Budal's latching-controlled-buoy type wave-power plant

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ABSTRACT

Results are presented from previous investigations of three different versions of latching-controlled, axisymmetric, wave-power buoys, intended to operate in the heave mode, and to force-react against anchors on the sea bed. One of them had hydraulic machinery for control and power take-off, while the two others had a latching mechanism for control and pneumatic power take-off. One of the three versions utilised an oscillating water column contained in a heaving structure. Hulls of various shapes were examined: cylindrical with hemispherical base, conical with the widest part up, and, finally, spherical. Model tests were performed that indicate reasonable agreement with theory. Technical assessments of full-scale power buoys indicated the necessity for further development work on some of the components. Economic assessments showed that these power buoys, in their present state of development, could not yet compete commercially with hydroelectric power plants in Norway.

1. INTRODUCTION

During the late 1970s and early 1980s Norway had a substantial wave-energy research programme, which was financially supported by Olje- og energidepartementet (OED – the Royal Ministry of Petroleum and Energy) [1,2]. Three significantly different concepts were assessed, technically as well as economically. Two of the concepts were the TAPCHAN and the Multi-resonant OWC, pursued by the two companies Norwave AS and Kværner Brug AS, respectively. Full-scale prototypes of both devices were built in 1985 and tested during some years to follow. For this reason these concepts are probably more well-known than the third concept, the phase-controlled point absorber, which is to be considered in more detail in the present paper.

After his initiative in 1973 to start research at Institutt for eksperimentalfysikk, NTH, University of Trondheim (now NTNU), on conversion of wave energy, Kjell Budal (1933-1989) put forward many proposals of devices. Most of them were heaving buoys, small enough to be considered as point absorbers, which means that their linear dimensions are much smaller than prevailing wavelengths [3]. Already in 1974 he proposed to apply control equipment, such as "a combined motor and generator", to maximise power take-off in irregular waves [4]. Such an optimisation method was also proposed independently by Salter [5]. Two years later Budal proposed the simpler method of applying latching control [6]. For optimum control with hydraulic machinery a combined pump and motor [7,8], may be applied instead of a combined electric motor and generator. Latching control may be applied by means of a mechanical latching device, or, with hydraulic machinery, by means of a controllable valve.

During five funding-rich years, starting in 1978, three different versions of latching-control, heaving-buoy, point-absorber WECs were model-tested and assessed in full scale. We refer to the three versions as "type E", "type M2" and "type N2". Several internal and technical reports, mostly in the Norwegian language, contain information from this work, but only minor parts of this information have been widely published. It is the purpose of this paper, firstly, to make additional information available to the international wave-energy community, and, secondly, to advocate ideas and concepts put forward by the creative person Kjell Budal.

2. "SMALL IS BEAUTIFUL"

If ocean waves always had had the same amplitude and the same wave period, it would have been rather easy to develop a feasible wave-energy converter (WEC). It is much more difficult with the stochastic nature of real ocean waves, for which there is a large variation in wave periods and an even more serious variation in wave heights. The latter problem, in particular, was seriously considered by Budal.

Optimum control of the amplitude may be an even more challenging problem than the phase-control problem. Since the target is to maximise the converted power relative to the cost for its practical realisation (rather than relative to the wave energy available in the sea), Budal advocated that WEC units should be relatively small [9,10]. Point absorber diameters should preferably be in the range of five to ten percent of prevailing wavelengths.

The following features are common to the three versions (types E, M2 and N2). The heaving point absorbers are force-reacting against an anchor on the sea bed. They are dimensioned (designed) to operate at full design amplitude most of the time. Thus only in rather rare low-wave situations the famous point-absorber maximum capture width of $l/2p$ should be approached. The power take-off machinery is designed to allow for a variation as large as possible in the effective (very non-linear) load resistance that is needed when wave heights vary, while the heave amplitude is kept closely at its design-specified value. Load damping of the heave oscillation occurs only when the heave position approaches its lower or upper extreme, namely when an appropriate check valve opens and makes connection to either a high-pressure reservoir or a low-pressure one, respectively. Cf. figure 1. For all three versions the latching device serves phase control, and, except for low-wave situations, also amplitude control (load control) [11]. Because of non-causality problems involved, it is necessary to apply measuring gauges and electronic software in order to predict the instantaneous values of the incoming wave and of the heave motion a few seconds into the future [9,12,13]. The latching device could also be operated for the purpose of reducing the oftenness of activation of end-stop devices [11]. The buoy should remain latched during extreme weather.

Three different geometrical shapes of the axisymmetric buoy were considered. Type E was a cylinder with hemispherical lower end [14]. Type M2 was conical with the broader part up [12]. Both of these buoys contained a cylinder with corresponding piston that was connected to the sea-bed anchor. Type N2 was shaped as a sphere that was open in the lower end. Instead of using a piston pump, the buoy entrapped an OWC, and the buoy was sliding up and down along a rod that was connected to a universal joint on the sea-bed anchor [13,15]. With type E, the connection to the sea-bed anchor was by means of a pre-tensioned steel cable. Because of the possibility of loosing pretension in extremely deep wave troughs, it was decided, with types M2 and N2, to use steel rod, which is able to take pressure forces too, and not only tensile forces as with a cable. The need for end-stop facilities is less strict with type N2 than with types E and M2. Most extensive design work was carried out on the N2-type power buoy.

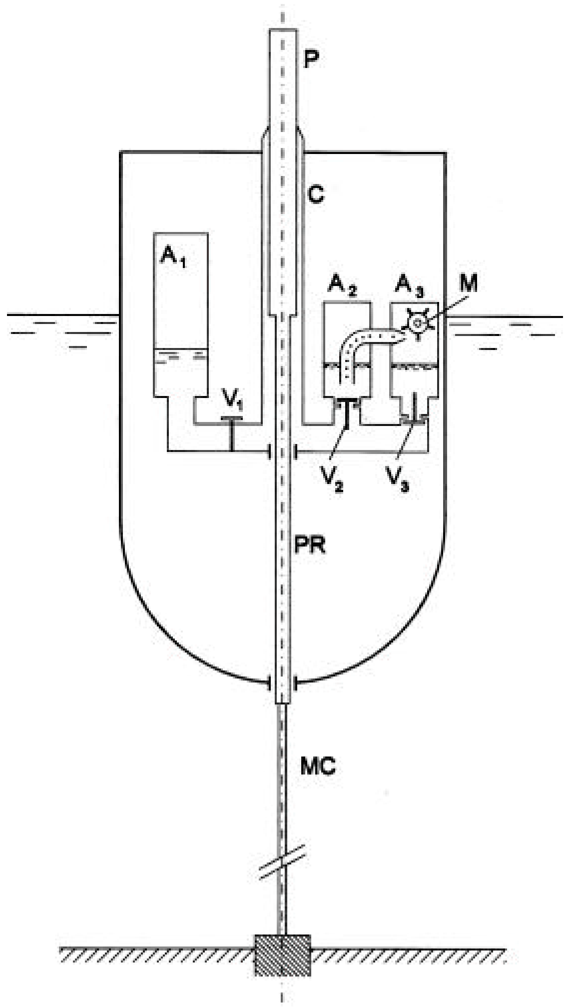

Figure 1. *Point absorber of "type E". The machinery consists of a hydraulic cylinder C, three gas accumulators A_1 , A_2 , A_3 , and three valves V_1 , V_2 , and V_3 . The piston P, with piston rod PR, is connected to a mooring cable MC, pre-tensioned by the pressure in accumulator A_1. Latching/ unlatching is obtained by closing/opening valve V_1. The check valves V_2 and V_3 serve amplitude control and power take-off. Hydraulic fluid is discharged at a relatively steady rate from high-pressure accumulator A_2 to low-pressure accumulator A_3 through a hydraulic motor or Pelton turbine M.*

The power take-off was hydraulic with type E, and pneumatic with types M2 and N2. It was envisaged that a group of ten type-E buoys were pumping pressurised fluid to a common, stationary located, hydraulic motor or turbine [14]. Each buoy of type M2 or N2 was envisaged to have rectifying air valves and an air turbine with electric generator onboard [13,16,17]. The latching device was a controllable valve in the hydraulic system of type E [11,14]. It was a controllable friction mechanism with types M2 and N2. For type N2 this friction mechanism was of a particular design, proposed by Budal, for the purpose of securing automatic latch action when the buoy, in its heave oscillation, had just reached extreme upper or lower position[17,18].

Models of all three versions were tested (in 6-to-10 times reduced scale) in the big towing-tank wave channel in Trondheim [12,13,14,15,19]. The type-N2 model was tested also in the sea near Trondheim [18,20]. Full-scale prototypes were assessed technically as well as economically [1,13,14,16,17,21].

3. POWER BUOY OF TYPE E

A buoy of type E is sketched in figure 1. The hull is shaped as a cylinder with hemispherical bottom. The mass, including mass of machinery and ballast, is less than the mass of water displaced by the hemispherical part. The equilibrium position indicated in figure 1 is obtained through the hydraulic system by means of the pressure in gas accumulator A_1 , provided valve V_1 is open. When this valve is closed, the buoy is latched, unless the fluid pressure in the cylinder is so high that valve V_2 is open or so low that valve V_3 is open. The intended vertical stroke length of the buoy should not exceed the height of the cylindrical part of the buoy.

In 1978 a reduced-scale buoy of diameter $2a$ = 1.1 m and cylindrical height $2l$ = 1.4 m was constructed and tested (at Skipsmodelltanken, NTH) in a wave tank of width d = 10.5 m and water depth h = 5.8 m. In this experiment the largest uncertainty (relative error at least ± 10 %) is to determine the incident wave amplitude A. In comparison, other experimental errors are negligible. For the experimental results presented below, with wave period $T = 2p/w$ = 3.1 s and wavelength $l = 2p/k$ = 14.8 m, we may assume that the amplitude A of the incident wave is in the range of 0.08 to 0.10 m. In figure 2, the upper graph shows the wave, as measured one wavelength "upstream" in the wave channel. The lower graph, indicating the measured heave position s of the buoy, demonstrates the dramatic influence of latching control. Balance between power input and power loss (including friction and intentional load) is obtained when the heave stroke is about five times the wave height. The average input power P_b from the waves to the buoy is obtained as the average slope P_b = 230 W of the graph of input energy E_b shown in the lower graph of figure 3. The increment in input energy between instants $t1$ and $t2$ is obtained as

$$\Delta E_b = \int_{s(t1)}^{s(t2)} F_b(t)\, ds(t) \qquad (1)$$

where $F_b(t)$ is the force that is measured in the mooring cable, and $s(t)$ is the measured position of the buoy as it moves up or down along the piston rod. The curve is much more smooth in the lower graph of figure 3 than in the upper one. A reason for this is that the heave motion is rather modest (or even latched) when the dynamic mooring force F_b is large, or rapidly varying.

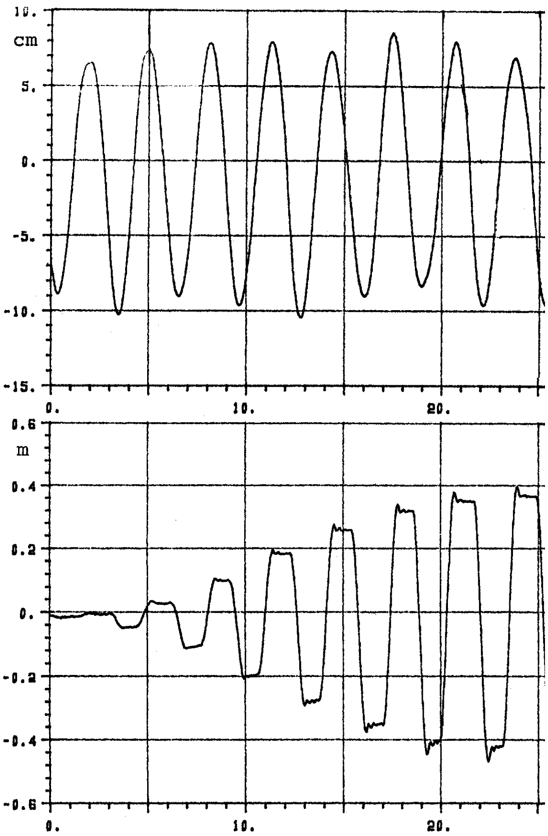

Figure 2. *Building-up of latching-controlled buoy's heave oscillation to a stroke length of 0.8 m in wave of height 0.16 m and period 3.1 s* [14].

For a sinusoidal incident wave, the input power P_b equals the excitation power P_e less the radiated power P_{r1} from the first harmonic heave oscillation less power P_{rS} radiated in higher harmonics less power P_v lost due to viscosity [12],

$$P_b = \tfrac{1}{2} f_{e1}A\; u_1 \cos b - \tfrac{1}{2} R_{r1}\, u_1^2 - P_{r\Sigma} - P_v \qquad (2)$$

where $f_{e1}A$ and u_1 are the amplitudes of the wave excitation force and of the first harmonic component of the heave velocity, respectively, and *ß* is the phase angle between them. (A phase-control target is to obtain *ß* = 0). Moreover, f_{e1} and R_{r1} are the absolute value (modulus) of the heave-excitation-force coefficient and the radiation resistance, respectively, at the fundamental frequency $w/2p$. For this symmetric deep-water case, they are related through the reciprocity relation [9, eqs.(2.5-6)]

$$R_{r1} = (w\, /\, rg^2 d)\, f_{e1}^2 \qquad (3)$$

A generalised version of this relation may be found elsewhere [22, eqs.(33&94)]. Assuming as a reasonable approximation [9] that, f_{e1} is proportional to $\exp\{-w^2/g)l\}$, known numerical results [23] for a heaving semi-submerged sphere may be used to find theoretical values, f_{e1} = 5.25 kN/m and R_{r1} = 55 Ns/m. An experiment with the buoy latched in its equilibrium position was carried out in order to measure the heave excitation force. Experimental values for f_{e1} in the range of 5.1 to 5.2 kN/m were found, well in agreement with the theoretical estimate. With the present curve shape of the heave motion (see figure 2, lower graph), the first harmonic heave amplitude $s_1 = u_1/w$ is approximately 20% larger than the maximum heave excursion s_m = 0.4 m. Thus s_1 = 0.48 m and u_1 = 0.97 m/s. If we now set A = 0.08 m, and $ß$ = 0, the two first terms in eq. (2) give $P_e - P_r$ = (204-26) W = 178 W. That this value deviates so much from the experimental value 230 W for P_b may be attributed to the inaccurate determination of the incident wave amplitude A. However, if we set A = 0.10 m, we get better agreement, with $P_e - P_r$ = (255-26) W = 229 W. It appears that the two last terms in eq. (2) are of minor importance. The wave power level is J = 123 W/m, and the absorption width is P_b/J = 1.86 m, which is 1.7 units of the buoy diameter. In comparison, the theoretically maximum absorption width with unrestricted heave amplitude would be $d/2$ = 5.25 m in the wave tank (and $l/2p$ = 2.35 m in an open-sea case). The volume of the buoy is $V = pa^2(2a/3 + 2l)$ = 1.68 m^3, and relative to this the input power is P_b/V = 137 W/m^3, which is 28% of the value 497 W/m^3, given by Budal's upper-bound relation [9,10] $P/V < prgA/(2T)$.

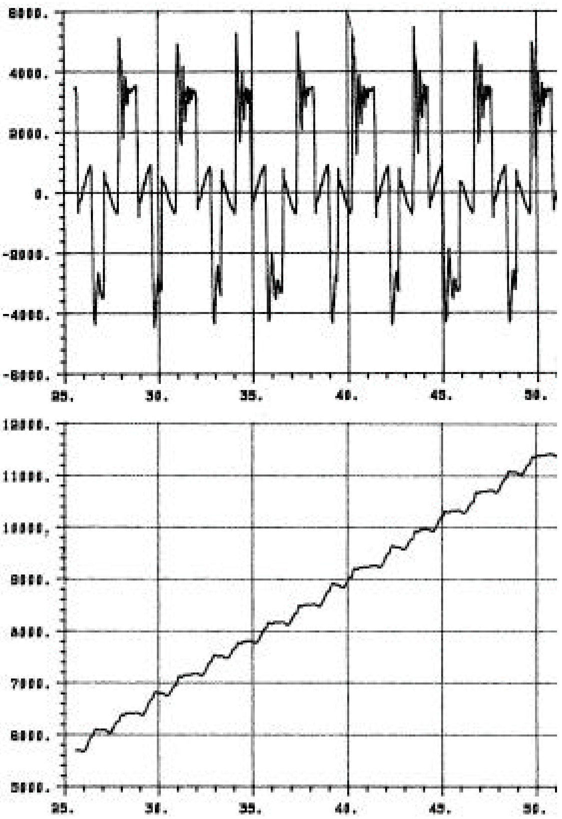

Figure 3. *Building-up of input energy E_b from 5.6 kJ to 11.4 kJ during 25 s when the incident wave has a height (0.18 ± 0.02) m and a period 3.1 s. The upper graph shows the force in the mooring strut varying over a 9-kN range* [14].

The Pelton wheel, indicated in figure 1, was in the experiment replaced by a throttle valve. Flow rate and pressure drop were measured. The corresponding average hydraulic power amounted to 50 W, which is significantly smaller than the input power P_b from the wave. Energy loss resulted mainly from friction, as well as leakage, in the seal of the piston pump. However, also hydraulic pipes and valves, improper latching, and elasticity in the mooring contributed to energy loss.

A technical and economic assessment was carried out in 1978 for a full-scale buoy of diameter 6 m and cylindrical height 8 m [9,14]. Kværner Brug AS, Oslo, and Institutt for marine konstruksjoner, NTH, Trondheim, assisted with design of hydraulic machinery and hull structure, respectively. It was envisaged that a stationary platform, common for 10 nearby power buoys, should contain a hydraulic motor and electric generator of capacity 3 MW, in addition to common high-pressure and low-pressure reservoirs. Thus the indicated gas accumulators B and C (figure 1) were replaced by a pair of hydraulic hoses leading hydraulic fluid between each buoy and the common platform. Using wave data from a sea location off Halten (64^o N, 9^o E), it was estimated that each power buoy, during an average year, could absorb an energy amount of 1.1 GWh [9]. Based on information from Norwegian companies, it was estimated that, for a 300 MW power plant consisting of 1000 type-E buoys, the production cost in 1978 would be in the range of NOK 1.2-1.8 million per buoy, not including electrical equipment. This investment cost would exceed the then current investment cost for hydroelectric power plants [9]. This fact, combined with Budal's great ambitions, was a good excuse for inventing other, hopefully more prosperous, types of wave-power buoys. Moreover, it was considered safer to use a stiff mooring strut, rather than a mooring cable.

4. POWER BUOY OF TYPE M2

After assessing the E-type power buoy, Budal proposed another type of point-absorber power buoy, where compartments of the hull serve as gas accumulators or gas pressure chambers A_1 and A_2 ; see figure 4. Following the wave-energy fashion of the time, we intended to use pneumatic, rather than hydraulic, power take-off machinery. Then a mechanical device L had to be used for latching. To reduce friction loss, rolling diaphragms RD were used as seals in the piston pump. The piston rod PR is connected, through a mooring strut MS, to an anchored universal joint UJ on the sea-bed – not through a cable MC as with the E-type buoy. – Pretension in the connection is obtained by means of air pressure in accumulator A_1, which, in addition, ensures that the gas pressure is higher on the inner side of the two rolling diaphragms. Accumulator A_2

serves energy storage. A conventional air turbine utilises the unidirectional flow of air from accumulator A_2 to the outer atmosphere – which corresponds to accumulator A_3 in figure 1. – A not very important new feature concerns the shape of the hull; see figure 4. A conical shape was chosen in order to test another geometry than the cylindrical shape indicated in figure 1. An idea behind this choice was to utilise wave-generating water displacement as high up as possible in the sea.

A type-M2 buoy model (of maximum diameter 0.885 m and height 2.4 m) was constructed and tested in 1980 in the 10.5 m wide ship model tank at NTH, with regular as well as irregular waves [19]. Some experimental results have been published previously [12]. For an incident sinusoidal wave with period T = 3.2 s, experimental hydrodynamic parameter values, f_{e1} = 2.9 kN/m and R_{r1} = 17 Ns/m were found. With a wave amplitude A = 0.106 m, heave amplitude values s_m = 0.25 m, s_1 = 0.30 m and u_1 = 0.59 m/s were obtained. The input energy E_b according to eq. (1) was measured, giving an average input energy P_b = 76 W, while the two first terms in eq. (2) yield P_e - P_r = (89-3) W = 86 W. That these power values and heave amplitude values were significantly smaller than corresponding values obtained with the type-E buoy, must be attributed to a heavier power take-off loading with the M2-type buoy. In spite of this fact, a slightly higher value P_b/V = 140 W/m^3 was obtained. This is reasonable because the M2 buoy model has a significantly smaller volume V = 0.54 m^3 than the volume of the E-type buoy model. For a smaller volume, less deviation from by Budal's upper-bound relation P/V $< pr\,gA/(2T)$ may be expected. The M2 buoy model was also tested in irregular waves. For an example, with an energy period T_{-1} = 3.2 s and a significant wave height H_s = 0.40 m (JONSWAP spectrum) an average input power of P_b = 60 W was obtained. In this particular run, there may have been more friction in the system, because this run was the final one with the M2 model, before the lower rolling diaphragm punctured, after the rubber had swollen as a result of oil leakage from the hydraulic system for operation of the latching mechanism.

A detailed technical assessment of a full-scale M2 buoy (of volume approximately 500 m^3) was made [16]. The envisaged hull had a total height of 26 m and a maximum diameter of 8.7 m. The stroke length was 8 m (9 m including end stops), and the installed power capacity was 300 kW. Annual production of electricity was estimated to be approximately equal for a buoy of type M2 as for type E. The assessed production cost (investment) appeared, however, to be higher. This finding was a motivation for a modification that avoided some of the expensive components of the M2-type power buoy. This led Budal to propose the N2-type power buoy.

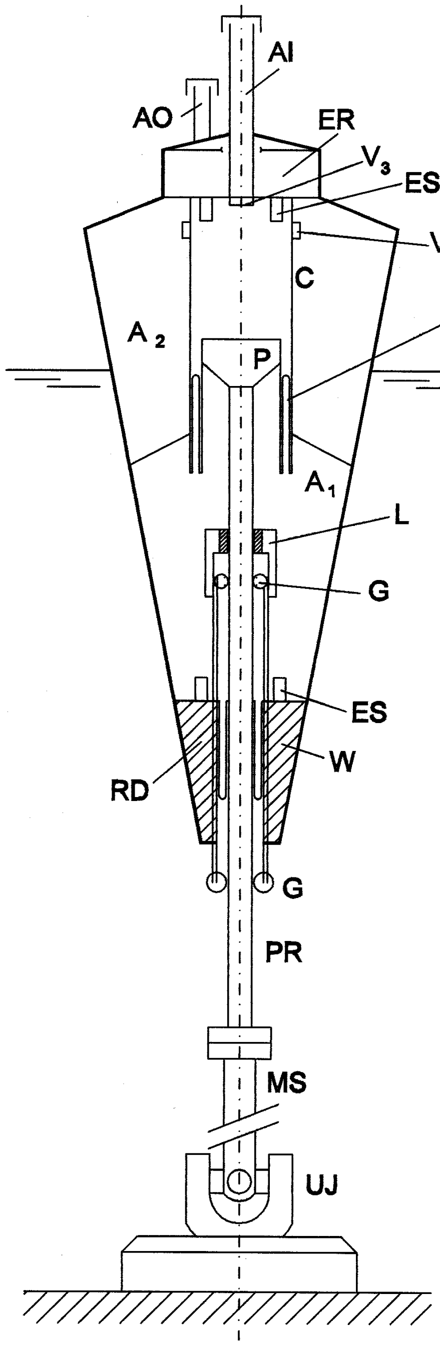

Figure 4. *Power buoy of "type M2". The machinery consists of a cylinder C, a piston P, a pneumatic energy-storing accumulator A_2 , two air-rectifying check valves V_2 and V_3 , besides air inlet and outlet pipes, AI and AO respectively, and an air turbine with electric generator (not shown) inside the engine room ER. The piston rod PR is connected to a mooring strut MS, pre-tensioned by the pressure in accumulator A_1 . The strut is connected to an anchor through a universal joint UJ. The relative motion of the buoy along the piston rod may be latched/unlatched by activating/deactivating the mechanism L. The system is provided with guiding rollers G, with end stop buffers ES, with ballast weight W, and with rolling diaphragm seals RD.*

5. POWER BUOY OF TYPE N2

The N2-type buoy differs from the M2 one in several ways. A spherical hull shape was chosen, firstly, in order to reduce the amount of material necessary to construct a sufficiently strong hull, and secondly, in order to reduce extreme-wave induced maximum bending moments in the mooring strut. Moreover, end-stop problems are reduced, and operation of the wave-power buoy is less influenced by the state of the tide. The pneumatic piston pump with necessary seals in the M2 buoy is replaced simply by an OWC contained within the N2 hull, where an opening in the bottom serves as the mouth for the OWC. Together the heaving hull and the OWC compose a two-degree-of-freedom oscillating system interacting with the waves. Inside the hull there is a latching mechanism which can latch the buoy to the strut when the relative velocity is zero, and which can unlatch at an appropriate instant. There are also two air chambers, one high-pressure chamber and one low-pressure chamber, between which an air turbine with electric generator is installed. Either air chamber has a check valve communicating with the air volume above the OWC. Additional check valves, communicating with the outer atmosphere, ensure that the pressure in the low-pressure/high-pressure chamber never gets higher/lower than the ambient atmospheric pressure. In order to provide for an upright equilibrium orientation of the mooring strut, a submerged weight W is connected to the buoy hull B, as indicated in figure 5. Produced electric power is brought to shore through electric cables EC. Between the weight W and the seabed this cable is carried partly by a submerged buoyant body BB, and here the electric cable has to be of a particularly flexible type.

A 200 MW wave-power plant, consisting of 410 units of 10-m-diameter N2 buoys, was assessed technically and economically [1,13,15,21]. The layout, in sea of depth 40 m, was an array of groups approximately 120 m apart. Each group was a line of five buoys, 30-40 m apart. A very detailed technical report [17] contains information on design details, on laboratory testing of mechanical components, on reports from technical consultants and on offers from industrial companies. This document would form the basis for asking for tenders with the purpose of constructing a test buoy in full scale. It was expected that the buoy would function with a reasonable degree of reliability during a testing period of two to three years, provided the functioning of critical components were sufficiently tested before installation. Among critical components were the latching mechanism and the guiding rollers. We also co-operated with the wave group at University of Edinburgh, where Stephen Salter, as an alternative, had proposed a magnetic-repulsion enhanced hydrostatic bearing for guidance of the buoy along the mooring strut [24].

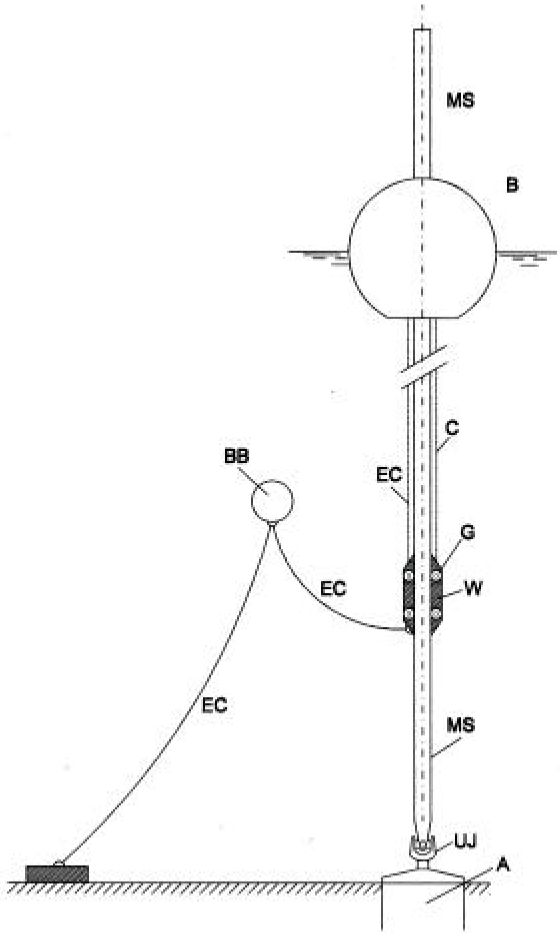

Figure 5. *Power buoy of "type N2". A spherically shaped buoy hull B, which is connected to a submerged weight W, through cables C, is arranged to slide up and down along a mooring strut MS, which is connected to an anchor A through a universal joint UJ. Guiding rollers G on the weight W, as well as on the buoy B, serves easy movement with little friction. The hull is open in the lower end, and it contains an OWC as well as compartments for providing buoyancy, high- and low-pressure chambers, rectifying valves and an air turbine with electric generator. It also contains a mechanism by which the buoy B may be latched to the strut MS. A flexible electric cable EC, supported partly by a submerged buoyant buoy BB, serves transmission of converted energy.*

An official assessment [1] of four different (three Norwegian and one British) proposed wave-energy devices, deployed as a 200 MW wave power plant off the Norwegian west coast (at 61.9 ^{o}N, 4.8 ^{o}E), estimated in 1981 the energy cost to be 1.2, 1.3, 1.4 or 2.3 NOK/kWh if the plant would consist of units of Kværner Brug's bottom-standing multi-oscillating OWC, of NORWAVE's Tapchan converter, of our N2 power buoy, or of NEL's bottom-standing OWC, respectively. (Because of more expensive labour in Norway, the figure of 2.3

NOK/kWh is more than twice the corresponding figure given in British assessments of the NEL device.) An updated assessment of the Norwegian proposals in 1983 showed estimates of energy unit costs about half of the figures estimated in 1981 [21]. It was estimated that the energy-recovery time would be in the range of 10 to 14 years for Kværner Brug's and Norwave's proposed wave-power plants [1]. For the N2-type of plant we estimated the energy-recovery time to be less than two years. Although less material-making energy is needed, it is, however, obvious that more labour is needed for constructing and maintaining a plant of the N2 type, than for the other proposed plants.

6. SEA TESTS WITH N2-TYPE MODEL

A model in scale 1:10 of an N2 buoy was tested in the sea near Trondheim, at a location where the depth varies between 4 and 7m (depending on the tide). The diameter of the model buoy is 1 m. Contrary to the full-scale buoy, the latching mechanism is placed outside of the hull, as indicated in figure 6. Moreover, the model buoy contains neither high-pressure and low-pressure chambers nor a turbine. Pneumatic power is dissipated through a calibrated orifice, and its quantitative value P_p is derived from measured values of the pressure of air above the OWC within the hull. Additional power P_f is dissipated by friction between the mooring strut and the buoy (including submerged weight). By measuring the buoy position relative to the strut and the longitudinal force in the mooring strut (using a strain gauge just above the universal joint), an experimental value for P_f is found, using an analogy of eq. (1). As experimental value for the absorbed wave power we set $P_a = P_p + P_f$. Note however, that possible wave power absorbed by viscous losses in the sea water is not included in P_a. Most difficult was to make good measurement of the waves. Several pressure transducers, placed at different locations in the water were used. To determine the significant wave height, transducers at some distance from the buoy were used. The signal $p_b(t)$ from another transducer placed on a collar on the mooring strut, giving better information on the wave phase at the buoy, was used as input to the computer for determining optimum unlatching instants. This collar was placed between the buoy and the submerged weight, which could kick the collar downwards and upwards along the strut, when the tide was falling and rising, respectively. The Kalman filter used for determining optimum instants of unlatching, is described elsewhere [13,18].

Some experimental results are shown in table I. Each line in the table shows experimental values derived from analysing measured values during $n_s/2$ oscillation cycles, lasting for a selected time interval, of length $\langle T\rangle n_s/2$, where n_s is the number of strokes and $\langle T\rangle$ the average oscillation period during this interval. However, the significant wave height H_s is derived from wave measurements during 25.6 s (512 data points), including the shorter said interval. The average wave amplitude is assumed to be $\langle A\rangle = H_s/2.83$.

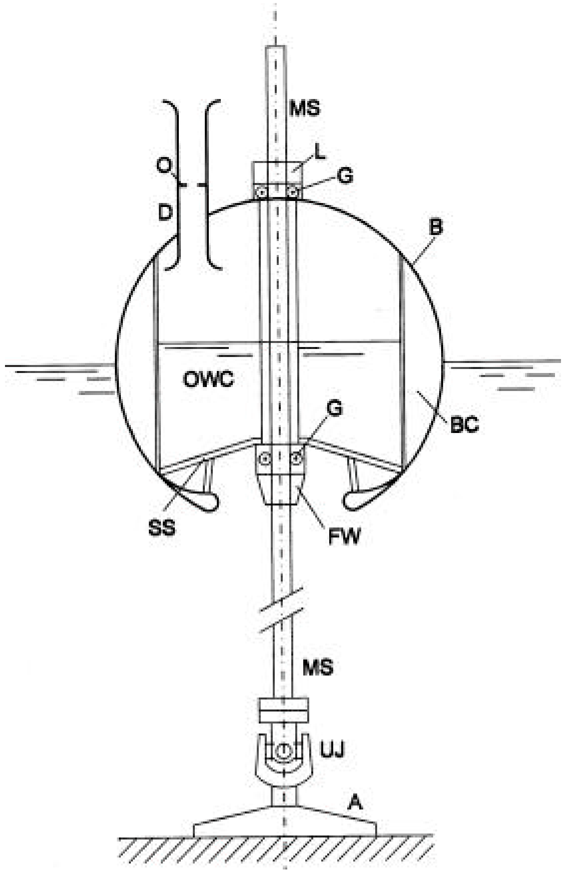

Figure 6. *Model (in scale 1:10) of power buoy of type N2. The hull B, which has a diameter of 1 m, is open in the bottom, for providing communication with an internal OWC. The annular air chamber BC provides buoyancy. By means of guiding rollers G, the buoy is easily movable up and down along a long mooring strut MS. The buoy may be latched to the strut by operating a latching mechanism L. D is a duct with calibrated orifice O. SS is a supporting stay, FW is a flow-evening housing, UJ a universal joint and A an anchor. The model includes also a suspended submerged weight W, which is not shown here (contrary to figure 5).*

In table I, the average maximum buoy excursion $\langle s_m\rangle$ from equilibrium is found as the total buoy walk (up and down) divided by $4n_s$. If latching control is to function for optimising the phase, then the velocity should have the same phase as the excitation force. For each stroke we measured the time difference t_b between the occurrence of the extreme of the heave velocity $u(t)$ and that of the measured pressure $p_b(t)$ assumed to represent the excitation force. We compute the quantity

$$a_0 = \overline{p_b(t)u(t)} \Big/ \overline{p_b(t)u(t+t_b)} \qquad (4)$$

which is a measure of how good the phase control was [12]. (Here the overbar symbolises averaging during the analysed time interval.) Note that this measure is dimensionless and, usually, smaller than unity. Setting $a_0 = \cos\langle ß \rangle$, the quantity $\langle ß \rangle$ may be interpreted as an "average phase angle" between the heave velocity and the excitation force. Experimentally, most stable latching-control operation was obtained in situations when the wave spectrum was close to a JONSWAP spectrum, peaking at approximately 0.3 Hz.

Table I. *Experimental results from sea tests with N2-buoy model. Table entries are:* n_s *is the number of heave strokes, and* $\langle T \rangle$ *is the average oscillation period during a selected analysed time interval of length* $\langle T \rangle\, n_s/2$, *within a measuring record,* $\langle A \rangle$ *is the average wave elevation amplitude,* $\langle s_m \rangle$ *is the average heave amplitude,* $\langle b \rangle$ *is the "average phase angle" between heave velocity and excitation force,* P_p *and* P_f *are measured pneumatic power and friction power, respectively. Finally,* P_a *is the experimentally absorbed power, and* P_t *is a theoretical estimate of the latter. The 14 last lines of the table are results obtained with the modified buoy hull* [18,20].

n_s	$\langle T \rangle$/s	$\langle A \rangle$/m	$\langle s_m \rangle$/m	$\langle ß \rangle$/°	P_p/W	P_f/W	P_a/W	P_t/W
12	3.3	0.11	0.18	20	49	9	58	105
6	3.3	0.13	0.19	20	56	12	68	133
10	3.4	0.10	0.19	10	53	11	64	102
12	3.3	0.09	0.21	14	59	15	74	97
8	2.8	0.04	0.13	10	20	10	30	23
14	2.6	0.04	0.06	9	5	5	10	14
14	3.0	0.04	0.15	0	21	9	30	27
8	2.9	0.10	0.11	8	15	8	23	67
10	3.4	0.11	0.24	0	42	16	58	141
12	2.8	0.14	0.22	16	33	16	49	172
12	3.1	0.12	0.19	11	34	14	48	130
10	2.0	0.05	0.12	0	13	13	26	19
------	-------------	-------------	-------------	-------------	-------------	-------------	-------------	-------------
16	2.9	0.055	0.06	28	9	3	12	18
8	2.95	0.08	0.15	27	29	9	38	59
10	2.85	0.04	0.09	36	10	9	19	15
4	2.85	0.06	0.13	45	25	6	31	27
4	3.55	0.11	0.29	29	99	21	120	135
4	3.15	0.13	0.30	25	111	23	134	185
4	3.05	0.17	0.38	26	134	34	168	305
4	3.0	0.19	0.33	35	79	23	102	282
4	3.5	0.15	0.38	24	149	30	179	260
4	3.2	0.12	0.29	58	65	9	74	75
4	3.5	0.11	0.29	55	74	12	86	75
4	3.3	0.14	0.38	51	106	21	127	141
4	3.25	0.13	0.28	59	37	13	50	80
4	2.9	0.14	0.25	64	29	11	40	64

The measured power $P_a = P_p + P_f$ may be compared with the theoretical estimate P_t shown in the last column of table I. This estimate is obtained by inserting experimental values into the approximate formula [18]

$$P_t = \tfrac{1}{2} f_{e1} \langle A \rangle \Omega \langle s_m \rangle \cos\langle b \rangle - \tfrac{1}{2} R_{r1} (\Omega \langle s_m \rangle)^2 \qquad (5)$$

which is analogous to the two first terms of eq. (2). Here $W = 2pg/\langle T \rangle$, where [12]

$$g = \frac{4 \cos(pd\ /\ 2)}{p\ (1-d^2)} \qquad (6)$$

and $\delta = T_b/\langle T \rangle$ with natural period $T_b \approx 1.5$ s for the heave mode of body B.

The plots shown in figure 7 present a comparison between the P_a and P_t values given in the two last columns of table I. The plotted diamond points, corresponding to the twelve upper lines in table I, present experimental results with the original buoy hull indicated in figure 6. We may observe that there is a fair agreement for low values of P_a and P_t, while for higher values of theoretical estimate P_t the really measured absorbed power P_a shows a saturation effect for values above 50 W. As we believed that this was caused by viscous losses at the entrance of the OWC, we decided to modify the hull. The diameter of the entrance and the radius of curvature there were both increased to 0.60 m and to 0.05 m, respectively [2, fig.5]. Since this hull has a deeper draft, it deviates more from spherical shape, and the natural heave period is increased ($T_b \approx 1.8$ s). Results from experiments with this modified buoy hull are presented in the lower 14 lines of table I. Correspondingly, plotted square points in figure 7 show results with the modified buoy hull. We may observe that the level of saturation is raised to about 120 W. Thus, the modification of the hull was successful. A lesson to learn is that it is important to avoid increased water-flow speed and small radius of curvature at the OWC entrance.

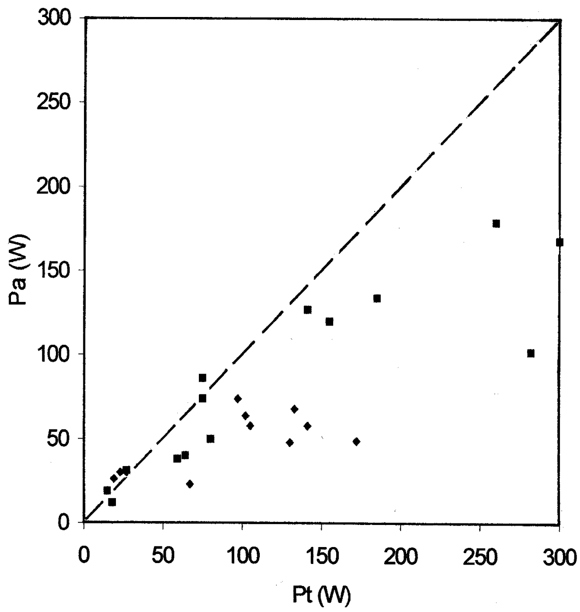

Figure 7. *Experimental results from sea tests with N2 model. Absorbed power P_a (measured as sum of pneumatic power dissipated by the orifice and of power lost in friction between buoy and strut) versus absorbed wave power P_t as theoretically estimated (based on measured wave and measured buoy oscillation). Plotted diamonds and squares show results obtained with original hull (figure 5) and modified hull, respectively* [18,20].

The upper graph in figure 8 shows a typical record of the relative motion between the buoy B and the strut MS during an experimental run. It is seen that the rapid oscillations are superposed on a more slow oscillation. The latter is a result of the mooring strut's pitching motion around the anchored universal joint UJ.

7. DIFFERENT LATCHING MODES

While latching operation should take place only at an instant of zero velocity of the buoy, there may be different options for the instant of unlatching. Assuming that the wave period is larger than the natural period of the buoy, the normal procedure ("mode 1") is to let the buoy be latched during two time intervals of each oscillation cycle. Cf. figure 8, upper graph. For achieving optimum phase, the target of unlatching is to obtain maximum oscillation velocity at the instant of maximum excitation force. Another alternative ("mode 2") is to let the buoy be latched only during one interval of each wave cycle. With the N2 buoy, we tested such an alternative by latching only at the instant when the velocity was zero in the lowest extreme heave position [20]. Cf. figure 8, lower graph. This choice was motivated by e.g. the need to increase vertical stability of the N2 system when, occasionally, strong currents appeared in the sea. For achieving (sub-)optimum phase, the target of unlatching is, in this case (mode 2), to obtain maximum heave position, and hence zero down-crossing of the velocity, at the instant of zero down-crossing of the excitation force. Cf. figure 9. With the N2 buoy we tested even another latching alternative ("mode 3") where the buoy hull was kept latched continuously. It may be necessary to apply this mode in situations of extreme weather. In this latching mode, the N2 system becomes an OWC contained in a non-heaving structure.

Using the modified buoy hull, we tested the three latching modes during three consecutive runs. For the first two of these runs the relative oscillation was as shown by the two graphs in figure 8. Results are shown in table II. (We may observe that the significant wave height was slightly reduced for the last run.) It is remarkable that, for this particular wave, where the energy period $T_{-1} = 2.5$ s is not very much larger than the buoy's natural heave period $T_b \approx 1.8$ s, energy capture is reduced by only 12 % with mode 2, as compared with mode 1. A larger difference would be to expect for much higher values of T_{-1}/T_b. With mode 3 the energy capture width (P_a/J) is reduced to approximately 50 %.

In the past, we have considered that application of latching in order to obtain optimum phase, is a useful method only when the wave period is longer than the natural period of the wave-absorbing oscillator. We have noticed, however, that it was claimed, already in 1982, that latching control could improve power conversion by the NEL breakwater wave energy converter also at wave periods shorter than the natural period [25]. This matter has, recently, been studied in more detail with application to a heaving wave energy device [26]. To explain this, we shall, for simplicity, assume that the incident wave is sinusoidal with period T. Because of latching, the oscillation of the device is not sinusoidal, but periodic. Let us assume that its period is nT (n = 1, 2, 3, 4, ...). Its fundamental frequency is $1/nT$, and its n-th harmonic frequency equals the frequency $f = 1/T$ of the sinusoidal excitation force. Hence, only the n-th harmonic of the velocity can contribute to the time average of the input power, the product of velocity times force. If the natural period of the device is larger than T, then it is necessary that $n > 1$. Even in such a case it may happen that latching control improves power conversion, in spite of the fact that the system has to remain latched during (n-1) wave cycles between each unlatching. To utilise the device as much as possible, it is desirable to choose the smallest possible integer n. Since the latching “mode 1” may result in a symmetric oscillation with negligibe even harmonics, it may be profitable to apply latching “mode 2”, which favours non-symmetric oscillation, and hence even harmonics. Thus, by applying “mode 2” instead of “mode 1”, it may in some cases be possible to choose $n = 2$, rather than $n = 3$. Usually, the higher harmonic components will be small compared to the fundamental component. When limitation caused by design bounds (on amplitude and/or power-handling capacity etc.) has to be taken into account, one will probably find it practical and beneficial to apply latching control only when it is possible to avoid larger period for the oscillation than for the wave. Then it is not practical to apply latching phase control when the wave period is shorter than the natural period of the oscillation system. In particular if latching "mode 2" (rather than "mode 1") is utilised, this statement, cannot, however, be applied rigorously in the case of irregular incident waves, where a certain proportion of individual zero-crossing wave periods may be shorter than the natural period. Among the individual waves, some of the shorter ones may then be omitted concerning unlatching operation of the heaving body.

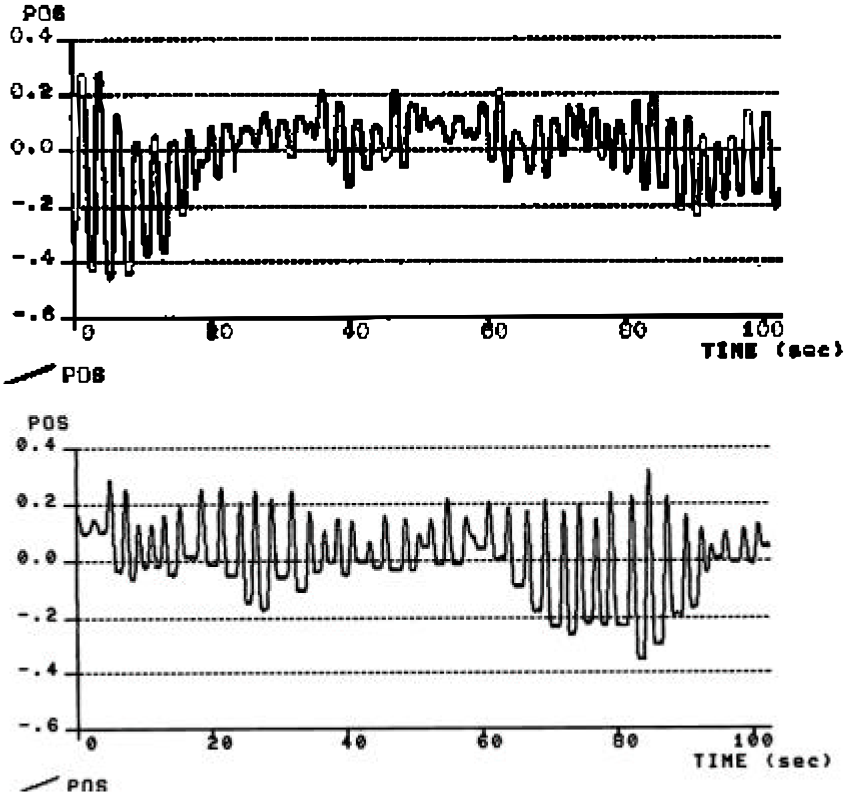

Figure 8. *Measured values (in metres) of the buoy position relative to the strut, during 100 seconds of two runs with the N2 model. The upper and lower graphs are obtained with two latching intervals ("mode 1") and one latching interval ("mode 2") per oscillation cycle, respectively* [20].

Table II. *Power absorbed for three consecutive runs of the N2-buoy model, with different latching strategies. The significant wave height H_s was slightly reduced at the time when the third run was made* [20].

H_s /m	T_{-1} /s	J /Wm^{-1}	Latching strategy	P_a /W	(P_a /J)/m
0.24	2.5	69	2 latchings/cycle	18	0.26
0.24	2.5	69	1 latching/cycle	16	0.23
0.22	2.5	58	Latched all time	7	0.12

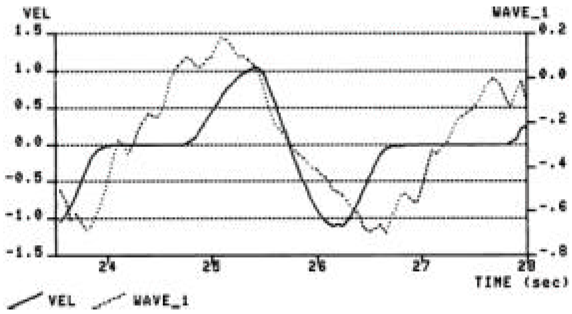

Figure 9. *Experimental values of the buoy velocity relative to the strut (in m/s) and of the hydrodynamic pressure (in kPa) at the collar on the strut, which pressure is a measure of the wave. Results are obtained in another "mode-2" run with the sea-tested N2 model* [20].

8. CONCLUSION

At the university in Trondheim, we have worked on phase-controlled point absorbers since the mid 1970s. Initially, we mostly considered hydraulic power take-off. However, about 1980, following the "wave-energy fashion" (of that time), we, unfortunately, changed our attention more to pneumatic power take-off. It is in retrospect that we may say "unfortunately". In the early 1990s, when Mr. Håvard Eidsmoen became our doctorate student, we again turned our attention more towards hydraulic machinery for control and power take-off. His study was directed to the continuation of researching the latching-controlled power buoy (type E) that Budal proposed in 1978. In retrospect, it may be considered as a pity that we did not continue research and development on that proposal during the 1980s.

We wish to comment here on pneumatic versus hydraulic power take-off for wave-energy converters: Although OWCs may certainly still be considerably improved, it is our belief that in future (when necessary components and technologies have been sufficiently developed) the best wave-energy converters will have hydraulic power take-off. Our main reason for this belief is that the needed, very flexible, control requirements may be better realised with hydraulic equipment, and desirable short-time energy storage by means of gas accumulators is possible. Moreover, it seems that hydraulic equipment have development potential to obtain high energy-conversion efficiency.

If we had had the possibilities (funding and personal resources) to continue further work on technological development on “Budal's latching-controlled-buoy type wave-power plant“, then we would propose to modify the N2-type power buoy by replacing the pneumatic power take-off by a hydraulic one. This would necessitate to replace the OWC contained within the buoy by a float, to which is connected the piston rod of a hydraulic cylinder-and-piston pump. A suitable hydraulic power take-off, to replace the pneumatic one in the N2-type power buoy, could be as indicated in figure 1, but with gas accumulator A_1 and valve V_1 eliminated. If not eliminated, a wave-power converter would result with latching control facilities in both oscillators of this 2-degree-of-freedom system. According to our knowledge, such a system has not yet been studied or analysed. As yet, we cannot know whether the added complication of including a second latching device (controllable valve) will be sufficiently beneficial to justify the added cost.

ACKNOWLEDGEMENTS

We wish to thank Professor Arne Kjølle (at Vannkraftlaboratoriet, NTH) for his contribution in 1978 to the design of hydraulic components, in particular the controllable valve, in the type-E model. The work reported in the present paper received substantial financial support from OED during the years 1978-83. Then this project and the simultaneous Kværner project [27] had a joint steering committee with two members from NTH, two from Kværner Brug AS and one from National Industri AS. In recent years, research funding has, however, been provided by the Industry and Energy Division of the Norwegian Research Council.

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