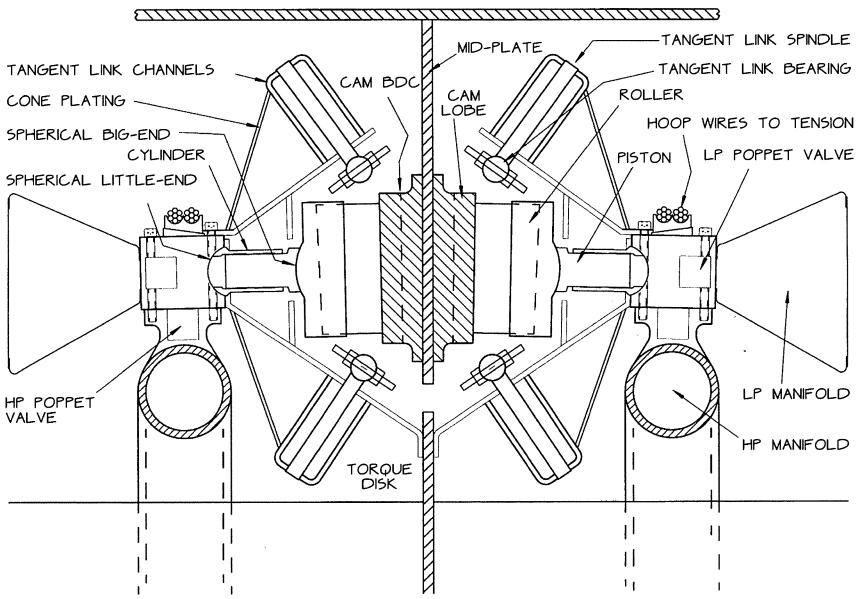

Fig. 3 A section view of a duplex ring cam designed for the duck. All the working forces pass through a common point in the valve block so that there are no bending stresses

that the ratio of weights of ring-cam to gearbox for a given torque is very great.

Involute gears are designed to transmit a constant velocity. This means that there must inevitably be a small amount of sliding between teeth. The consequent loss of energy is about 1 per cent per mesh. With ring-cams, the contact is very nearly perfect rolling, with 'friction coefficients' usually below 0.0003.

The pitch of gear teeth and the separation of their axes must be carefully controlled. In the ring-cam, the roller diameters, cam heights and wavelengths can safely have gross dimensional errors and distortion under load provided only that the conical relationships are reasonably accurate. Misalignment can be corrected by a small side-slip when the pressure is relieved at the end of each power stroke.

Gears transmit a fixed ratio and cannot be disengaged. Ring-cam pumping modules can be fitted with poppet valves which can enable or disable selected pumping elements and so control the pump output to provide variable displacement with all the benefits that this can provide. The output of a ring-cam pump can be stored as pressure energy in an accumulator.

Finally, damage to a single gear tooth will quickly infect all the others. The injured one cannot be skipped. However, a microphone on the ring-cam can detect early spalling of the cam or roller surfaces and the computer which controls poppet valves will know to avoid their future use. A long working life can be achieved by providing some spare initial capacity.

3.4 Direct electrical generation

In the early 1970s the Edinburgh team took advice from a distinguished expert on electrical machines about the possibility of using a directly coupled permanent-magnet generator between the duck and the spine. The velocity across this gap is about twice the surface velocity of the waves, often less than 1 m/s compared with the 100 m/s, which is convenient for flux cutting machines. The cheapest source of magnetic flux in those days was anisotropic barium ferrite, which has a respectable coercivity but a remnant flux density of only 0.35 tesla. It turned out that even if the design could afford the cost of iron and copper the total weight would be a severe embarrassment. Some of the smaller ducks would have sunk. There was also the problem that the output would have been a wild and rapidly changing frequency and amplitude. At that time the official technical advisory groups did not believe that this could be converted to a synchronous output. The situation has changed with the advances of magnet technology, particularly with the development of neodymiun–boron magnets, which can operate at much higher flux densities, and of switching-mode converters, which can convert any frequency to any other with rapidly falling costs. Direct electrical machines, admittedly large and heavy, are now in service for wind energy.

Mueller *et al.* [**18**] are investigating the use of transverse-flux machines for waves. A rotary device will be more suitable because all the magnets and coils can be in action all the time. There will be some interesting mechanical problems in maintaining the close air gaps on elements that are subjected to very high torques and in resisting the attraction forces between opposite poles. One possibility might be to use a fully flooded generator with pumped sea water separating poles to permit clearances smaller than any land-based electrical machine could ever have. It may also be necessary to find a way to prevent generation during maintenance. The

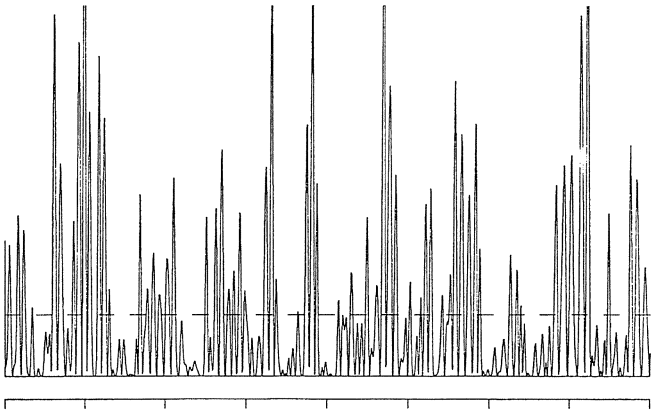

Fig. 4 The instantaneous power of a linear device working in a 10 s Pierson–Moskowitz spectrum over 8 min. The bank has to be paid for the peak capability but the customers pay only for the mean (dashed line)

semiconductor industry has a track record of very large advances in capability and reductions in cost. The benefits of an all-electric system would be large and very well worth the effort.

4 ENERGY STORAGE

The variation of water position versus time in a train of sea waves has a close approximation to the Gaussian distribution. For a narrow spectrum the instantaneous power depends on the square of wave amplitude and so the instantaneous power is close to the Rayleigh distribution. Figure 4 shows a record of instantaneous power together with the long-term mean level. The plotter clipped some of the highest spikes. These variations are the most unattractive feature of wave energy. It is a slight consolation to reflect that very similar mathematics governs the impact of steam molecules on the turbine blades of fossil fuel plant and enforces a severe limit on thermal efficiency. Figure 5 shows the sum of 16 uncorrelated units. This would still be an unpleasant contribution to a slender electricity network.

Some form of energy storage will reduce the capital cost and power losses of all subsequent elements of the powertrain. In most wave spectra the provision of storage for about 10 times the wave period will give an almost constant output. A smaller amount, say three times the wave period, would allow a gradual variation which would slowly ramp up and down at the rate of wave groups, within the response rate of diesel plant on land. However, the operation of diesel plant is much more satisfactory if it can run steadily and a constant output from a wave plant is undeniably attractive. A generous amount of storage in wave plant and efficient import–export would allow it to provide welcome assistance to land generators on small islands rather than an unwelcome intrusion. This bidirectional flow is easier with synchronous a.c. systems than with d.c.-to-inverter networks. The conclusion is that there is a need to store energy in quantities of the order of 10^8 or even 10^9 J per module for final designs, and perhaps 10^7 for smaller experimental prototypes.

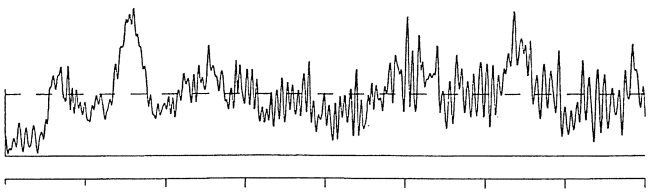

Fig. 5 The combined output of 16 widely separated devices working in the sea state of Fig. 4. This would still be unacceptable to a slender island network

4.1 Storage mechanisms

The two conventional contenders for storage are flywheels and pressure accumulators. Both are used in land-based applications but not usually in such large sizes. A third, less conventional scheme may be more suitable for some wave plant. A comparison of the methods is useful.

Flywheel storage was an incidental feature of the gyroscopic reaction frame proposed for the 1981 spine-based ducks. The speed–inertia product needed for torque reaction was much more than could be justified for smoothing the power flow. There was much dispute about the value of storage and the cost of the sheet steel gyro discs. Gyros for reaction need cheap, strong, *heavy* materials. Rolled sheets of low-carbon steel were chosen.

The newcomer to flywheel technology is surprised to discover that the ideal material for energy storage per unit weight should be strong and *light* with a thin rim rather than a thick, heavy rim like the brass wheels fitted to model steam engines. However, there are other considerations. Materials that have very predictable properties are attractive. The very high rotation speeds or very large diameters that would be needed to exploit the properties of the latest materials such as carbon fibre or Kevlar may be inconvenient. Materials must be produced with no voids. Finally, it is desirable that the material should have a long range of graceful yield to local stress rather than any sudden crack propagation.

Yielding can be used to make discs of constant thickness achieve the same stored-energy density as more carefully shaped ones. Plain wheels could be deliberately run overspeed after first assembly. The material near the centre would yield but be contained by the less heavily stressed outer region. After the flywheel slows, the central region would be left with a compressive pre-stress which could be subtracted from later stress calculations. Several authors—e.g. Roark and Young [**19**]—report that flywheels with central holes do not fail at the speeds that would be predicted on elementary elastic analysis.

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This is probably because of the compressive pre-stress at the centre. However, the explanation might not sound convincing at a court of enquiry.

Clerk [**20**] has pointed out that the centrifugal flattening effect of a conical disc will also induce a central compression at the expense of higher rim stresses. After a long study of the problem he advised that, despite having a higher weight than modern fibre composites, a wheel made from laminated cold-rolled medium carbon steel sheets offers more predictable yield behaviour and more convenient interfacing speeds than the newer materials.

The task of the designer does not stop with the choice of the flywheel. It must be enclosed in a vacuum casing (which can weigh as much again), have low-loss bearings and, most important of all, have an input/output device on its shaft which can take the highest power peaks but drain the least idle energy. Flywheels transfer energy only by changing speed. Typical speed ratios are 2 or 3, giving energy ratios down to 25 or 11 per cent of the maximum. It is not sensible to empty the last dregs of flywheel energy because the rate at which they can be recharged depends on the product of input torque and speed. If the latter is too low it takes a long time to achieve an acceptable input power rate.

If the maximum safe kinetic energy of a spinning disc is divided by its mass the expression is obtained that depends on the ratio of working stress to density σ/ρ. This should be multiplied by a shape factor, usually between 0.25 and 1, which is determined by the thickness profile and the presence or absence of a central hole. For a flat disc with no hole the factor is 0.6. A small hole reduces this to 0.3 unless some pre-stressing has been used. The kinetic energy is stored at the rim, but the highest stresses are at the hub. The problem for the flywheel designer is therefore to find a way to let the hub material retain the rim safely.

The ratio of stress upon density times the shape factor has the same dimensions as the square of a velocity, which turns out to be the safe peripheral velocity of the wheel. This means that flywheels of the same material and shape will have the same maximum peripheral velocity regardless of their diameter.

4.2 Gas accumulator storage

Energy can be stored by compressing a gas, which is the preferred method of the hydraulics industry. The rates of change associated with sea waves mean that adiabatic rather than isothermal equations must be used with the term γ appearing in many terms. This is the ratio of the specific heat of the gas at constant pressure divided by the specific heat at constant volume. For nitrogen it has the value 1.4. The adiabatic energy difference ΔE which results from moving from pressure P_1 and volume V_1 to P_2 is

$$\Delta E := P_1 V_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

A value is decided for P_1 which is the lowest pressure acceptable at minimum storage content. The dimensions of the vessel are chosen in order to set the volume V_1. The maximum pressure P_2 which will fix the necessary wall thickness is chosen, which gives the weight of the vessel. If volume and weight are important, the most efficient shape to contain a pressurized gas would be a sphere, and spheres are used for airborne applications. However, tubes are cheaper to make and cylindrical tubes with hemispherical ends are the more usual form. If the wall thickness of a tube needed to contain a pressure safely is calculated the energy–mass ratio can be worked out. If the tube ends are ignored and the gas pressure is run right down to ambient then the energy is 2.5 $P_{max} V_{max}$ and the energy–mass ratio is 1.25 stress over density.

The ratio of σ/ρ is the same indicator of energy density as for flywheels and it is interesting to ask how this could be related to any velocity in a pressure vessel. The grim, and one hopes hypothetical, answer is that it is the square of velocity of the broken fragments of a burst vessel obtained by equating their kinetic energy to stored potential energy. High-pressure gas vessels are much more dangerous than oil ones. They are sometimes compared to unexploded bombs. This is unfair and inaccurate. They are more like bombs that have already exploded but which have not yet undergone case fragmentation. It would be interesting to see if the water-bag suppression techniques now being used for explosives [**21**] are also effective with accumulators.

A tubular gas pressure store, expanded right down to zero pressure through some notional mechanism that does not occupy any of its volume, is slightly better in energy-to-weight terms than a bare flywheel, and even more so if the flywheel casing is taken into account. However, the last dregs of pressure accumulator energy cannot be used. If gas pressure is to force oil out of a container, then that container must have a total volume for both gas and oil. Pressure accumulators are used with pressure ratios which are directly comparable and of similar magnitude to the speed ratios of flywheels—often between 2 and 3. They share the same problems of power output limitation as their energy content decreases. However, a variable-displacement pump can pass high peaks directly into an accumulator without suffering any flow limitation.

Although pressure accumulators have none of the standing losses, sealing and alignment problems of flywheel bearings they do require some means of separating oil and gas. If this is not done, gas will dissolve in the oil under high pressure and come out of solution under low pressure, producing bubbles and foam in the low-

pressure parts of the system. This causes a steady loss of gas, unacceptable suction conditions for hydraulic equipment and spongy hydraulics. The separation of oil can be done with bladders or machined free pistons. There has to be some means of stopping the bladder being extruded from the vessel along with the last drops of oil. Poppet valves are usually employed.

An examination of manufacturers' literature shows that accumulator steel is stressed to about 190×10^6 N/m^2, giving a generous factor of safety for misuse even with many cycles of fatigue. With a pressure ratio of three this stress would allow storage of only about 2000 J per kilogram of accumulator. Unfortunately the manufacturers want to charge nearly £10 per kilogram for the largest sizes (50 l) in small quantities. This is close to the cost per weight of big machine tools, which is difficult to understand. Storing 1 MW for 100 s would cost £0.5 m, or £500/kW, nearly the cost of complete wind turbines. If pressure accumulators are to be viable for wave energy storage they must achieve very large reductions in cost. It is a surprise to discover that about 30 per cent of the cost is attributed to the bladder that separates oil from gas. Bladders have to be individually moulded in an expensive tool. Advice from the accumulator industry is that, for large sizes, a free piston design is cheaper than a bladder, despite the need for machined parts.

It is frustrating that there seems to be no combination of a gas and a liquid with zero mutual solubility. Advice from chemists is that, ignoring combinations with strong reactions like ammonia, the low-solubility gases would be the ones with the lowest boiling points. Helium is the preferred option. It may be possible to have helium over oil with a floating sponge on the oil surface to slow the rate of diffusion. However, this method really needs a magic new liquid floating on the oil surface, and chemists are not optimistic.

Another solution might be the use of continuous rubber tubing fitting around two pipes to prevent complete flattening. Figure 6 shows a sectional view. This form of separator could well be fitted inside any large pipework such as the radial collecting manifolds of a big ring-cam machine so that pipework could also be used as part of the energy-storing system.

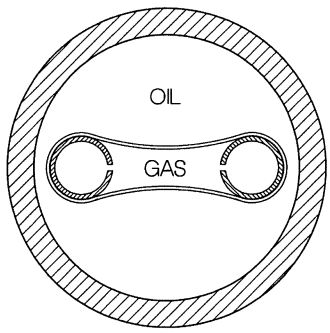

Fig. 6 A gas-separating bladder made from straight extruded tubing has well-defined changes of curvature and could be built into connecting pipes

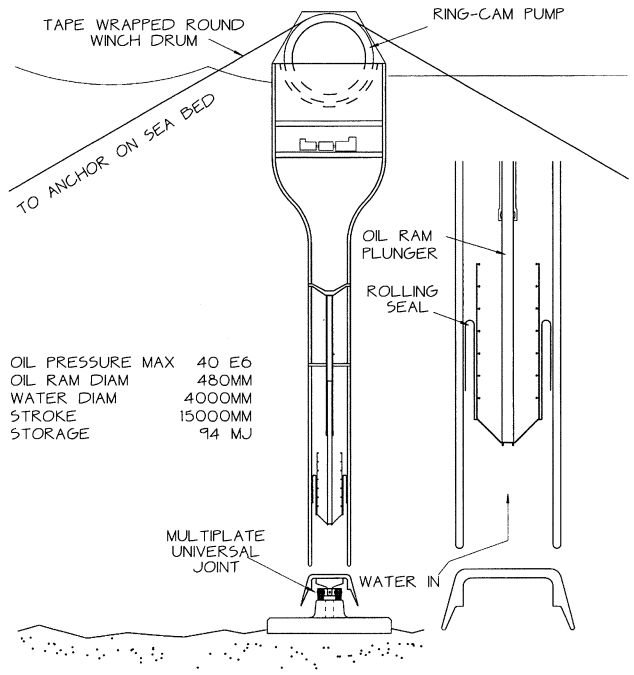

Fig. 7 The stalk of the mace can contain an oil plunger, ram, rolling-seal combination worked by the external water pressure which gives a large amount of energy in the form of oil near-constant pressure

4.3 Oil accumulator storage

An unconventional solution to the storage problem can be applied to some deep-water devices with vertical tubular members, like the Edinburgh mace shown in Fig. 7. The mace is a buoyant, surging displacer with a winch and ring-cam power take-off and an unusually long range of excursion. It has the form of a stalk which pivots about a point on the sea bed and is intended to survive most of the misfortunes that will affect early sea prototypes. Indeed its survival characteristics and convenience as a test bed are more important than its efficiency.

At a depth of 60 metres the water pressure at the bottom of the mace will be about 6×10^5 N/m^2. This pressure acts on a piston about 4 m in diameter which is sealed to the inside of the stalk by a Belofram rolling seal. The water piston force is resisted by an oil plunger rod which is pushed into an oil vessel. The plunger diameter will be related to the piston diameter by the square root of the oil pressure to water pressure and so, for a 400 bar system pressure, the diameter will be a little under 0.5 m. With a plunger length of, say, 15 m nearly 10^8 J can be stored. Some extra capacity can be obtained by evacuating the concrete stalk.

This arrangement has a nearly constant pressure

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characteristic rather than the inverse relationship of pressure to content of the gas accumulator. This reduces the amount of steel and improves the power rating at low store levels. The plunger rod would be easier to machine than the long cylinder of the free-piston accumulator. It is under triaxial compression and could be a thin-walled steel tube filled with concrete so that the energy steel–mass ratio of the whole system will approach $0.5\,\sigma/\rho$, giving about 10 000 J/kg. The low-pressure variation reduces problems of case fatigue. Further improvements may be possible if the insurance regulations (which are very strict where gas pressure vessels are concerned) can be relaxed for unmanned liquid-filled plant at near-constant pressure inside another vessel.

4.4 Vacuum expansion storage

Pressure in a sub-sea manifold could be regulated by using it to drive a piston to expand a vacuum chamber. The right shape of chamber would require mainly compressive stresses and so could be made of concrete. The idea becomes increasingly attractive for deeper water installations. Sliding seals of the required size would be a problem, but Belofram rolling seal technology would be suitable, particularly if some means could be found to distribute the pressure loading evenly between a number of rolling seals in series. If it is necessary to regulate the pressure to different levels, the expansion could be achieved by a cluster of seven Beloframs in a close-packed hexagonal array. Any number of cylinders from one to seven could be switched in to give pressure suited to the wave conditions without producing an off-axis thrust.

4.5 Chemical storage

A recent and exciting development for electrical energy storage is the Regenesys method being demonstrated by Innogy at a power level of 15 MW in conjunction with a gas-fired generator at Little Barford near Cambridge [**22**]. The method resembles a fuel cell, but uses two liquid electrolytes rather than gases. The liquids chosen are sodium bromide and sodium polysulphide. Like most electrolytic processes the voltage is quite low, only 1.54 V, but stacks of cells can be arranged in series to build a voltage suitable for switched-mode conversion to synchronous a.c. There is nearly always plenty of space for liquid storage tanks in a floating-wave energy device.

It might also be possible to make hydrogen or ammonia from feedstock available at sea, but the presence of the other materials would complicate normal electrolysis. Very large amounts of energy could be sent very long distances as pumped hydrogen. The complications of making hydrogen at sea do not sound insurmountable and would appear less so if there were to be wider perception of the need to move from carbon to hydrogen ashore.

4.6 Thermal storage

The ability of some chemicals to change phase at a particular temperature allows substantial amounts of energy to be stored. However, the transfer from heat to mechanical work is subject to the laws of thermodynamics and so has inherent efficiency limitations. To some extent these can be offset by the gain in performance of a heat pump when work is turned back into heat. During both processes there will be losses due to irreversibility and poor thermal insulation. A recent development in Stirling engine design removes the problem of the hydrogen seal by transferring the functions of piston, connecting rod and crank to a multi-input hydraulic machine to be described later. The Stirling cycle can be used for a very effective heat pump. This may allow an element of reversible thermal storage to be coupled to mechanical energy at the shaft of the generator. It also opens the possibility of a partially fuel-burning renewable energy device which can deliver firm power and keep itself safe in conditions of no renewable energy input.

5 FAST HYDRAULIC MOTORS

5.1 Conventional designs for land

In land-based applications, fast hydraulic machines are normally used as pumps and are driven by induction motors or diesel engines providing oil for slower actuators. With waves, power moves in the opposite direction and so fast *motors* are needed. Many designs of hydraulic machines date from the energy-rich days when there was less incentive to improve efficiency for all operating conditions. Some designs have a peak efficiency of just over 90 per cent, but performance falls badly at pressures, speeds and displacements away from the optimum.

In wave energy there is a much stronger incentive to improve efficiency, and in particular the part-load efficiency, because electricity is the only product. A one per cent improvement in the efficiency of a hydraulic machine driving the electrical generator will add one per cent to the value of all the civil structures and all of the power-handling plant upstream and so might justify *doubling* the cost of the hydraulic unit. The gain is more if machines are used in series. A detailed list of the loss mechanisms provides a useful insight into the problem. The losses are as follows.

5.2 Solid-to-solid friction

There are still many designs in which the useful work is passed as a force between two components with relative movement. No matter how careful the selection of materials and surface finishes, it is difficult to get a coefficient of friction below 0.05. Such contacts are very sensitive to dirty hydraulic fluid and will themselves produce debris as they wear. Furthermore, if the parts wear, they may change the intended hydraulic clearance so that leakage increases. There is hardly ever any need for solid-to-solid friction to resist those forces that are caused by hydraulic pressure because hydrostatic bearings can provide separation or very large reductions in contact force. For the pistons of a swash plate machine, which have at the same time to be freely sliding hydraulic seals and heavily end-loaded cantilever beams, the fractional loss will be about the value of the coefficient of friction times the square of the swash angle in radians. Solid-to-solid contacts mean a short life and poor reliability.

5.3 Churning

Many hydraulic machines run with their insides fully flooded. This means that any oil in contact with the moving parts is accelerated to their velocity and then thrown off to dissipate kinetic energy against the case of the unit, just as if it were a hydrokinetic torque converter. For fast machines the loss can be several per cent of the full power rating, and it remains at this level at all pressures and displacements. The use of a dry sump removes the problem, but requires careful attention to the lubrication of every moving interface.

5.4 Leakage

Hydraulic fluids under pressure will flow through any available gaps and so will lose energy equal to the product of pressure and flow. The equations that govern leakage flows are well known and easy to apply, except for two uncertainties. The first is that the leakage depends on viscosity, which depends on temperature, which in turn depends on the energy being dissipated in the oil. The second is that the flow depends on the cube of clearance, which depends on the distortion of parts under pressure, thermal expansion, wear and part tolerance. Low leakage is obtained by having fine clearances, high viscosities and long leakage paths.

5.4 Shear

If two parts move relative to one another and are separated by a thin layer of fluid there will be a viscous force opposing movement. The power loss will be the product of this force times the local shear velocity. It can be minimized by having large clearances, low viscosities and low shearing areas, which will in turn imply short leakage paths. These requirements are directly opposite to those for low leakage loss. The task of the machine designer is, therefore, to pick the best compromise between these variables for a given combination of pressure and velocity and acceptable temperature rise. The machine layout should minimize the leakage perimeter and tangential velocity of fast pressurized interfaces. In many parts of a machine the absolute minimum loss will occur with a leakage loss one-third of the shear loss, but the safer, normal practice is to equalize these so as to reduce the exit temperature of the fluid. Because leakage power loss depends on the square of pressure and shear power loss depends on the square of speed, it would be very desirable to keep pressure and speed in proportion so that these losses become a constant fraction of the throughput. Unfortunately, as synchronous generation demands fixed rotation speeds while the pressure of a gas accumulator usually depends on its contents, the ideal of constant efficiency may not be achievable.

5.6 Compressibility

Although hydraulic fluids are often regarded as incompressible, their bulk modulus B is not infinite. For water at 15 °C the value is 2.05×10^9 N/m^2. For typical oils the value is about 1.6×10^9. If a chamber of oil of volume, V, is subjected to pressure, P, it will store some potential energy from compression. The amount of energy is

$$E = \frac{1}{2}\frac{VP^2}{B}$$

In a motor chamber the volume under pressure will be larger than the volume swept by the piston because of the dead space above the piston. If the total volume is K times the useful swept volume and if nothing is done about recovering the compressibility energy, this will lead to a fractional loss of

$$\frac{KP}{2B}$$

At high pressure—say 400 bar with $K = 1.2$—this loss is 1.5 per cent of full power. Similar amounts of energy can be stored in the elastic deformation of chamber walls and connecting rods. They will be lost if the chamber to which they are connected is suddenly depressurized. There is a further problem with compressibility loss, which is that the stored energy will be dissipated, noisily and destructively, if a pressurized chamber is suddenly connected to a gallery at a different pressure. The problem is worse with motors than with pumps because a change from high to low pressure happens at bottom-dead-centre with a full cylinder rather than at top-dead-centre. One solution used by designers of swash plate and bent-axis machines is to choose the angles of kidney ports so as to adjust the

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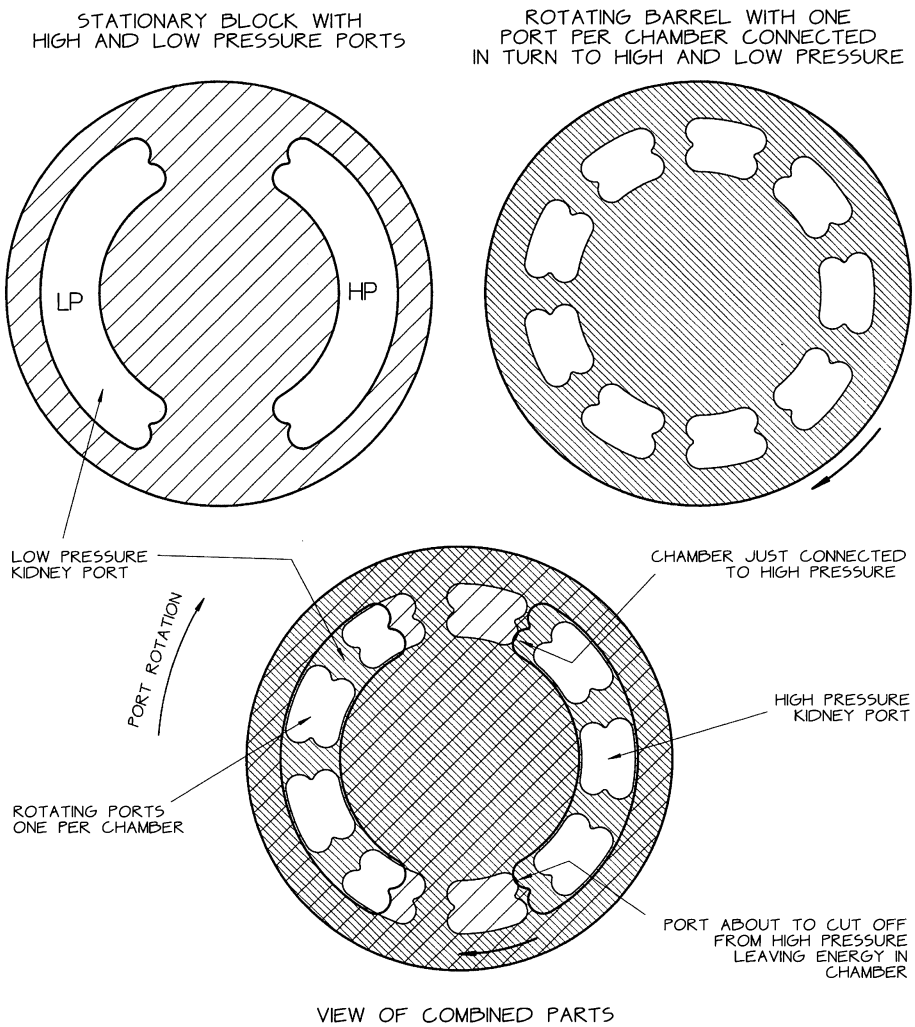

Fig. 8 The port face does the job of a commutator in a d.c. brush motor. It suffers both shear and leakage losses which are hard to balance when large machines are distorted by high pressures. It is by far the most difficult mechanism in the design of bent axis and swash plate machines

connection times in order to minimize the pressure differences at the instant of connection. For example, if the angle of the high-pressure port of a swash plate motor shown in Fig. 8 is chosen so as to disconnect the chamber from the high-pressure gallery at some angle before bottom-dead-centre, then the last part of the work stroke will be using up the stored energy usefully rather than wasting it. However, the required angle is a function of many variables, particularly the angle of swash. It is not possible to do anything about compressibility losses when the swash angle is zero, which is why these machines can be so noisy when they are running with zero displacement.

5.7 Breathing

The velocities of fluid in high-pressure machines have a smaller fraction of kinetic energy than those in low-pressure plant. However, sharp changes of flow direction and passage section will involve changes in kinetic energy which may be difficult to recover. The problem is most acute in the low-pressure passages because there is a strict requirement that the absolute pressure must never approach zero. An estimate of the pressure drop of the inlet passage can be made from the inlet boost pressure specified by a manufacturer. In some designs this is as much as 20 bar for a 350 bar machine, implying low-pressure passage losses of over 5 per cent with as much again if the design of the high-pressure gallery is similar.

6 A NEW DESIGN OF A FAST HYDRAULIC MACHINE FOR WAVE ENERGY

The objectives for a proposed new design are as follows:

1. It should run at synchronous speeds of 1500 r/min (or 1800 r/min for 60 Hz networks).
2. It should be adaptable to operate at power levels well

above the present most powerful designs so that generators of 10 MW can be used and the technology will be suitable for any proposed wave plant.
3. It should be able to combine simultaneous energy flows to and from a number of different components, particularly energy stores.
4. Its efficiency should be as high as possible, particularly at part load.
5. It should have a high tolerance for geometrical distortion by means of self-set clearances and spherical bearings.
6. It should have no heavily loaded, moving, solid-to-solid interfaces.
7. It should have very rapid changes from pumping to motoring to idling, independently for each of the separate flows.
8. It should allow direct control of displacement by computer at the digital databus level.

6.1 Geometry

The requirement to combine energy from multiple sources has led to the decision to use the radial eccentric geometry rather than any of the axial ones usually used for fast machines. The normal oil distribution for radial machines is a long and tortuous path from a valve mechanism at the axis of the machine. It must be placed there to minimize leakage perimeters and shear velocity, but the long distance prevents the use of more than two banks of pistons and imposes such severe breathing restrictions that high boost pressures are needed even for speeds of a few hundred revolutions per minute. The problem can be solved by using poppet valves to connect the chambers of the pump to large galleries at its outer circumference. Figure 9 shows an axial view and Fig. 10 a radial view. In very large machines individual pumping modules could be removed like slices of wedding cake.

The conventional design of single-bank radial-piston machines suffers from both static and dynamic out-of-balance from its eccentric. Because of breathing restrictions, the units are used at such low speeds that this unbalance is unimportant. The proposed design is a multi-bank arrangement with a stack of at least four eccentrics to give static and dynamic balance. A spherical little end and side clearance at the big end mean that frame stiffness and accurate machine frame alignment are not necessary. If a central main bearing is fitted, up to eight layers can be used without over-stressing the shaft in torsion.

The piston-to-eccentric interface has an effective area larger than that of the piston, so that full hydrostatic separation using a feed from the chamber pressure is straightforward. However, unlike normal practice, the big-end pad is offset towards the direction of approach of the eccentric so that, when the piston is not under pressure, the pad can behave like a self-pumped hydrodynamic thrust pad, as shown in Fig. 11. The amount of offset is chosen so that the centre of gravity of the piston and the centre of effort of the hold-down spring are at the optimum point at about 40 per cent forward of the heel of the pad giving the best oil-wedge angle. This means that the off-load pad clearance is much larger and the off-load shear drag much lower than in the normal arrangement. The offset arrangement could not work for the opposite rotation, but it will never be necessary for it to do so.

The pumping chamber extends an idea first proposed by Clerk in reference [**23**]. The hollow piston of Fig. 11 is thinner than its cylinder. It will *expand* under pressure more than the cylinder and so *close* the clearance. This means less leakage than a conventional design on the pressure cycle and less shear on the return stroke. However, its wall thickness is deliberately not constant, so the clearance closure will be less in the middle of the stroke when the piston is moving at maximum velocity. This maintains the correct balance between leakage and shear loss through most of the stroke, both on and off load.

Poppet valves at each chamber going to the high-pressure and low-pressure manifolds are operated by currents in coils, but only at times just before or just after bottom-dead-centre. When a section of the machine is being used as a variable-displacement pump, some of the inlet poppets will be magnetically latched open so that the oil which their pistons have drawn from the inlet manifold is returned there rather than being pumped to the high-pressure manifold. This will reduce the displacement by the delivery of one chamber. The valve flow-path is carefully designed to minimize the energy loss associated with the unused flow to about 1/500 of a working stroke. For larger sizes the use of toroidal valves as shown in Fig. 11 is proposed.

Feeding the oil from the outside rather than the inside of the machine allows extremely generous dimensions for the flow passages. The entire outer surface is a continuous network of axial and circumferential galleries forming the inlet manifold.

Operation as a motor due to Salter and Rampen [**24**] and now being implemented by Caldwell [**25**] is more complicated and is best understood by reference to Fig. 12. The valve timing-sequence for the motor mode starts just after top-dead-centre with oil flowing from the high-pressure manifold into the chamber through the high-pressure poppet. Work is done on the eccentric. Just before bottom-dead-centre the high-pressure poppet is closed by a signal to the coil from the computer. The piston stroke continues using the energy stored by the finite bulk modulus of the oil in the chamber. The chamber pressure decays until, at bottom-dead-centre, it is possible to open the low-pressure poppet valve. The oil in the chamber can now be discharged to the low-pressure gallery. This piston now moves towards top-

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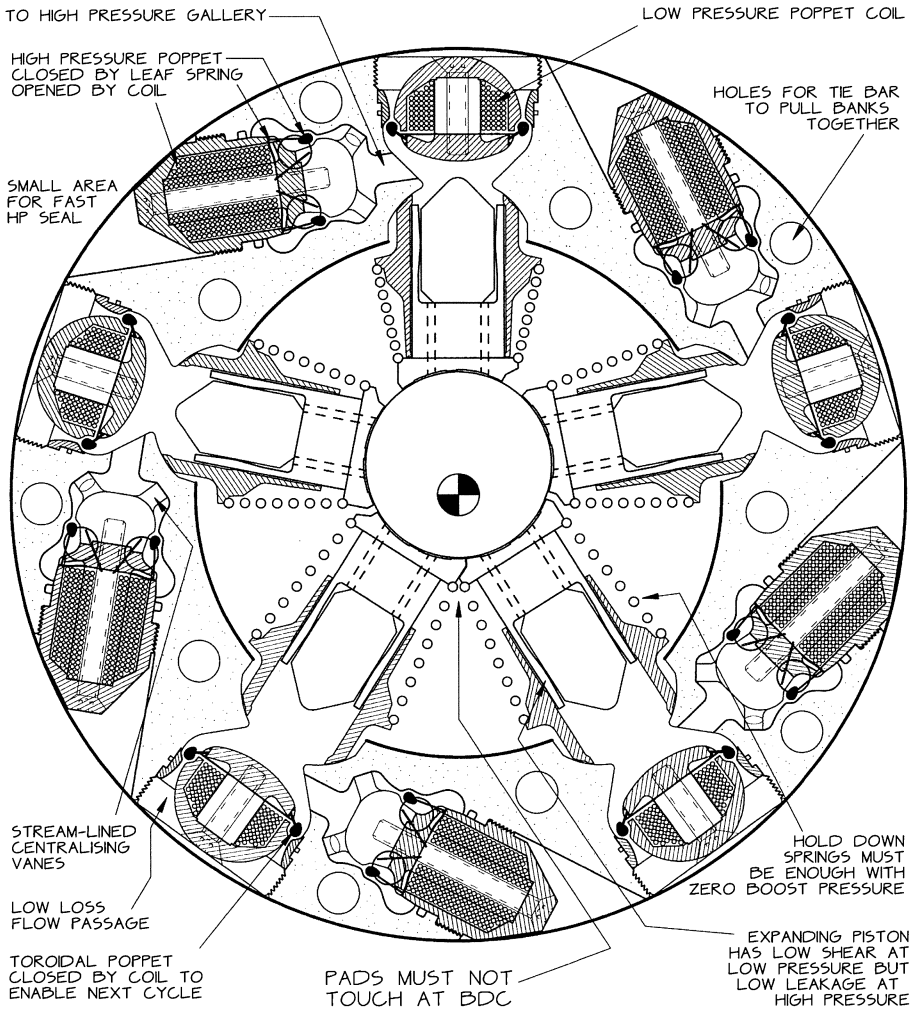

Fig. 9 An axial view of the new design. It is unusual to use radial eccentric geometry with fast machines but this is made possible because flow is controlled by digital poppet valves. The fast-moving high-pressure surfaces are at the big end of each pumping model and are at a much smaller radius than their equivalents at the rim of a portface

dead-centre until, shortly before it reaches the top of the stroke, the inlet poppet is shut. The machine now imitates the action of a diesel engine and begins to compress the oil. Pressure in the chamber rises until the high-pressure valve can be opened so that the sequence can be repeated. The valve timings shown in Fig. 12 are accurate for typical hydraulic oils.

The idea relies on the finite bulk modulus of the working fluid and the fact that computers are very good at precise timing. Some layers of a machine could be pumping to provide reactive energy to a primary element while other layers could, at the same instant, be motoring using energy from an accumulator or idling to minimize shear.

The power flow diagram for using multi-eccentric poppet valve machines with the IPS buoy, equally suitable for the PS Frog, is shown in Fig. 13. It will provide energy storage and any flavour of control strategy.

7 WATER TURBINES

The advantages of using sea water as a power conversion fluid for wave energy are as follows:

1. There is very little danger of a shortage.
2. There is no need for a return pipe.
3. Leakage causes no environmental problems.

Water turbine technology has evolved with very similar design incentives to those of wave energy and is now mature, with an extremely long service life demonstrated in many installations. Sea water has a neutral pH value and is *less* corrosive than rain water in an industrial city.

The disadvantages are that sea water is a complex medium with unpredictable biological constituents. In near-shore installations it may contain abrasive particles that can damage seals and valves. Sediment can clog

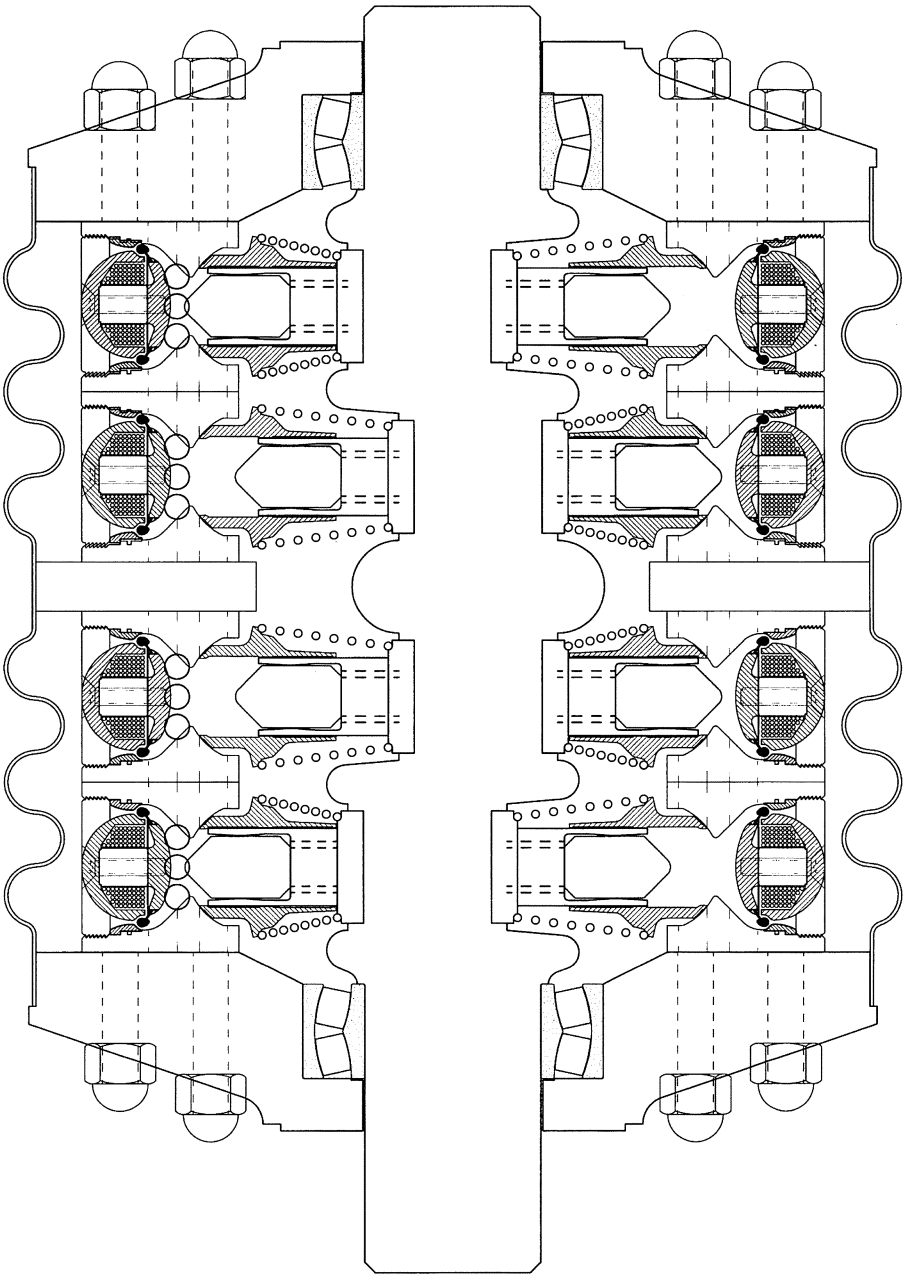

Fig. 10 A radial view of four banks of the new machine design. With a central main bearing it is possible to have as many as ten banks connected to various services. The change from motoring to idling or to pumping takes place in half a revolution of the shaft

ducts during calm spells. Except for very short distances the losses are higher than for electrical cables.

Some offshore wave plant offer a desirable feature which can be quite expensive on land: turbine design is bedevilled by the problem of cavitation. If the absolute water pressure ever falls to zero then vacuum bubbles will be generated. When these collapse the implosion velocities are so high that the hardest steel can be eroded, leaving a surface like the open cells of a slice of bread. The turbines of onshore hydro plants are often buried deep below the exit level so as to maintain a positive pressure. In a deep water wave energy application this pressure could come free.

7.1 Machine types

Water turbine technology is so widely known that it is difficult to make an original contribution. The classes of machine are distinguished by the pressure difference for which they are most suitable. The optimum flow velocity will be proportional to the square root of this pressure.

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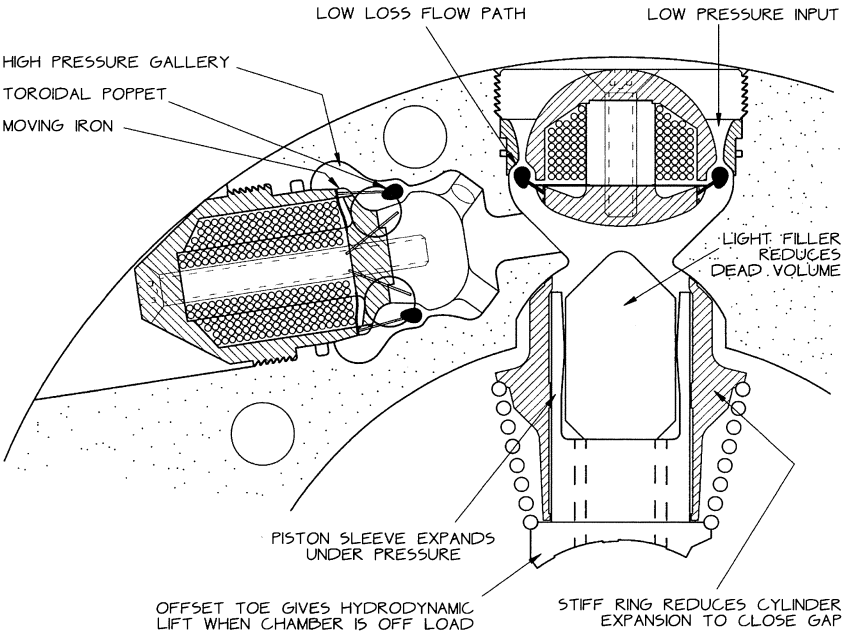

Fig. 11 A closer view of a pumping module

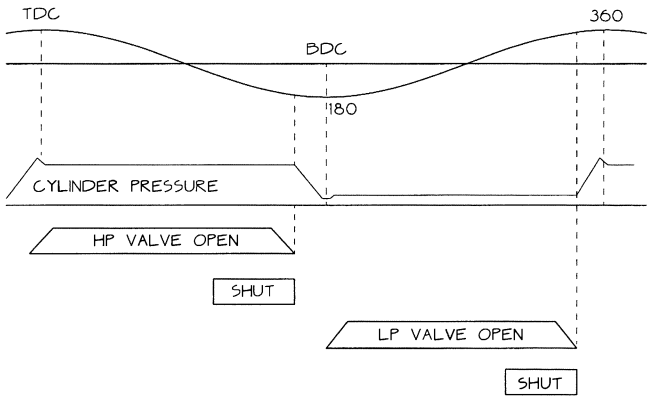

Fig. 12 The valve timing sequence that allows poppet valve machines to act as motors

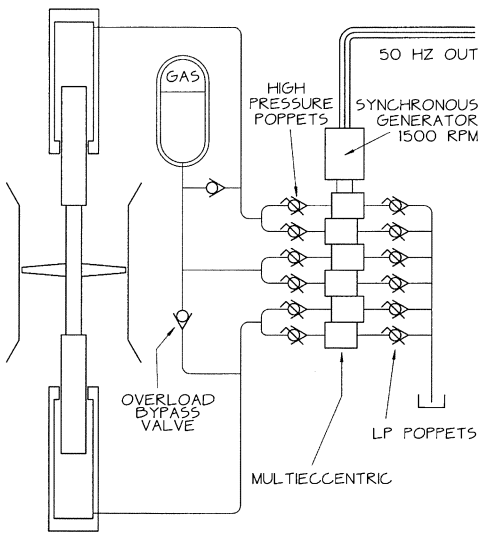

Fig. 13 A multi-eccentric machine controlling two ram chambers and a pressure accumulator for use in the IPS buoy or PS Frog. The shaft of the machine combines torques just like the summing junction of an operational amplifier combines currents

The highest pressures are those used in Pelton wheels in which nozzles direct jets of water at rotating hemispherical buckets on the rim of a disc. The buckets should ideally move at just under half (0.48) of the water velocity, leaving almost no kinetic energy in the exit flow. Pelton wheels would be specified for hose-pump devices. Slightly lower pressures are suited to the Turgo design in which the jets of the Pelton wheel are repositioned so as to hit buckets at a smaller radius on the face of a disc.

Medium pressures are suited to the Francis turbine, where the blade tips move at just less than the flow velocity. Medium-to-low pressures are suited to the cross-flow design. This uses a cylindrical runner rather like a horizontal axis lawnmower. Water is admitted through a rectangular slit and has two passes through the blades. The lowest pressures use propeller-type turbines, such as the Kaplan design in which the blade tips move at about 2.5 times the water velocity. This was used for Tapchan and the Danish Wave Power Buoy design.

In general, cost per kilowatt of turbines falls with rising pressure. The boundaries of the pressure regions are not sharply defined, and at pressures around 10 bar (100 metres head) any of the turbine types could be used if there was some good reason, although the Kaplans would be moving rather fast and would need to be well submerged to prevent cavitation.

All turbine types offer a moderate amount of control

of water flow to allow operation at variable power levels. The Pelton and Turgo use spear valves to change the area of their nozzles. The Francis turbine has a ring of guide vanes with a variable angle. The Kaplan machines can change the pitch of their blades as well as guide vanes. Manufacturers claim a wide range of efficient operation for both Pelton and Turgo designs.

For wave energy devices (with the inevitable exception of Tapchan) it is the control mechanism and the way in which it can accommodate the variability of instantaneous power level that is of chief interest. Wave energy devices require a wider range of control than do land-based hydroelectric schemes. Wave-by-wave changes would need very many more movements of any control mechanism. The power variation problem is made worse if the electrical machine has to rotate at a constant speed—a very likely requirement. New designs of machines for low and variable heads would be welcome for all the over-topping devices and would also find many applications for land-based run-of-river sites.

8 AIR TURBINES

The major attraction of air turbines for wave energy is that the slow velocities of waves (often below 1 m/s) can be painlessly increased to the high velocities (of the order of 100 m/s) regarded as desirable for cutting magnetic flux in an electrical generator. This wonderful result is achieved by the change of the area of an air duct and by the properties of aerofoil sections, which make the tips of turbine blades move even faster than the duct flow.

8.1 The Wells turbine

By far the most popular turbine proposed for wave energy is the Wells design. It has been specified for many oscillating water columns and for the clam air-bag device. The features of the turbine which make it attractive are now very well known, but perhaps a brief summary will be useful.

The drag of the blades when there is no wave-induced air flow is as low as it possibly could be. This means that the idle losses of the machine are below those of any other design. The Wells turbine blades are much cheaper to manufacture than the larger numbers of blades with concave surfaces that are needed for impulse turbines.

The turbine uses symmetrical aerofoils with their chords lying in the plane of rotation so that there is no pitch angle. To understand its behaviour it is necessary to study the velocity triangle of the air flow relative to the blades. An observer riding on the leading edge of a slice of blade would see air approaching which would have two components of velocity. The first would be due to the blade rotation and would be constant for a constant rotation speed. The second would be due to the velocity of the air through the duct. This would be variable and would be up or down according to the phase of the incoming waves. The resultant velocity, and in particular the angle of incidence, usually called α, which it makes with the chord line of the aerofoil, will determine the force on the blade. Aerodynamicists resolve this force into two components.

The first component, which in aircraft design would be called the lift force F_L, is, by definition, at right angles to the incident air flow. There is also a drag component F_D, by definition in the same direction as the incident flow. Elementary aerodynamics allows the calculation of these forces in terms of two coefficients for lift and drag known as C_L and C_D. Values of these coefficients for various aerofoil sections at various angles of incidence and different Reynolds numbers have been measured in wind tunnels. The collection by Abbot and von Doenhoff [**26**] is an excellent data source. Many aerofoils have similar lift coefficients with a linear relationship between C_L and α up to a value between 1 and 1.5 at the stall angle of $\alpha = 15°$. After this, C_L begins to fall with further increases of angle α but rises again for values above 20°. The value of C_L can be increased to 2.5 or more by sucking air from the low-pressure side of the foil and by appendages such as slats and trailing-edge flaps. These enable aircraft to fly more slowly.

Drag coefficients at small angles of incidence are much lower than lift coefficients and are between 0.005 and 0.01 for the symmetrical sections that are often chosen for a Wells turbine. However, at larger angles the drag coefficient rises sharply, reaching 0.4 (half the lift coefficient) at $\alpha = 24°$. Aircraft spend little of their life (or all that remains of it) with stalled air flow. However, the situation is quite different for Wells turbines, because large waves will produce high values of axial air velocity, large values of α, with only moderate lift coefficients and high drag coefficients.

The forces F_L and F_D can be combined by the parallelogram rule to give the resultant force on the blades. An unfortunately small component of this provides the useful thrust that drives the turbine blade to do useful work. The power from a slice of blade would be this thrust times the forward velocity at the radius of the slice.

8.2 Energy accounting

Understanding what happens to all the energy in an engineering system is a valuable exercise. For the Wells turbine the energy paths are as follows.

The air coming past the turbine will still have some kinetic energy. Most can be recovered by slowing the flow with a gentle expansion of the duct.

The torque on the turbine will induce an equal and

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opposite torque on the air sleeve, which will give it a helical flow pattern with some associated kinetic energy of rotation. Part of this can be recovered by the use of guide vanes, but these will have some drag.

If the pressure is not the same along the radial length of the blades (and it will not be for constant chord blades because the outermost tips are moving faster than the parts near the hub), then there will be a tip vortex with more rotational kinetic energy. This effect can be reduced by using a large hub–tip diameter ratio or blades that taper in their plan.

If designers want to avoid undesirable wave loads on high 'chimney stacks' they may bend the air path so as to make the duct outline more compact. Each bend will cause some loss. It helps to bend at large diameters. However, the greatest cause of loss is that the turbines are only momentarily at their most efficient operating point. Figure 14 shows the results of tests by Inoue *et al.* [**27**] in a small reciprocating-flow wind-tunnel with a typical Gaussian input. Efficiency is plotted against a parameter of significant wave height divided by the duct ratio times the wave period times the tip speed, which would, of course, vary with day-to-day sea conditions. The peak efficiency is only 35 per cent. These measurements were made with a small turbine at Reynolds numbers well below those likely at full scale. Bigger turbines will be more efficient. However, there seems to be uncertainty about the size of the improvement. A personal communication from Kaneko predicts a gain of about 10 per cent at the correct Reynolds numbers. This is confirmed by a further personal communication from Ivar Oftebro at Kvaerner Brug, based on theory and the observations from the Norwegian test site at Tofteshallen, where Reynolds numbers were 2.4 to 3.8×10^6 and the peak efficiency was 'slightly lower' than 48 per cent at the best flowrate. Figure 15, from Whittaker [**28**], shows the overall annual efficiency of the best of several machines tested by Queen's University Belfast at the first Islay oscillating water column which had a nominal rating of 70 kW, about one-tenth of the likely power rating of a commercial turbine. The close match of the efficiency curve to the MW h bars of annual pneumatic power shows how well the design has been optimized to suit the column and the wave climate at the site. The mean annual efficiency was 43 per cent. Computer predictions using lift and drag coefficients appropriate for open-channel flow at a set of stations along the blade span suggest a peak efficiency of 55 per cent in an air flow with a Gaussian distribution on a 1.7 m 1500 r/min machine. Results from the fixed-pitch machines being used in the Pico and second Islay oscillating water columns will be very interesting.

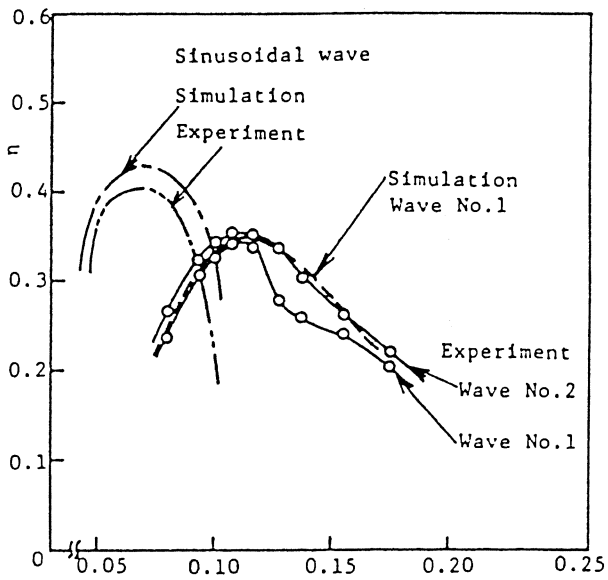

Fig. 14 The efficiency of small Wells turbines as a function of flow in realistic alternating flow patterns measured by Inoue *et al.* [**27**]

Figure 16a shows a set of parallelograms of force for angles of incidence up to 32° using coefficients published by Critzos *et al.* [**29**]. They were measured at a Reynolds number of 1.8×10^6 with standard roughness models. They do not take into account any tip vortex, torque reaction or duct losses. The vertical, snaking curve joins the ends of the resultant force vectors. If this curve lies to the right of the vertical line, work must be fed *into* the rotor to maintain the turbine speed. Repeating the calculation for all the slices of all the blades and dividing by the pressure drop across the turbine (deduced from the axial force) and the volume flowing through the turbine gives total efficiency.

8.3 Possible solutions

Several solutions to the Gaussian efficiency problem are being considered. Even a small improvement in the ratio of lift to drag coefficients will tilt the snake of Fig. 16a to the left and so will have a profound effect on the resultant thrust force. Any method by which this can be achieved must be vigorously pursued. Suction of air from the boundary layer may prove effective.

If inertial considerations permit, it may be possible to increase the rotation speed through the wave cycle so as to reduce the angle α to below the stall value. This would also allow some reactive loading, as discussed in the section on control strategy. It would be possible to fit wave plant with several turbines and bring more on stream in larger seas. It may be possible to improve efficiency by including guide vanes on either side of the turbine to redirect the flow and obtain more favourable values of α. Work in Portugal by Luis Gato at IST is aimed at this.

It would also be possible to vary the pitch of the blades to prevent α from getting too big at high air flows. The blades work very well if α is between 8° and 12°. Things would be much better if the blade tracked any change

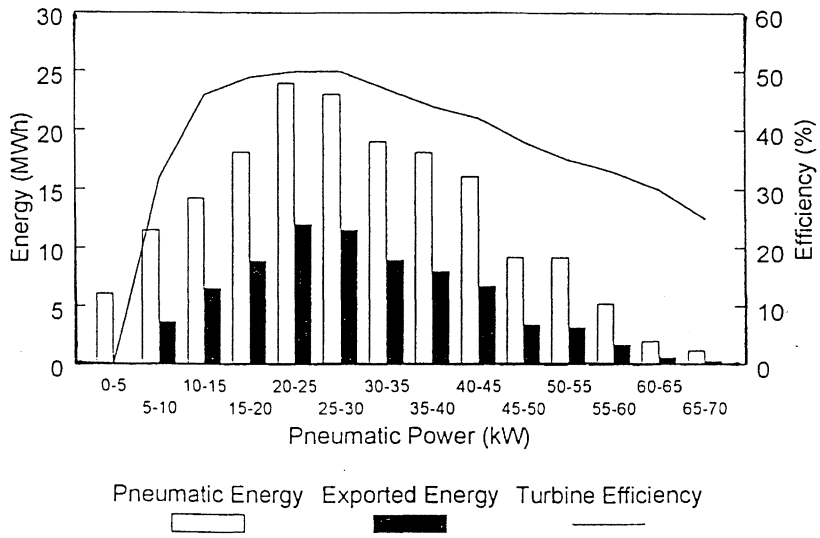

Fig. 15 Annual average conversion efficiency of pneumatic to electrical energy of the best turbine tested in the first Islay oscillating water column measured by Queen's University Belfast [**28**]

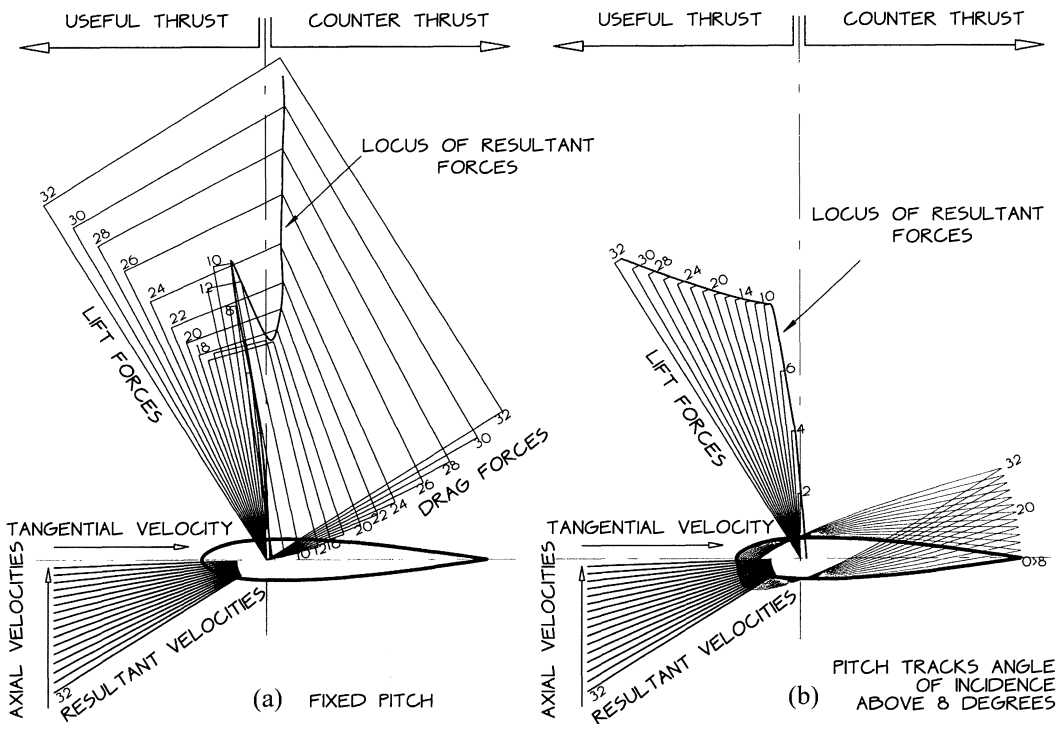

Fig. 16 The lift drag and resultant forces of fixed-pitch and tracking-pitch aerofoils. The increase in drive from the variable-pitch foil is accompanied by a drop in damping coefficient, which may be unwelcome

over 18°. Figure 16b shows the result. The loss of power at high flow is eliminated.

Changing pitch would be more difficult for rotating blades than for the stationary guide vanes, but the technology for doing so has been in operation since the 1930s for aircraft propellers at similar speeds and powers. The Messerschmitt bf 109 had pitch control as original equipment, but its use by the Royal Air Force was fiercely resisted on grounds of complexity by officials of the British Air Ministry until an unauthorized test by de Havilland in June 1940, which gave a Spitfire an extra 7000 feet of altitude. All the fighters of the Royal Air

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Force were converted during July and August, not perhaps the most convenient moment for such a major modification.

9 OUTLINE REQUIREMENTS FOR A VARIABLE-PITCH TURBINE

A consideration of practical values reveals the following requirements:

1. The need to make a pitch change for a large fraction of the waves—one or two million operations a year—far more than the few per flight of an aircraft.
2. The need to resist the very large centrifugal forces. At 1500 r/min and a diameter of 1.7 m, each kilogram of mass weighs over two tonnes.
3. The need not to obstruct the air flow with any protrusions at the blade root.
4. The need to present the right damping coefficient to the sea but not to suffer damage to the blades because of droplet erosion.
5. The need to minimize rotating inertia if wave-by-wave speed variation is desirable.

It is comforting to know that the value of the potential productivity gain multiplied by the entire upstream capital cost is so large that money is almost no object—an unusual feature in renewable energy work. The design, construction and testing of a 1.7 m 500 kW variable-pitch turbine intended for the Pico oscillating water column in the Azores has been described by Taylor and Caldwell [**30**]. Views of the internal mechanism are shown in Figs 17 to 20.

Pitching power was taken from the shaft rotation using two eddy current discs which act as brakes when magnetic flux passes through the disc thickness. One disc

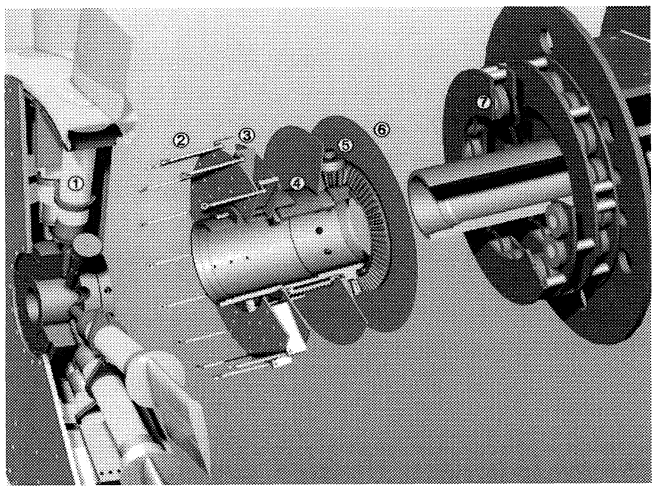

Fig. 17 A solid model of the pitch-change mechanism showing ball-screw and eddy-current discs. All the actuator power is taken from the shaft rotation with the lowest possible idling losses

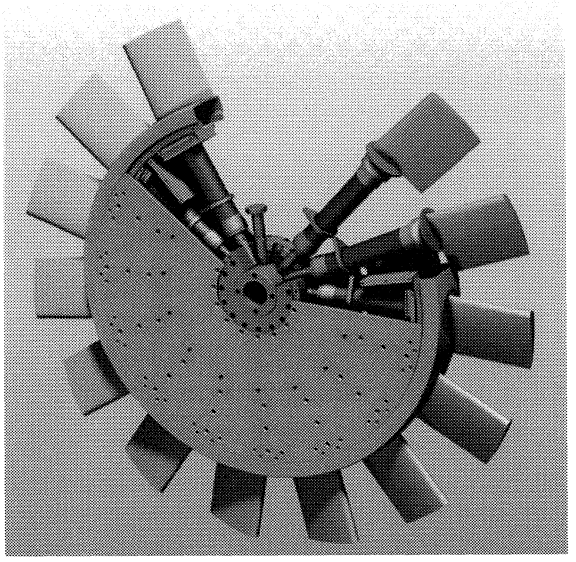

Fig. 18 A solid model of the turbine from the front showing a bearing stalk. This gives room for conventional roller thrust bearings. A tension–torsion strap can also be used

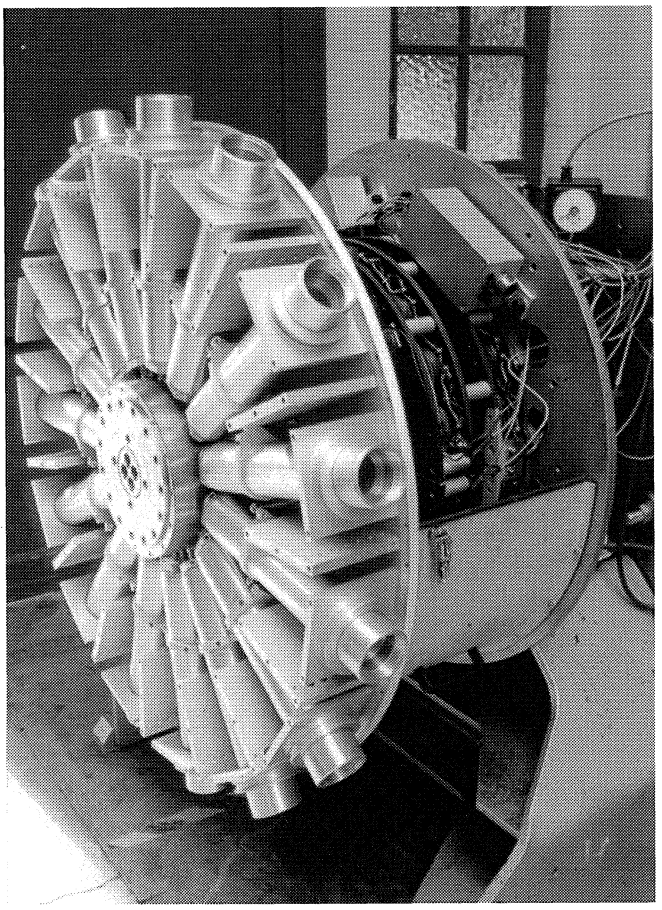

Fig. 19 The turbine assembled for spin and pitch testing with blades replaced by equivalent mass radius tubes. Spin testing with blades would have risked sucking people into the machine

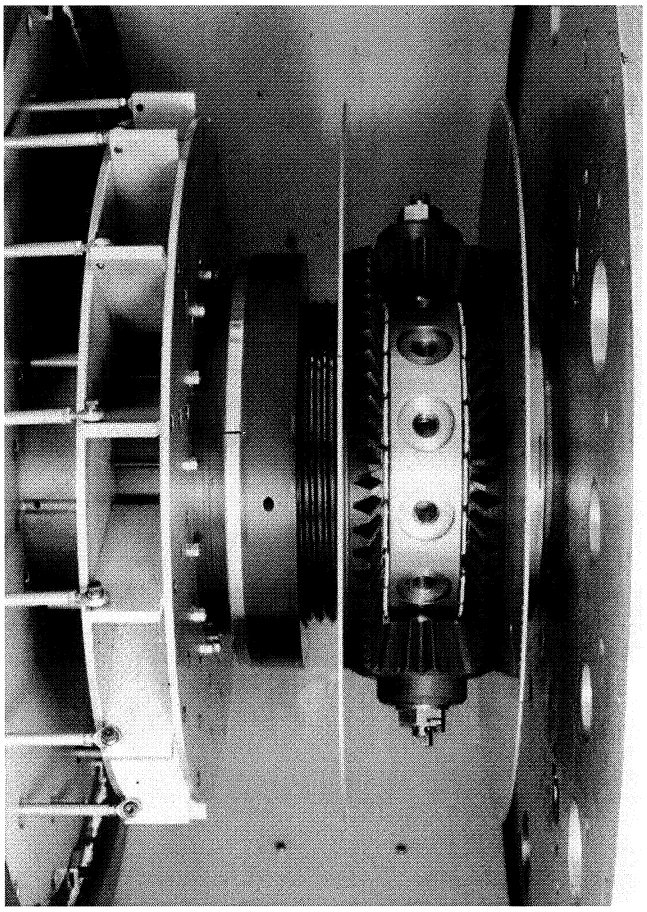

Fig. 20 The ball screw and differential gear system driving the pitching rods

was coupled directly to a tubular ball screw surrounding the shaft. The other was coupled via a crown wheel and pinion to rotate the ball screw in the opposite direction. The ball nut drove a pitching horn on each blade.

The amount of room available for the roller bearings which take the centrifugal loads was increased by mounting them on stalks protruding from the hub. By taking the force line back towards the hub and having wells of grease outboard of the bearings, it is possible to keep the bearings lubricated despite the centrifugal loads. The design is compatible with a future tension–torsion band, as used on some helicopter rotors, which would replace the rolling elements and extend the replacement life.

There is concern about using mechanisms like ball screws in the salt-spray environment that is likely to occur in an oscillating water column. However, while the machine is spinning there will be a controlled flow of warm dry air from the generator compartment flowing out of the turbine hub and any droplets of spray will have to flow through a double labyrinth passage against this air flow and also against a centrifugal gradient of 1500 *g*. The main danger will come when the machine is not spinning, but then the isolating gate of the turbine hall can be kept shut.

The blade profile varies from root to tip and was optimized by Luis Gato taking the tip Mach number into account. A very lightweight design has been achieved using a carbon fibre construction with different thicknesses of carbon according to the local stress. The mould tools for the blade spare and the cores have been machined. The main core will be cast in lightweight polyurethane foam. Its surface has depressed regions to allow different numbers of layers of carbon cloth to be wrapped like 'bandages'. The bundle will then be assembled into the outer mould and injected with thermo-cured epoxy resin.

Unprotected carbon fibre may not have a very good life expectancy because of water droplet erosion. If unacceptable erosion occurs and the gains in efficiency of variable pitch justify it, then carbon-fibre epoxy can be clad with a layer of electroless nickel or one of the polyurethane coverings used for hovercraft blades.

9.1 Turbine tests

Reports from the air turbine pioneers all confirm the extreme difficulty of work when the sea is in control of the test conditions, and access to the equipment under test is restricted when conditions are particularly interesting. A full-power reciprocating-flow wind-tunnel will be necessary if all the aerodynamics are to be fully understood, if the hardware of guide vanes and variable-pitch blades is to be made efficient and reliable, and if the required rate of experimentation is to be achieved in safety. Figure 21 shows a sketch arrangement of the design. It is intended to give accurate absolute calibration, easy variation of conditions over a wide range, rapid access to turbines, ducting and instruments, and much faster experiments on many different machines and ducts.

The traversing of Pitot tubes across a flow duct to measure flowrates will be avoided. It will be possible to cover the conditions of a whole year in a few hours and then turn off the air flow in a few seconds. This might take six months with a natural wave input. In a well-chosen wave site there is the risk of long periods in which it is too dangerous to approach the unit to attend to the simplest repair.

The civil cost of the reciprocating flow tunnel should be less than for an oscillating water column of similar power rating because there are no extreme wave loads, because the tunnel can be built at a convenient site and because a power cycle of only two or three seconds will give all the necessary information so that the overall volume can be smaller than that of an oscillating water column. The operating costs will be lower because of convenient transport and accommodation. The versatility will be better because there is easy access for

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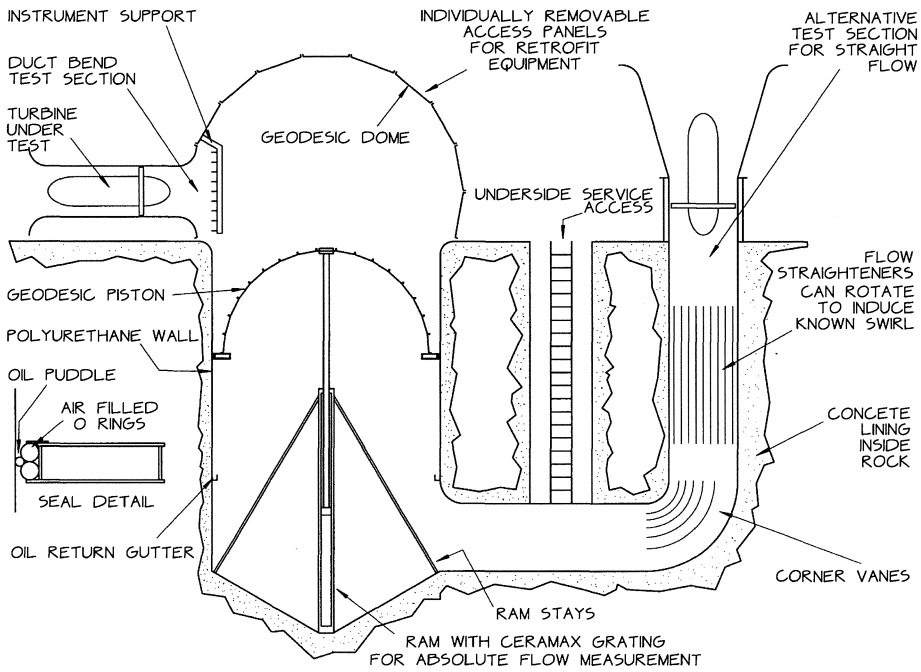

Fig. 21 A sketch of the reciprocating-flow wind-tunnel

cameras, flow-tracing instruments, spray-injectors and unforeseen equipment.

Figure 22 shows the drive layout intended to recirculate the energy so as to minimize running costs. The electricity grid would have to supply only the mean value of the system losses. The capital cost would be reduced and the running costs slightly increased if the input induction motor were to be replaced by a diesel engine. The electrical return path could also include any d.c.-to-inverter equipment width needed for non-synchronous turbine designs. With double-ended electrical machines and some attention to the diesel speed control extra input power could be raised for occasional tests using a combination of electrical and diesel power. The wind-tunnel drive is another example of the flexibility of the multi-eccentric wedding cake machine. Suggestions from turbine designers for the best power rating of a reciprocating-flow wind-tunnel will be appreciated.

10 CONCLUSIONS

1. The correct power take-off strategy can improve the power capture of wave plant from the 12–15 per cent claimed by a vertically moving IPS buoy (which has particularly attractive overload protection) to over 200 per cent referred to the device width for a pseudo complex-conjugate-controlled, asymmetric, point absorber.
2. Energy storage placed close to the front of the powertrain will reduce downstream capital costs and standing losses as well as making the output more acceptable to slender island networks.
3. The cost of gas accumulators must be substantially reduced and their safety requirements never forgotten, even if the safety factor can be reduced for an unmanned plant.
4. Devices with a leg into deep water, such as the mace, can use oil accumulators with an attractive, near-constant pressure rate and slightly safer behaviour than gas-filled pressure vessels.
5. New developments with the Regenesis chemical storage method may have a large effect on waves as well as other renewables.
6. There is considerable dispute about air turbine performance in the Gaussian air flows caused by sea waves. There are many ways in which performance might be improved. A combination of variable blade pitch and variable rotor speed would extend the range of efficient operation, with correct damping and a moderate amount of reactive loading.
7. The inertia of an electrical machine is too large for wave-by-wave speed variation of a turbine, but this might just be possible with suitable hydraulic pump/motors.
8. Linear hydraulic rams will be satisfactory for power take-off inputs, particularly if large Belofram seals are combined with Ceramax coatings. The main concern lies in choosing the range of stroke and swivel angle with some lesser concern about seal velocity.
9. Rotating inputs, either directly or via rope and winch, will resolve range problems but large-diameter winches and multiple cables will be needed to reduce bending fatigue.

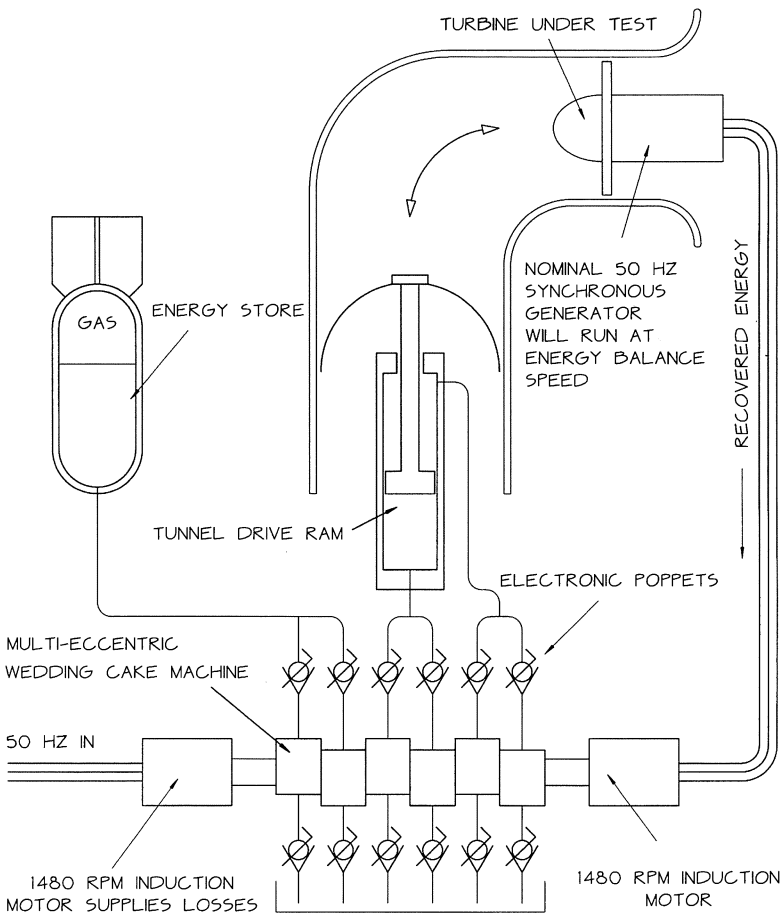

Fig. 22 The power flow diagram of the reciprocating-flow wind-tunnel with power from the turbine fed back to the shaft of a double-ended multi-eccentric machine. The inclusion of the pressure accumulator means that only the losses are drawn from the grid. Such a test facility would allow a confidence-building burn-in of the entire power chain, including storage, electronic frequency conversion circuitry and all controls

10. Ring-cam pump technology with large numbers of pumping modules and cam lobes can provide sufficient torque and tension.
11. Direct electrical generation using transverse flux machines, previously thought to be too heavy and expensive, may now be possible because of new magnetic materials and the reduced costs of frequency-changing electronics. Careful thought will have to be given to maintaining fine air gaps despite strains in the mounting structure.
12. Designers of conventional hydraulic machines for land use say that they have no incentive to improve the efficiency of their plant. Wave energy provides very strong incentives to do so because the value of much of the preceding equipment is multiplied by efficiency gains. Variable-displacement, energy-summing and low part-load losses are desirable.
13. Theoretical analysis—still requiring experimental proof—suggests that both slow and fast hydraulic pumps and motors with direct computer control of displacement using poppet valves can be produced with losses less than those of an electrical generator and weights below those of comparable gears. Many of the uncertainties can be removed with the laboratory tests of small submodules now under way.
14. Achieving reliability and optimum performance of air turbines will need a land-based reciprocating-flow wind-tunnel with full power rating. This should cost less than a single salt water demonstration and would be used for many different turbines. It would provide absolute calibration of the input, easy access for instrumentation and repeatable test conditions over the entire range of operation including rare extremes.

ACKNOWLEDGEMENTS

Wave energy research at Edinburgh University has been supported by the UK Department of Energy (now Department of Trade and Industry) and also by the European Commission JOULE programme. Commercial exploitation of poppet valve hydraulics is being undertaken by Artemis Intelligent Power. Richard

Yemm, founder of Ocean Power Delivery, built the very fast brake for the Azores stop valve. Ross Henderson, now also with OPD, drew the solid models of the turbine during his undergraduate studies.

REFERENCES

1 **Count, B. M.** On the hydrodynamic characteristics of wave energy absorption. In Second International Symposium on *Wave Energy Utilisation*, Trondheim, June 1982, pp. 155–174 (Tapir, Trondheim).

2 **Budal, K.** and **Falnes, J.** The Norwegian wave-power buoy project. In Second International Symposium on *Wave Energy Utilisation*, Trondheim, June 1992, pp. 323–344 (Tapir, Trondheim).

3 **Salter, S. H.** and **Taylor, J. R. M.** Tl gh-speed stop valve for oscillating water- ›nd European Wave Power Conference, Lisbon, November 1995, 1996, pp. 337–344 (CEC, Luxembourg).

4 **Caldwell, N. J.** and **Taylor, J. R. M.** Design and construction of elastomeric parts for the Azores oscillating water column. In Third European Wave Conference, Patras, October 1998, pp. 318–323.

5 **Henriques, J. C. C.** and **Gato, L. M. C.** Adaptive control of the high-speed stop valve of the Azores plant. In Fourth European Wave Energy Conference, Aalborg, December 2000, paper J6.

6 **Justino, P. A. P.** and **Falcao, A. F. de O.** Active relief valve for an OWC wave energy device. In Fourth European Wave Energy Conference, Aalborg, December 2000, paper J2.

7 **Nebel, P.** Maximizing the efficiency of wave energy plant using complex conjugate control. *Proc. Instn Mech. Engrs, Part I, Journal of Systems and Control Engineering*, 1992, **206**(I4), 225–236.

8 **Nebel, P.** PhD thesis, Edinburgh University, 1994.

9 **Belmont, M. R., Morris, E. L., Horwood, J. M. K.** and **Thurley, R. W. T.** Deterministic wave prediction linked to wave energy absorbers. In Third European Wave Conference, Patras, October 1998, pp. 153–161.

10 **Evans, D. V.** A theory of wave absorption by oscillating bodies. *J. Fluid Mechanics*, 1976, **77**, 1–25.

11 **Salter, S. H.** and **Lin, C.** Wide tank efficiency measurements on a model of the sloped IPS buoy. In Third European Wave Conference, Patras, October 1998, pp. 200–206.

12 **French, M.** and **Bracewell, R. H.** PS Frog: a point absorber wave energy converter working in pitch/surge mode. In International Conference on *Energy Options*, University of Reading, 1987.

13 **Salter, S. H.** Changes to the 1981 reference design of spine-based ducks. Report to the UK Department of Trade and Industry. June 1992; reprinted as *Renewable Energy Clean Power 2000*, IEE Conference, 17–19 November 1993, pp. 121–130 (IEE, London).

14 **Yemm, R. W., Henderson, R. M.** and **Taylor, C. A. E.** The OPD Pelamis wave energy converter. Current status and onward programme. In Fourth European Wave Energy Conference, Aalborg, December 2000, paper E3.

15 **van Dijk, J.** Hydraulics with a little extra. In *European Oil-Hydraulics and Pneumatics*, February 1992, pp. 31–34; also technical information from Hydraudyne, http://www.hydraudyne.nl/engels/hydraudyne/ceramax_cims.html.

16 **McConnell, J.** Comparative fatigue tests on 3-core 22 kV A.C. submarine cable for floating wave energy converters. Pirelli Report 8526, WESC (80) GT 125 (from UK ETSU).

17 **Roark, R. J.** and **Young, W. C.** *Formulae for Stress and Strain*, 1989, Ch. 13 (McGraw-Hill, New York).

18 **Mueller, M. A., Baker, N. J.** and **Spooner, E.** Electrical aspects of direct drive wave energy converters. In Fourth European Wave Energy Conference, Aalborg, December 2000, paper H4.

19 **Roark, R. J.** and **Young, W. C.** *Formulae for Stress and Strain*, 1989, Article 15.2 (McGraw-Hill, New York).

20 **Clerk, R. C.** The prestressed laminated flywheel and its hydrovac ambience. In Flywheel Technology Symposium, San Francisco, California, 1977, CONF 77, 1053, pp. 167–180.

21 **Salter, S. H.** and **O'Dwyer, D.** Analysis of the reduction of fragment velocity from shell bursts achieved by water bags. *J. Inst. Explosives Engrs*, September 1999, 5.

22 Commercial literature from Innogy plc Harwell. From regenesys@innogy.com.

23 **Salter, S. H., Rea, M.** and **Clerk, R. C.** The evolution of the Clerk tri-link machines. In Eighth Symposium on *Fluid Power*, Birmingham, 1988, pp. 611–632 (BHRA, Bedford).

24 **Salter, S. H.** and **Rampen, W. H. S.** The wedding cake multi-eccentric radial piston hydraulic machine. In Tenth International Conference on *Fluid Power*, Brugge, April 1993, pp. 47–64 (Mechanical Engineering Publications, London).

25 **Caldwell, N. J.** PhD thesis, Edinburgh University (to appear).

26 **Abbot, I. H.** and **von Doenhoff, A. E.** *Theory of Wing Sections*, 1949 (McGraw-Hill, New York); reprinted as a student edition (Dover, London).

27 **Inoue, M., Kaneko, K., Setoguchi, T.** and **Saruwatari, T.** Studies on the Wells turbine for wave power generator. *Int. J. Japan Soc. Mech. Engrs, Ser. II*, 1988, **31**(4).

28 **Whittaker, T.** Operation of the Islay Shoreline Wavepower Plant as marine test bed for turbine generators. Project Phase 5, ETSU V/02/00171/REP.

29 **Critzos, C. C., Heyson, H. H.** and **Boswinkle, P. W.** Aerodynamic characteristics of NACA 0012 airfoil section at angles of attack from 0 to 180. NACA TN 3361, 1955.

30 **Taylor, J. R. M.** and **Caldwell, N. J.** Design and construction of the variable-pitch air turbine for the Azores wave energy plant. In Third European Wave Conference, Patras, October 1998, pp. 328–337.