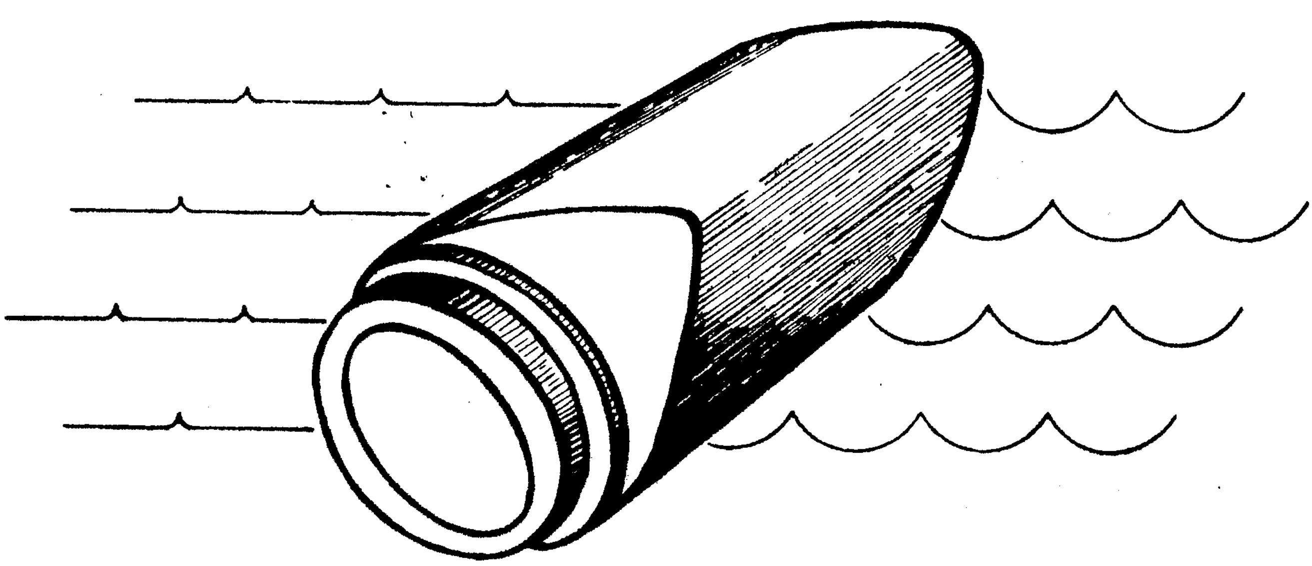
COMMERCIAL IN CONFIDENCE

Edinburgh - Laing

(in association with Merz & McLellan)

Wave Energy Device



October 1981

COMMERCIAL IN CONFIDENCE

UPDATED REPORT ON THE EDINBURGH - LAING WAVE ENERGY DEVICE (THE DUCK)

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SECTION 1

INTRODUCTION

1. INTRODUCTION

1.1 A Brief History of Development

John Laing Limited have been associated with Stephen Salter, of Edinburgh University, since the late autumn of 1978. Resulting from this association, we were invited by ETSU in March of 1979, together with Scottish Offshore Partnership, to become integrated members of the Edinburgh Project Team and to undertake a review of the civil and mechanical structural aspects of the Project. This work culminated in a report submitted to ETSU in 1979 and subsequently presented at a Workshop in December 1979 at Maidenhead. Since that time any further work that has been carried out has been generic rather than device specific with limited funding from the Department of Energy. In May 1981, at the request of ETSU, a proposal was submitted to The Wave Energy Steering Committee for updating the 1979 report. This presentation led to an Instruction to Proceed being issued by the Department of Energy in August 1981.

The overall concept of the Duck remains unaltered in that it uses the rotational characteristics of Gyroscopes to create a reference platform within the duck body as it 'nods' under the wave action. The differential movement of the two parts operate ring cam pumps and they produce hydraulic pressure. This oil is fed to a swashplate pump/motor unit driving a synchronous generator. The continuous spine on which the ducks are mounted still has flexible and intelligent joints, although these joints too have been the subject of re-design in the interests of making the costs very much lower. The Duck body is now somewhat larger in both diameter and length, and although the power rating per metre has been reduced to obtain a better duty factor, the output per duck remains the same.

1.2 Philosophy Statement

To maximise the energy capture there are advantages in locating wave energy devices in deep water. Adverse consequences from this are the distance from a support base and the increasing difficulty of effective maintenance as the distance gets larger. The further offshore the device is located the shorter the time available in weather windows and the less effective any "in-situ" maintenance is likely to be due to the hostile environment.

From the early conceptual stages, the Edinburgh Device Team decided that its principal objective would be that the device should require minimum maintenance throughout its design life. The device team's target is 25 years, although from work carried out by EASAMS a period of 5 years results in a viable installation.

This approach to the design has been developed in three ways:-

1. By building in redundancy of essential items during construction, so that normal anticipated failure and wasteage during the effective life of the device will not reduce efficiency. Condition monitoring will be installed to maintain efficient operation automatically.

2. By the use of parallel circuitry to ensure that failure of any one separate system is not a catastrophic failure of the whole.

3. By providing the best environmental climate to ensure that any item of equipment operates effectively at its maximum potential.

4. By sealing the energy conversion unit and electrical generating systems in purpose-made machinery canisters to isolate them completely from the marine environment.

It is well known that high pressure oil hydraulic systems can develop high power levels from small components and that, if the correct working environment is provided, they will have a very long, reliable and unattended life. The device relies for its effective operation upon High Pressure Oil Hydraulics. The design of the device is such that a vacuum stripped oil not only operates in an ideal ambient atmosphere, but also that this oil is being centrifuged throughout its entire operating life. Thus it is hoped that the ever present problem of "dirty" hydraulic oil is eliminated.

A unique advantage of the power take-off chain of the device is the capability of transforming the random input of energy from the waves into a uniform supply of high pressure oil to the generator motor. This is achieved by using the flywheels as short-term energy stores, thus permitting synchronous generation of alternating current, with the consequent saving on electrical costs.

The Device Team have consulted firms specialising in all the major items of the proposed power conversion system and have explained their design philosophy. The approach has been fully endorsed by these specialists and they have expressed their confidence not only in the overall scheme but in each case have stated that, in respect of their own speciality, they can either meet the requirements with current technology or from development of their current techniques.

The Device Team consider that the Minimum Maintenance Concept is the most realistic approach to developing a cost effective Wave Energy Device.

The device has not, however, been developed beyond the outline schematic stage necessary to confirm feasibility. The development to this stage has been meticulously carried out and carefully verified by tank testing, but in the narrow tank only. It is essential that all future development work should be to this same standard if confidence is to be established in the Minimum Maintenance Concept. The Device Team consider that it would be wrong to sacrifice the prospect of long term success for short term expediency. For these reasons, it is unlikely that the device could be developed to the requisite standard for 3 - 5 years. Confirmation of results in the wide tank is currently taking place and early results are exactly in line with narrow tank predictions and cosine directionality corrections.

A full size power canister should be built and put on test. It is appreciated that this would entail the development of suitable hydraulic motors. Since the whole philosophy is to isolate the power chain from the sea, this testing may be carried out ashore, by using simulated wave motion, with the resulting accessibility for instrumentation to monitor performance. The object here would be to evaluate the individual power train components for long maintenance-free life.

SECTION 2

DESCRIPTION

2. DESCRIPTION

2.1 Location and General Description of Scheme

The anticipated location for the Duck Wave Energy Device is to the west of the Outer Hebrides, off the island of South Uist or Benbecula. It is proposed that a string of devices, approximately 46 km long including navigation gaps every 5 km, should be located at a distance of some 35 km off shore in a depth of water of 100 m. The devices would be arranged in a shallow zig-zag, each leg of which would consist of 4 Ducks. The mean normal of the zig-zag would be 10^{o} South of West (See Drawing No. 10021).

This arrangement will give a 2 GW output with a rating of 60 KW per metre and a 45 metre theoretical length for each Duck. The mean annual yield is 656 MW of synchronous AC into the grid..

2.2. Principle of Operation Device

The Duck comprises a series of concrete structures mounted in pairs on a cylindrical concrete spine which is jointed at intervals to limit the bending moments. These structures, or Duck bodies as they are called, respond to the wave action with a nodding motion in the same way as a ship at sea rolls on the waves.

The Duck structure carries at its front edge two sealed canisters each housing a pair of gyroscopes arranged so that the precessional motion creates a reference platform against which the nodding action of the Duck can be used to drive a series of pumps producing a flow of high pressure oil.

This high pressure oil is first led to swashplate hydraulic pump/motor units which drive the gyro discs, thus enabling them to act additionally as energy storage flywheels and enables them to smooth out the pulses of power from the waves into a continuous hydraulic pressure which is used to drive a further swashplate pump/motor unit coupled to a synchronous AC generator. The power from this generator is led into the central spine and the outputs grouped in accordance with Drawing No. 10071 Fig. 3. Transmission ashore is shown on Drawing No. 10071 Figs. 1 & 2. The mooring arrangement allows the duck string to move towards the wave front and away from it by a distance of some 50 metres; hence the string can adjust to the wave climate. In tank tests a freak wave, 50% above the 50 year wave, has been driven at a spine model. It produces joint angles of 5^{o} and mooring tension variations of 30% above the mean level of 1 ton/metre.

2.3 FORM OF STRUCTURE

2.3.1 Primary Structure

The primary structure comprises two elements, namely the "duck body" and the "spine".

The Duck Body (Drawing No. 10102.)

The "duck body" is a buoyant cellular structure in reinforced and prestressed concrete carrying the power canister.

Along one side a semi cylindrical recess matches the cylindrical surface of the spine, although separated from it by a bearing which permits relative rotation about the spine.

The "duck body" is 36 metres long in the axial direction and approx. 14 metres wide across the cylindrical recess (the spine being 14 metres in diameter). Within the duck body the power canister, 5.550 m external diameter, is located so that the distance from the centre of pod to centre of spine is 10.420 m. When attached to the neutrally buoyant spine, the duck body floats with the axis of the spine at a depth 8.7m below still water surface. With the centre line of the duck at an angle of 36° to the horizontal, the freeboard is 1.85 m and the length of waterline is 13.25 m.

The cellular voids in the duck body extend longitudinally and provide space for the power canister as well as for both permanent and adjustable ballast in order to achieve the correct attitude and buoyancy. Space is also available for ancillary equipment.

The Spine (Drawing No. 10101.)

The "Spine" is a 14 metre diameter reinforced and prestressed concrete hollow cylinder with a wall thickness of 800 mm. Sections are 90 metres long between the centres of the moment controlled joints. Each section carries two ducks.

At the joints, the ends are closed by bulkheads and the walls of the cylinder are extended beyond it at two points on the perimeter to carry the trunnion bearings of the Hooke's Joint.

At a point half way between the moment-controlled joints, the spine has a construction joint which can be joined or separated at sea. For this reason a bulkhead is provided in each half together with other provisions such as local wall thickening for the prestressing bolts clamping the joint. In each half a bulkhead is provided at the midpoint to stiffen the cylindrical walls and to limit the consequences of damage.

Externally, peripheral ribs are provided for lateral location of the ducks at each end of the 45 m spine length. Means of access, in the form of "conning towers", are also provided and these carry warning lights, radar reflectors and other communications equipment.

2.3.2 Basic Calculations

1. Scope

This appendix to the report describes briefly the calculations prepared to justify the design of the spine and duck described herein.

2. The Spine

a) General Arrangement of Spine Joints

The Spine consists of a 14 m O/D longitudinally prestressed concrete pipe, having a wall thickness of 800 mm. The pipe is formed in 45 m lengths, adjacent lengths of pipe being connected together by non-yielding and flexible joints in an alternate sequence. The rigid joints are formed by prestressing together the ends of 2 adjacent spines, the mating surfaces being formed by means of a thermo-setting resin in a thin stainless-steel envelope. The joint is designed to be of equal strength to the standard spine sections, and has been arranged to allow it to be joined at sea.

The flexible Hooke's joints are capable of transmitting bending moments by means of 8 hydraulic rams, and will be assembled on land. The rams are connected via primary hydraulic motors to a pair of high and low pressure hydraulic mains and the pressure is controlled to offer resistance up to a maximum surge or heave moment of 20,000 tonne-metre (20 KTM) and then to yield at constant moment.

b) Design Moment for the Spine

Wave action can cause extra bending moments between the flexible joints (located at 90 m centres) to be greater than that at the joints. An allowance of 18% has been made to cover this and hence the maximum surge or heave moments of 23.6 KTM are considered. However particles within waves follow circular orbits and typically heave and surge moments will be 90^{o} of phase. A maximum "combined surge and heave" moment of 23.6 x $\sqrt{2}$ = 33 KTM is also considered.

c) Spine Size Analysed

The reference design spine is 14 m in diameter, has a wall thickness of 800 mm and weighs about 80 tonnes per metre. Rather than simply design this one section, it was decided to consider two sections each weighing the same as the actual section used; hence costing approximately the same. Spines of 12.5 m diameter, 900 mm thick and 15 m diameter, 750 mm thick were considered, which neatly bracket the size chosen. Both sections were prestressed to 5 N/mm^2 (after all losses) and were longitudinally reinforced at a rate of 0.4% by cross-sectional area.

d) Materials Used in Spine Design

Concrete

Aggregate - Crushed Rock

Fines - Natural Sand

Cement - Ordinary Portland (partial replacement by flyash permitted)

Minimum cement content 400Kg/m^3
Maximum w/c ratio 0.4.

The concrete to be a designed mix having a characteristic 28 day strength of 50 N/mm^2 and a mean 365 day strength of 63 N/mm^2.

Reinforcing Steel (Rebar)

Reinforcing steel to be cold worked mild steel to BS 4461 having a characteristic yield strength of 425 N/mm^2 and a mean yield strength of 470 N/mm^2.

Prestressing Tendons

Tendons shall be low-relaxation strand to BS 3617/3 having a minimum breaking strength of 1700 N/mm^2 and a mean breaking strength of 1819 N/mm^2.

The design tendon stress after all losses shall be 952 N/mm^2.

Prestressing Bolts (Rigid Joint Only)

Prestressing bolts shall be 50mm diameter Mac-Alloy bolts or similar, to be stressed to 950 N/mm^2 after all losses.

e) Ultimate Design

The ultimate moment of resistance (Mu) was determined in accordance with the Consultants Design Guide using partial factors of safety for materials:-

Concrete	=	1.5
Rebar & Tendons	=	1.5

giving: Mu = 149 KTM for 12.5 m Spine
Mu = 187 KTM for 15.0 m Spine

which ensures a minimum ultimate load factor of 4.5 over the maximum design moment of 33 KTM.

The ultimate moments of resistance "Mu" using mean material properties and partial factors of safety for materials as:-

Concrete	=	1.25
Rebar & Tendons	=	1.0

were: Mu = 166 KTM for 12.5 m Spine
Mu = 207 KTM for 15.0 m Spine.

f) Stress - Moment Curves

To check the fatigue life of the spines, the sections were analysed in accordance with the Consultants Design Guide using partial factors of safety for materials as:-

Concrete	=	1.25
Rebar & Tendons	=	1.15

and the stress/applied moment diagrams were plotted for all 3 materials (see Drawing No. 10103 - 10105).

The analysis was repeated using the mean material properties and partial factors of safety of materials as:-

Concrete	=	1.25
Rebar & Tendons	=	1.0

and the stress/applied moment diagram for concrete was plotted (see Drawing No. 10109).

g) The S - N Curve

The S - N curve specified in the Consultant's Design Guide is based on the Det norske Veritas rule, with the stress threshold of clause D8.1.2 omitted.

The curve has been plotted on Drawing No. 10106, together with various test results (referred to on drawing).

From these results, together with the discussion in the DnV rules, it can be deduced that the design curve from the Consultants' Guide is a "characteristic" one. By "characteristic" is meant those values corresponding to the lower 95% confidence limit. A mean S - N curve is proposed.

h) Miners Number

A Miners Number of 0.2 is specified in the Consultants Design Guide (see also DnV rule). This is believed to be a "characteristic" value and a mean value of 0.6 is proposed based on the figures in tables 4, 6 and 7 of the Van Leeuven & Siemes paper:- "Miners Rule with Respect to Plain Concrete", Heron Vol. 24 1979 No. 1.

i) Wave Climate

The Consultants have specified 399 wave spectra, based on a water depth of 42 m. Since the duck will be located in water 100 m deep these have been corrected to represent that depth, and plotted as a histogram on Drawing No. 10107.

j) Bending Moment - Wave Height Function

Wide tank tests indicate that non-yielding spine RMS bending moments in heave or surge plotted against RMS wave height have an approximately linear relationship. Curves have been plotted for the upper bound values of the experimental results as characteristic bending moment on Fig. 1 of Drawing No. 10108 and for the mean value on Fig 1 of Drawing No. 10110.

k) Rigid-Spine RMS Bending Moment Occurrences

By combining the RMS moment - RMS wave-height graph with the wave climate histogram one can obtain curves showing "non-yielding spine RMS bending moment" occurrences per year (Drawing No. 10107). (See fig. 2 of Drawing Nos. 10108 and 10110.)

l) Spine Moment Histogram

The spine moment histogram (plotted as a table in fig. 4 of Drawing Nos. 10108 and 10110) of actual spine moment occurrences per year, n, is drawn, using the RMS bending moment frequency curves (fig. 2) and assuming a normal distribution of moment exceedence (fig. 3).

m) Characteristic Fatigue Life

Combining the stress range/bending moment tables with the S - N curve (Drawing No. 10106) the number of occurrences to failure 'N' for each moment range is determined (fig. 4 Drawing No. 10108). Hence Miners Sum, defined as n/N for 7 moment ranges is calculated.

The "characteristic" fatigue life of the spine is given by:-

$$\text{Life} = \frac{\text{Miners Number}}{\text{Miners Sum}}$$

Using Miners Number of 0.2 in accordance with the Consultants Guidelines, the "characteristic" fatigue life of the spine = 22 years for 12.5 m spine and 38 years for the 15 m spine. Clearly by interpolation between these limits, the "characteristic" fatigue life of a 14 m spine would exceed 25 years.

n) Mean Fatigue Life

An exercise similar to 2.13 above was carried out to find the mean fatigue life of a spine:-

i) Using mean material properties and mean RMS moments but using the Miners Number and S - N Curve from the Consultant's Guidelines:-

Fatigue Life = 154 years 12.5 m spine
Fatigue Life = 202 years 15.0 m spine.

ii) Using mean material properties, mean RMS moments and proposed mean S - N Curve, with Miners Number of 0.2 gives:-

Fatigue Life = 18700 years 12.5 m spine
Fatigue Life = 27700 years 15.0 m spine

iii) Using mean material properties, mean RMS moments, the proposed mean S -N curve and Miners Number of 0.6 gives:-

Fatigue Life = 56000 years 12.5 m spine
Fatigue Life = 83000 years 15.0 m spine.

Clearly some considerable research must be done before meaningful fatigue calculations are possible.

o) Secondary Reinforcing

The amount of circumferential reinforcement was determined primarily by consideration of the spine in the temporary states during construction.

The amount of reinforcement in the diaphragms and in the joint areas was assessed at 125 kg/m^3, the sections being sized by limiting direct stresses to 5 N/mm^2 and M/bd^2 ratio to 3.0.

p) Shear and Tension

In view of the uncertain nature of the fatigue calculation with respect to beam bending, no fatigue calculations were done for shear.

However the pipe section was assessed for ultimate shear and the resultant load factor (for an uncracked section) = 3.04, compared with 1.4 required by the Consultant's Design Guide.

The principal stresses in the spine wall (ignoring the small wall bending stresses due to differential hydrostatic pressure) are + 5.092 N/mm^2 and - 0.092 N/mm^2.

3. The Duck

a) General Details

The Duck is formed of multiple cell, longitudinally slip-formed units having a minimum wall thickness of 425 mm. The largest cell contains the power pods (2 per 36 m long duck) in an environment of treated fresh water.

b) Materials

Materials are generally similar to those used for the spine, except that the concrete, which shall be grade 45, shall be formed using Lytag aggregate instead of crushed rock. The final saturated (reinforced) density should not exceed 1.9 tonnes/m^3.

c) Design

The duck is designed to withstand an external working pressure of 1.0 atmosphere and an external ultimate pressure of 3.5 atmospheres. The longitudinal prestressing results in an average stress 4 N/mm^2. M/bd^2 ratios are never greater than 3.5 (ultimate).

Time and funding have not permitted a full finite element analysis but the team are confident that a future analysis will justify the present design and will probably allow a reduction in wall thicknesses.

2.4 Mechanical Details

2.4.1 Alternative Spine/Duck Bearings

(Drawing Nos. 10050/51/52.)

The Spine/Duck Bearing proposed utilises a bellows damping system used in connection with bands of magnetic repulsion units. The whole unit is built up on 2.0 m wide strips, which are basically conveyor belting and manufacturers of such have been involved in the overall scheme.

The magnets would be of isotropic ceramic material cast in units 150 mm by 12.5 mm by 12.5 mm thick. The whole bearing is submerged in treated water. Access from the sea is partly restricted. The inhibitor, which could be 3 ppm of chlorine, will prevent any marine growth occuring within the bearing area. The outer section of the bearing comprises a second row of magnets, this time 20 mm wide by 12.5 mm thick, arranged on a diaphragm backed by a series of bellows arranged to act as dampers. The whole bearing has been designed to absorb wide tolerances on circularity of the duck or spine while eliminating any mechanical contact between surfaces resulting in an extremely low coefficient of friction. It will carry a load of 20 kN/m^2. Work at Edinburgh University continues. A paper summarising work to date follows.

It is appreciated that a bearing of this type may well involve a long development period. A more conventional bearing comprising multiple rollers mounted on a series of chains is shown on the drawings as an interim step. We acknowledge Easams Ltd. as the authors of this design.

2.4.2 Spine Joints (Drawing Nos. 10040 & 10041.)

A flexible joint occurs in the spine every 90 metres along its length and this uses a Hooke's joint, comprising a spider with 1.250 m diameter arms and spherical bearings laminated steel and rubber at approx 3 m radius. A series of 8 rams is mounted around the perimeter of the joint. These rams are coupled in pairs, the upper and lower pairs being in opposition to create a heave bending moment and the side to side pairs being coupled to create a surge moment.

The joint is designed to flex at a bending moment of 20,000 ton metres and the rams are operating at a pressure of 6,000 psi with a stroke of 2.2 metres. This stroke enables a movement of $\pm$ 12^{o} to take place at each joint. 5^{o} of this movement is sufficient to accommodate wave motion in both the heave and surge directions. The remaining 7^{o} are available to accomodate the movement necessary for the zig-zag arrangement of the string. The total movement at every joint has the advantage that the points of "zig" or "zag" can be changed from time to time in the surge direction to share the duty throughout the whole line of joints. The entire joint is enclosed and isolated from the sea by a pair of part-spherical housings carrying a rolling seal between them (designed by Andrea Rubber Co.). These seals will protect the complete space occupied by the joint. As a further safeguard similar, but smaller, seals have been incorporated in housings placed immediately around each ram. Alternating with the flexible joint is a rigid connection allowing the removal of spine lengths as necessary. The arrangement of this fixed connection is shown on Drawing No. 10101.

2.5 Moorings

2.5.1 Concept

The concept behind the design of the moorings is to establish a low, constant tension in the rodes.

This is achieved by using a system in which the geometry changes allowing variation of forces with changes of angle, but preserving constant tension (see Drawing No. 10090).

The horizontal load on each Duck is a maximum of 52.4 tons. The configuration is shown on Drawing No. 10090.

Each rode is a continuous member from clump one anchor through both floats and sinkers to the other clump anchor. Bending at the attachment points is being given careful attention. The load in each rode on the wavefront is of the order of 80 tons, a value derived from tank testing. This load is well within the normal capacity of currently used rope. The sinkers adjacent to the 250 tons capacity upstream clump anchor will be 100 tons approx. Downstream the sinkers are smaller being 25 tons submerged weight with a net buoyancy of 25 tons for a downstream buoy. The downstream clump anchor is 120 tons capacity. The system has been tank tested on a bare spine and tension variations have been found to be very low.

The total deflection at any given point of the spine under a freak wave of 35m H_{tc} is less than 2 spine diameters. Low frequency movements are well damped.

2.5.2 Rodes and End Connections

The rodes would be of 2" standard wire rope PVC covered as normally used for anchorage of many floating structures, or of 4" Parafil whichever is considered most suitable at the time of installation.

The end connections will be of suitable shackles. Because of the low angular movement and the constant tension in the ropes achieved by the concept of the mooring, British Ropes state that they are confident that the rope and the end connections would have a life of some 25 years between changes.

2.5.3 Sea Bed Anchorages

Because of the low mooring forces it is not necessary to design any special attachment to the sea bed. The Team was surprised that clump anchors of mass concrete are cheapest and all that is necessary, though the final shape and configuration will be decided to suit actual sea bed conditions. Clump anchors would be equally suitable for the rocky bottom off S. Uist or the sandy bottom of the S. Western approaches.

In suitable bottom conditions, however, a modified version of the Bruce anchor is being considered.

2.5.4 Other Details

The floats or risers to be used within the system appear to be within the current technology and therefore have been given no priority in this up-date.

2.5.5.

Duck Mooring 1981

Introduction

For earlier duck designs our predictions for mooring forces were very low. Indeed they were so low and we had so many other problems to engage our attention that we put no effort into reducing them. The 1981 ducks are bigger. If we apply scaling rules we find that while forces are still small by wave energy standards we can no longer be complacent about them. This note outlines the background and shows that there are good reasons to hope that we can reduce present values by a factor of two or three.

Theory

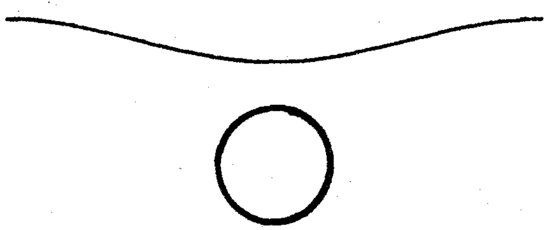
The starting point for mooring design is the fact that in addition to the alternating loads caused by waves and the direct loads caused by currents and winds, there is also a direct load caused by wave momentum. The classical paper is by Longuet-Higgins and Stewart, 1964. It turns out that for the currents in the open Atlantic and for deeply immersed devices like ducks, this is by far the biggest force. The value per metre is

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

$$\rho = 1020\text{kg/m}^3, \qquad g = 9.81\text{m/sec}^2,$$

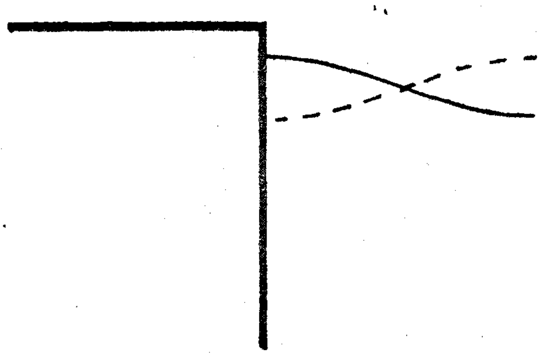
If A = amplitude in metres then F will be in Newtons. But for rough calculatic by busy civil engineers we should note that ρg is almost the same value as the conversion from Newtons to Old tons.

We can understand the interplay of incident reflected and transmitted amplitude by considering the following cases with incident wave amplitude of one metre.



1. A cylinder held rigidly some distance below the surface is transparent to waves. The reflected amplitude is zero and, as no power is absorbed, the incident and transmitted amplitudes are equal. It follows that the mooring force is zero.

$$F = \tfrac{1}{4}\rho g \left[1^2 + 0 - 1^2\right]$$



2. A rigid, infinite, vertical cliff is a perfect reflector. The transmitted amplitude is zero but the reflected amplitude is equal to the incident one.

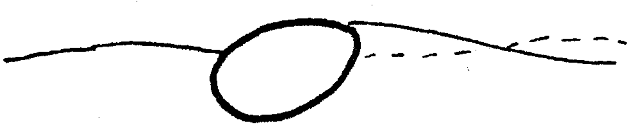
$$F = \tfrac{1}{4}\rho g \left[1^2 + 1^2 - 0\right]$$

$$\simeq \cdot 5 \text{ tons per metre}$$

3. A 'perfect' wave energy device transmits nothing and reflects nothing.

$$F = \tfrac{1}{4}\rho g \left[1^2 + 0 + 0\right]$$

$$\simeq \cdot 25 \text{ tons per metre}$$



4. An imperfect but still linear wave energy device is 80% efficient and loses 5% by reflection and 15% by transmission. These losses will result from a reflected amplitude of ·223 metres and transmitted amplitude of ·387 metres.

$$F = \tfrac{1}{4}\rho g \left[1^2 + \cdot 223^2 - \cdot 387^2\right]$$

$\simeq$ ·225 tons per metre - just less than the case above.

Implications

The lesson so far is clear. We can do nothing to control A incident. But we must reflect as little as possible and make sure that we transmit as much as possible when the incoming power is too big for conversion. It is the tendency to reflect excess power which puts up the mooring forces of OWC terminators. Fortunately for us, ducks tend to transmit excess energy by yeilding in joints and by capsizing.

Now the problem becomes more interesting. Longuet-Higgins and Stewart say nothing about wave *period*. We have so far only discussed linear devices which send off their reflections at the same period as the incident energy. But wave energy devices can be non-linear and radiate energy at harmonics of the incoming periods. When we carried out experiments with cylinders very close to the surface (p23 et seq. of our 1975 report; p17.1 of our 1976 report) we found that there was sometimes a *negative* mooring force. As we were trying to measure the mean value of a large alternating signal and as the same result would arise from a non-linearity in strain gauges we were careful about drawing conclusions. But when we released the cylinder from its mounting it set off steadily towards the wave maker. The phenomenon was explained by Longuet-Higgins in 1977. Wave period is the key.

Suppose that we have to radiate a given amount of power but can use a non-linear mechanism to do so. Non-linearity means that some power will be going as higher harmonics. Power is proportional to amplitude squared times period so that if we reduce period we must *increase* amplitude to keep the power equations right. The cylinders near the surface were breaking up the waves passing above them so that the transmitted energy had its period reduced and its amplitude increased.

In rough conditions the same thing can happen with ducks. We have actually recorded negative mooring forces during some tests. While it is excellent to transmit at shorter periods it is undesirable to reflect harmonics. In small to moderate conditions in the narrow tank we record mooring forces about three times above what we would expect from a linear device of the same power absorption. This must be because we are losing energy from high frequency reflection. There are two escape routes: the first is to modify the shape to make swept volume more linear with angle; the second is to introduce a compensating harmonic control to the damping term. If we can do this we should be able to relate mooring force to extracted power as both are proportional to the square of an amplitude.

Taking 55kw/metre at a period of 8 seconds would lead to mooring forces of about 20 tons per duck. We may be able to reduce this by breaking up the transmitted wave into shorter wavelength components. We can also expect some reduction because of the angular spread of incoming waves: both incident and reflected momentum stresses ought to reduce with incident angle whereas joint yielding at excess bending moments creates stern waves with a relatively narrow angular spread.

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References

Longuet-Higgins, M.S. and Stewart, R.W., "Radiation Stress in Water Waves: a Physical Discussion with Applications", Deep Sea Research 1964 vol 2, pp 526 - 529.

Longuet-Higgins, M.S., "The mean forces on floating or submerged bodies with applications to sand bars and wave-power machines", Proc. Roy. Soc. A 1977 vol 352, pp 463 - 480.

Equations

$$\text{Power} \quad P = \frac{\rho g^2 A^2 T}{8\pi}$$

$$\therefore \quad A^2 = \frac{8\pi P}{\rho g^2 T}$$

For the perfect device the Mooring Force

$$F = \tfrac{1}{4}\rho g \frac{8\pi P}{\rho g^2 T} \quad \text{per metre width}$$

$$= \frac{\text{Power} \cdot 2\pi}{gT} \quad \text{newtons/metre}$$

2.6 Power Conversion System

2.6.1 Primary Power Absorber

The primary interface with the water is the body of the Duck, which is free to rotate about the spine in response to wave action. The profile is to computed profile ref. D.0031 shown on Drawing No. 10102 (see Specification Sheet No. 1).

Normal nodding angles are about one radian and extreme motions about 4 radians. Marine fouling, according to the Scottish Marine Biological Association, will grow to a maximum of 100 mm thick and have a density of 1.3. No reduction in output was produced in model tests with fouling equivalent to 600 mm.

2.6.2 Mechanical Power Conversion

(Drawing No. 10080)

The motion of the Ducks reacts against a reference cage held by the precession of spinning pairs of gyros. This relative motion is used to drive a series of pumps operating on a ring cam. Power can be absorbed by a number of these pumps; the number selected should be chosen to provide damping to match the sea state. The output from these pumps is fed through swashplate pump/motors to the flywheels of the gyroscopes. A parallel take off feeds a further swashplate pump motor unit driving the synchronous generator at the output of the unit.

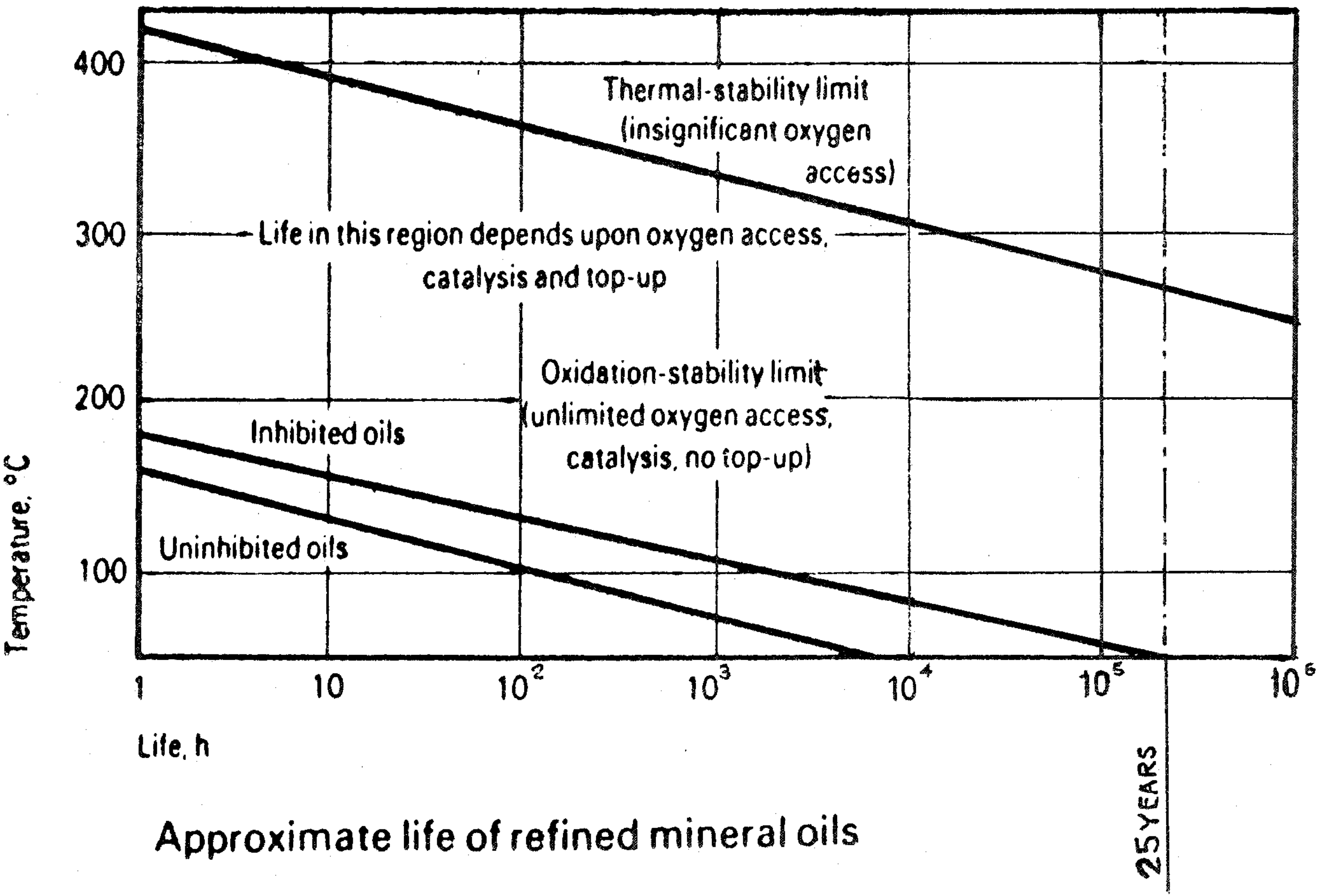
The swashplate angles on the pump/motor unit driving the flywheels vary from positive to negative at twice the wave frequency (approx. 1 per 5 secs). When there is surplus energy it is fed into the flywheels to speed them up and when there is a lack of energy this surplus may be drawn from flywheels thus ensuring a constant supply to the motor unit driving the generator. The whole of this arrangement is mounted within the low pressure area contained within the canister. The speed of the flywheel will only vary by 2 RPM total for a 1500 RPM nominal speed.

The pressure within the canister is below 5 torr ensuring that windage loss for each flywheel is limited to 7 kW. The working fluid is a special, narrow cut, low viscosity, mineral oil containing an antiwear or EP additive, which has a boiling range typically from 260 to 335^{o}C and a viscosity of approx. 21 cst at 8^{o} C. The vapour pressure even at 100^{o}C of the product is reasonably low. With a system pressure of 5 mm Hg the oil makes no significant contribution to the pressure. The oil recommended by B.P. Oil Limited is priced at 38p per litre and is estimated to have a life well above 25 years. The hydraulic system has a return pressure of 50 p.s.i.; this pressure is achieved with the use of an eductor type pump (i.e. a pump with no moving parts) and the pump is submerged to a depth of 350 mm which is sufficient to keep it operating without cavitation.

A proportion of the oil is bypassed through the flywheel shafts and subjected to 1000 G for 180 seconds transit time thus removing all metallic particles and other debris. The absence of oxygen greatly extends the life of the oil. A chart extracted from "Engineering" Tech. File No. 93 illustrates this (Fig. 1 Page 27) and shows a permissible maximum operating temperature of 250^{o} C for a 25 year life. Excessive heat is extracted by a heat exchanger mounted at one end of the vessel. The appropriate specification sheets are appended to this paragraph.

2.6.2. Mechanical Power Conversion

BASED ON CURVE IN "ENGINEERING"
TECHNICAL FILE NO. 93 (SEPT 1981)



Approximate life of refined mineral oils

WITH ACKNOWLEDGEMENTS TO T.I. FOWLE

FIGURE 1.

DUCK SPECIFICATION

	Unit	Value at 30.9.81			
Profile shape	-	D0030			
Base diameter	m	14			
Hub depth	m	9.1			
Water line length	m	13.8			
Hydrodynamic width	m	45			
Physical width	m	37			
Torque limit	Nm	22.5×10^6			
Power limit mean long term at generator output	MW	2.25			
Power limit instantaneous at 3200psi Δ	MW	17.8			
Duck angular velocity	rad/Sec	.75			
Survival depth	m	30			
Canister inside diameter	m	5.5			
Power density limit at sea interface	kW/m	57.5			
Power density at duck terminal ($\eta = .87$)	kW/m	50			
Power density at Skye terminal ($\eta = .87 \times .97$)	kW/m	48.5			
Input power limit instantaneous at 4800 psi	MW	26.25			
Overload output (20 mins)	MW	3.5			

RING CAM SPECIFICATION

		Unit	Value at 30.9.81			
Number of gyros		-	4			
Ring cams per gyro		-	2			
Cam ring diameter mean		m	4.375			
Lobes per face (2 faces)		-	27			
Followers per face		-	32			
Width of cam face		mm	125			
Mean height of over cam lobes		mm	236.5			
Weight of cam ring		kg	3250			
Lobe wavelength mean at ring centre line		mm	509			
Rise of cam trough to crest		mm	63.5			
Line force at a nominal pressure	(38,800 lb)	kN	172.5			
Roller diameter		mm	200			
Maximum slope tangent	(28°)	rad	0.436			
Minimum crest curvature radius	Corrected arc	mm	382.2			
Maximum roller acceleration		M/sec^2				
Line stress	(7880 lb/in)	N/m	1.38×10^6			
Boost pressure to prevent lift	(5 psi = 10% nominal boost)	N/m^2	34.5×10^3			
Maximum angular velocity		rad/Sec	.75			

PRIMARY HYDRAULIC SPECIFICATION

		Unit	Value at 30.9.81			
Number of lobe stations per cam ring	(2 lobes per station)	-	27			
Number of pumping stations per cam	(2 cylinders per station)	-	32			
Number of ring cams per gyro	(4 gyros per duck)	-	2			
Primary cylinders per duck	(2 x 32 x 2 x 4)	-	512			
Bore		mm	70			
Stroke		mm	127			
Nominal radial clearance	(.001")	mm	.025			
Volume per cylinder stroke	(489cc = 29.83in^3)	m^3	489x10^{-6}			
Volume per cam revolution per each ring cam pump	(2x489x10^{-6}x32x27)	m^3	0.845			
Strokes per second per MW		Hz	91			
Working pressure	(3250 psi)	N/m^2	22.4x10^6			
Boost pressure	(50 psi)	N/m^2	345x10^3			
Power strokes per cylinder in 25 years		-	175x10^6			
Oil viscosity	(21 centipoise ≃ 3 μ Reyn)	NSec/m^2	.021			
Max operating rate of roller followers		strokes/sec	3.22			
Max duck torque with 85% rollers at 3200 psi		Nm	20.2x10^6			

SECONDARY HYDRAULICS SPECIFICATION

	Unit	Value at 30.9.81			
Number of gyro drive units	-	8			
Number of generator drive units	-	2			
Displacement per motor revolution at maximum swash	m^3	3.23×10^{-3}			
Maximum flow into generators at 1500 rpm	m^3/Sec	161.5×10^{-3}			
Maximum flow into gyros at 1500 rpm	m^3/Sec	646×10^{-3}			
Maximum total instantaneous flow	m^3/Sec	807×10^{-3}			
Maximum instantaneous power (at 3200 psi) ($22.06 \times 10^6 N/m^2$)	MW	17.8			
Mean flow at full power	m^3/Sec	.091			
Response time: zero to full flow	mSec	120			
Response time: normal to emergency pressure	mSec	10			
Ratio of instantaneous maximum to mean maximum	-	8.9			

GYRO DISC SPECIFICATION

	Unit	Value at 30.9.81			
Number of gyros	-	4			
Diameter at rim	m	3.05			
Diameter of hole	m	1.0			
Disc thickness say 100 laminations x 6 mm	mm	600			
Weight of disc	kg	30,713			
Weight of total rotating assembly extras are 3220 for shaft, 940 clamps	kg	34,624			
Moment of inertia (each gyro) " " 1910 " " 246 clamps	kgm^2	41,692			
Nominal working speed	rad/sec	157			
Maximum stress at 1500 rpm (12 tons in^2)	N/m^2	$185x10^6$			
Stored energy per gyro at 1500 rpm	J	$514x10^6$			
Stored energy per gyro at maximum storage speed (2000 rpm)	J	$913x10^6$			
Storage time at maximum output from 1500 rpm to 375 rpm	Sec	857			
Maximum bearing force	N	$1.66x10^6$			
Bearing diameter (nominal)	mm	280			
Bearing length (nominal)	mm	457			
Projected bearing area	m^2	0.128			
Nominal bearing clearance radial (.001")	mm	0.025			
Maximum precession angular velocity at mid point and 1500 rpm	rad/sec	.85			
Iω (whole duck) at 1500 rpm	kgm^2/sec	$26.2x10^6$			

HIGH SPEED GYRO BEARINGS SPECIFICATION

		Unit	Value at 30.9.81			
Speed normal		rpm	1500			
Speed max		rpm	2000			
Maximum radial force		N	1.66×10^{6}			
Axial separation of bearing couple		m	3.7			
Outside diameter		mm	280			
Length		mm	457			
Projected pocket area effective		m^2	.123			
Shear stress	(2.04 tsi)	N/m^2	44.7×10^{6}			
Bending stress	(14.2tons/in^2)	N/m^2	220×10^{6}			
Deflection	(.011 inch)	mm	.27			
Ovality at max. load						
Radial land clearance	(.001")	mm	.025			
Oil viscosity	(21 centipoise)	$NSec/m^2$	.021			
Inside diameter		mm	165			
Bearing source pressure from eductor	(2250 psi)	N/m^2	15.5×10^{6}			
Full load high side pocket pressure	(2067 psi)	N/m^2	14.3×10^{6}			
Full load low side pocket pressure	(110 psi)	N/m^2	758×10^{3}			
Off load high side pocket pressure	(602 psi)	N/m^2	4.15×10^{6}			
Off load low side pocket pressure	(435 psi)	N/m^2	3.0×10^{6}			

OIL SPECIFICATION

		Unit	Value at 30.9.81			
Total quantity	(3000 galls)	m^3	15			
Viscosity at $20^{\circ}C$	(21 centipoise ≈ 3μ Reyn)	$Nsec/m^2$	0.021			
Viscosity index						
Cooling sump temperature		$^{\circ}C$	30			
Maximum temperature rise at return point		$^{\circ}C$	15			
Maximum pressure normal	(3250 psi)	N/m^2	22.4×10^6			
Maximum emergency	(4800 psi)	N/m^2	33.1×10^6			
On gyro eductor output	(2250 psi)	N/m^2	15.5×10^6			
Computing pressure	(500 psi)	N/m^2	3.45×10^6			
Boost pressure	(50 psi)	N/m^2	345×10^3			
Eductor input head relative to vacuum		m	.35			
Vacuum pressure		mmHg	1 to 5			
Oil vapour pressure at $100^{\circ}C$						
Residual gas			CO_2			
Filtration acceleration	(989g)	km/sec^2	9.7			
Filter depth radial		mm	75			
Transit time		sec	75			
Filter throughput per gyro	(110 gpm)	m^3/sec	8.3×10^{-3}			
Proportion of flow filtered		%	11			
[illegible] at full power		sec	165			

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2.6.3 Power Canister

The present costings are based on conventional pressure vessel design for this item although the team are considering concrete alternatives with a steel inner membrane.

Advice on leakage rates and production methods was sought of British Oxygen Co. (Cryoplant Ltd). Their standard leakage rate is 1×10^{-6} torr litre/sec. and the accompanying calculation is based on this production leakage rate which we are told applies to quite complicated arrangements of branches and nozzles. Our power canister will have no more than 6 individual branches. The canister is designed against an external pressure of 3 atmospheres and is fitted with a frangible disc that will implode at this pressure in the event of excessive submergence thus avoiding a total collapse of the canister onto the machinery inside.

2.6.4 Electrical Generation

(Drawing No. 10071.)
The 1981 wave energy device differs significantly in some respects from the 1979 device which was also based upon a.c. synchronous generation. There are now two generators per duck and energy is also generated from the movement at the flexible joints of the spine. An intermediate transformation from 3.3 kV to 33 kV has been included to reduce the number and cost of the cables down the spine.

2.6.3 Power Canister - Leak Rates - 1981 Reference Design

Power Canister is 5.5m dia x 18.25m long

$$\text{Volume} = \frac{\pi\, 5.5^2}{4} \times 18.25 = 433.59\text{m}^3$$

$$= \underline{\underline{433590 \text{ litres}}}$$

Leak rate permissible for 5 Torr rise in 25 years

$$= \frac{433590 \times 5}{25 \times 8766 \times 3600} = \underline{\underline{2.75 \times 10^{-3} \text{ Torr litres/sec}}}$$

Standard leakage allowed by BOC $= 1 \times 10^{-6}$ Torr litres/sec.
(Cryoplant Limited on Cryogenic Vessels)

Permissible no of leaks to achieve standard leak rate for vessel is

$$\frac{2.75 \times 10^{-3}}{1 \times 10^{-6}} = \underline{\underline{2750}}$$

Out-gassing does not become a problem above 10^{-4} Torr or 1/27th of the pressure we are using.

N.B. 1 Torr litres/sec = 2120 micron ft^3/min = 78.97 atm cm^3/min

P.B.WILLIAMS
JOHN LAING

The collection and transmission scheme described gives the best cost of alternatives studied. A 275 kV cable transmission from the spine to shore was included in the alternative schemes studied.

Drawing No. 10021 shows the geographical location of the device array, which is arranged in eight groups with 1 km separation.

2.6.5 Generators

Further enquiries with a number of manufacturers have enabled us to reduce the cost per kW of the generators to £25 per kW compared with £35.5 per kW used in the 1979 report even though the size has been reduced from 2.4 MW to 1.2 MW. Even so, the cost, at first sight, may appear high compared with prices for machines of this size advertised by manufacturers of standby generators which are available at £9 per kW.

Such machines are not available with 3.3 kV windings. The cost of the necessary transformer to raise from 415 V (about £4 per kW) offsets some of the cost advantage of using 415 V machines.

The need for many years of unattended running necessitates the best bearing lubrication system which further increases the cost. Each machine will be capable of generating at 0.8 pf lag and 0.85 pf load. An additional £10,000 has been allowed to cover the cost of the circuit breaker and the control and monitoring equipment.

POWER CANISTER ELECTRICAL EQUIPMENT	Unit	Value at 30.9.81	
Generator			
Rated output	kW	1200	
Rated power factor		0.85 lag	
Terminal voltage	kV	3.3	
Rated frequency	Hz	50	
Rated speed	rev/min	1500	
Enclosure classification (Drip-proof and screen protected)		IP22	
Cooling classification (Self-ventilated open machine) (Fan mounted on shaft)		IC01	
Mounting		Horizontal	
Bearings (End shield mounted)		Plain sleeve, oil ring lubricated	
Overall dimensions (L x W x H)	mm	2000 x 1000 x 1405	
Weight	t	3.9	
Excitation system		Brushless	
Excitation power supply		Shaft mounted PMG pilot exciter	

POWER CANISTER ELECTRICAL EQUIPMENT (CONTINUED)	Unit	Value at 30.9.81	
Auxiliary Power Transformer			
Type		Resin encapsulated	
Rating (3 phase/50Hz)	kVA	100	
Voltage ratio	V/V	3300/433	
Cooling		Natural air	
Overall dimensions (L x W x H)	mm	900 x 660 x 1120	
Weight	t	0.5	
3.3kV Switchgear			
Type		Vacuum interrupter	
Rated continuous current	A	250	
Fault current rating	kA	1.6	
Voltage transformers (For protection, metering, synchronising and AVR requirments)		2 x 3300/110V	
Current transformers (For protection, metering and AVR requirements)		2 x three phase sets 250/5A	
Overall dimensions (L x W x H)	mm	320 x 1375 x 1905	
Weight	t	0.4	

POWER CANISTER ELECTRICAL EQUIPMENT (CONTINUED)	Unit	Value at 30.9.81	
Battery/Charger Unit			
Battery type		NiCad	
Nominal battery voltage	V	50	
Capacity of battery (at 10 hour rate)	Ah	50	
Overall dimensions (L x W x H)	mm	970 x 430 x 1295	
Weight	t	0.15	
Dormancy Pump Motor			
Rated output	kW	1.5	
Rated voltage (3 phase, 50Hz)	V	415	
Rated current	A	3.5	
Rated speed	rev/min	1420	
Enclosure classification (Totally enclosed)		IP44	
Cooling classification (Oil drip over externally) (ribbed frame.)		IC0141	
Overall dimensions (L x W x H)	mm	295 x 185 x 185	
Weight	kg	17	

POWER CANISTER ELECTRICAL EQUIPMENT (CONTINUED)	Unit	Value at 30.9.81	
Scoop Pump Motor			
Rated output	kW	4	
Rated voltage (3 phase, 50Hz)	V	415	
Rated current	A	9.8	
Rated speed	rev/min	710	
Enclosure classification (Totally enclosed)		IP44	
Cooling classification (Oil drip over externally) (ribbed frame.)		IC0141	
Overall dimentions (L x W x H)	mm	595 x 315 x 370	
Weight	kg	81	
Control Cubicle housing the following			
AVR and Excitation Control Equipment			
Autosynchronizer			
Protection Relays			
kW/kVAr Control Equipment			
Metering Transducers			
Instrumentation Transducers			
Overall dimensions (L x W x H)	mm	500 x 500 x 1500	
Weight	kg	50	

HEAT LOSSES FROM ELECTRICAL EQUIPMENT

	Unit	Value at 30.9.81	
Power Canister			
Generator	kW	$37 + 29\left(\frac{P}{1125}\right)^2$ where P=actual output in kW	
Auxiliary Power Transformer	kW	$0.44 + 1.7\left(\frac{S}{100}\right)^2$ where S=actual output in kVA	
Control Cubicle	kW	0.5	
Black-Start Pump Motors			
Dormancy	kW	0.5	
Scoop	kW	0.88	
Maximum losses to dissipate	kW	67	
Minimum losses to dissipate	kW	1.14	
	kW	2.54 (With 'black-start' pump running)	

SPINE SUB-GROUP ELECTRICAL EQUIPMENT	Unit	Value at 30.9.81	
Generator			
same as Canister Generator except for:			
Enclosure classification (Totally enclosed)		IP54	
Cooling classification (Closed air circuit, water cooled (with separately mounted heat exchanger)		ICW37A91	
Overall dimensions (L x W x H)	mm	2200 x 1200 x 1700	
Weight	t	4.5	
Sub-Group Transformer			
Type		Oil filled	
Rating	MVA	20	
Voltage ratio	kV	3.3/3.3/33	
Cooling		Water	
Overall dimensions (L x W x H)	mm	4000 x 3500 x 3500	
Weight	t	50	

SPINE GROUP ELECTRICAL EQUIPMENT	Unit	Value at 30.9.81	
Group Transformer			
Type		Oil filled	
Rating	MVA	320	
Voltage ratio	kV	33/33/132	
Cooling		Water	
Overall dimensions (L x W x H)	mm	8500 x 4500 x 5000	
Weight	t	250	
33 kV Switchgear			
Arrangement (2 x 5 panel)		Switchboard	
Type		Vacuum or SF_6	
Rated continuous current	A	320	
Fault current rating	kA	31.5	
Voltage transformers (For protection requirements)		2 x 33/0.110 kV (1 per 33kV bus)	
Current transformers (For protection requirements)		1 x 3 phase set per circuit 300/5A	
Overall dimensions (Length x Width x Height)	mm	5000 x 2000 x 2500	
Weight	t	10	

SPINE GROUP ELECTRICAL EQUIPMENT (CONTINUED)	Unit	Value at 30.9.81	
132 kV Switchgear			
Arrangement		4 Switch mesh	
Type		SF_6 metal enclosed	
Rated continuous current	A	1600	
Fault current rating	kA	40	
Overall dimensions (L x W x H)	mm	6000 x 5500 x 5000	
Weight	t	40	

2.6.6 Operational Control Systems

The control system for each Duck is based on micro-processors. Accelerometers within the pressure vessel measure the duck motion and feed this information to the micro-processors which in turn set the levels of activity of certain functions, the main ones of which are:-

(i) The Enabling or Inlet Valve of the ring cam pump cylinders. The secondary circuits will inhibit this initiation if they have previously sensed a chipped lobe, a cracked cam roller or any other mechanical damage, all of which can be sensed by a strategically placed piezo-electric crystals. Normal output will not be affected by the failure of 15% of the cylinders.

(ii) Torque of the hydraulic units is the product of supply pressure and swash plate angle. This angle is controlled by a hydraulic computing valve within the unit itself. The control pressure will be a parameter which is set by the rate of gyro spin.

The major requirement is that the hydraulic units associated with the flywheels shall have their swashplates moving gently from positive to negative in sympathy with the wave arrival periods. This oscillation will cause a swing in the RPM of the main flywheel disc from 1499 to 1501 and will allow an output from the main alternator driven by a further swashplate unit, synchronised with the grid at a nominal 50 Hz.

(iii) Auxiliary circuits associated with the micro-processor will monitor such parameters as oil and temperature, oil flow, vacuum levels in the power canisters and the other parameters required for the successful operation of the device. The micro-processors will be connected to an on-board computer. This computer will normally store information for a period (say ½ hr.) before transmitting ashore but will also have a priority system such that any parameter exceeding its built-in limit will be notified immediately.

The team is anticipating that fibre optic channels built into the main undersea cables will be the best technique for signal transmission to and from shore and between units.

Duplicate or triplicate methods of condition monitoring will be built into the on-board computer system to enable reliable information to be sent ashore.

The system design will not require wide band width communications. Most decision making will take place on board.

(iv) Detailed design of the system has not been pursued as the rate of advance of this technology is so rapid that any system designed today would be out-of-date when it is needed even if this were only in 3 years time.

2.7 Auxillary Plant and Equipment

2.7.1 Auxillary Power Systems

Each duck power canister needs electrical power of 6 kW to start from 'cold'. This power would normally be drawn from operating units but could be supplied from a ship borne generator, or from land.

Heating, lighting and ventilation have been generally considered but not designed in detail at this stage of development.

The duck string will be provided with navigation lighting, radar reflectors and any other devices thought necessary by the appropriate authority.

2.8 Power Collection and Transmission

2.8.1 Power Collection - Sub-Grouping

The most economical arrangement for the sub-grouping is shown in Fig 3 of Drawing No. 10071. The cable length and therefore the costs have been kept short. 33 kV is chosen as the intermediate voltage as the power transformers and smaller quantity of cables at this voltage gives the best cost.

The 22 kV scheme was approximately 10 per cent more expensive for this sub-grouping.

Where the cables pass through watertight bulkheads epoxy resin bushing spouts, similar to those used on switchgear has been included. For 33 kV this would be one per phase per cable and for 3.3 kV we envisage a three-phase moulding. These 33 kV and the 3.3 kV barrier bushings of these types can be accommodated in a 1 m diameter barrier.

Each sub-group will have 12 duck and 1 spine generator connected to a 3.3/33 kV 20 MVA transformer, which will be accommodated in the spine. Each generator cable will be fuse protected adjacent to the transformer whilst the transformer and its 33 kV cable will be switched by a circuit breaker adjacent to the main group transformer.

Each sub-group will occupy 270m of spine and is rated 15.6 MW.

2.8.2 Main Groups

Each main group will comprise 18 sub-groups, 9 to each low voltage winding of the 300 MVA, 33/132 kV transformer. To reduce the number (and cost) of the 33 kV cables and switchgear of the sub-groups, they will be paired on a 400 mm^2 cable, with the extension to the outer sub-group of the pair having 185 mm^2 cable size.

Vacuum or SF_6 33 kV circuit breakers are available. These have small operating energy requirements and require maintenance not less than every 10 years. A 100 kVA transformer will provide the necessary switchgear energy requirements (and charge a battery) and the circulating water cooling for the main transformer.

A four switch mesh-connected 132 kV switchboard adjacent to the transformer will bus together the main transformer, the spine-to-shore cable and the interconnecting cables between the adjacent main groups. The outer main groups will not require this switchgear (as indicated in Fig 1 of Drawing No. 10071).

The electrical arrangement of these main groups is shown in Fig 2 (Drawing No. 10071.)

Each main group is rated 280 MW and occupies 4.86 km of spine.

2.8.3 Power Transmission

Each main group is assumed to be separated by 1 km to allow ship traffic and facilitate towing and installation of each main group spine.

Six 132 kV 1600 mm^2 3 core cable circuits will extend between the spines and shore, each capable of carrying the maximum power of 1 main groups. A submarine cable inter-connection along the spines will facilitate lateral power flow and enable full power to be generated with one main cable out of service. The revenue from 280 MW for 6 months at 80 per cent load factor, which would be lost if there was no alternative cable route, would cover the cost of providing the interconnection. At lower than full power transfer, some of the main cables can be switched out to reduce the charging current which will in turn assist the voltage control and reduce losses.

The estimated costs of the submarine cables is based on world competitive cable prices and laying costs advised by Wharton Williams.

They are, in our opinion, a reasonable estimate although the amount of submarine cable supplied in recent years does not allow the same confidence in cable costs as we have in the other cost estimates.

The 132 kV main cables will be grouped to two - 1 circuit breaker switching stations on the west coast of S. Uist. Two double circuit 132 kV overhead lines from each of these stations to the main transforming station on the east coast will carry the full power with one line from each station out of service.

At the main transforming station a 1 circuit breaker switching arrangements have been chosen for both 132 kV and 400 kV as it has very high reliability and is only 10 per cent more costly than the less reliable traditional switching arrangement (1 circuit breaker, double busbar).

Three 400 kV - 1300 mm^2 submarine cable circuits are included for the crossing to Skye. This arrangement was suggested by the WESC Consultants. Full power can be transmitted with one circuit out.

It is believed that XLPE cables will be sufficiently developed by the end of this decade and propose their use. The much lower charging current of this type of cable makes the transmission over larger distances possible and reduces the amount of short reactance compensation that is necessary to keep the voltage profile within reasonable limits.

Three 100 MVAr shunt reactors at the main transforming station and one 50 MVAr shunt reactor at each station on the west coast of S.Uist, have been included to adequately compensate for the charging current.

2.8.4 Control and Monitoring of Electrical Transmission

No attempt has been made to carry out a study of the detailed requirements for remotely controlling and monitoring the transmission equipment. The technological advances in electronics are rapidly increasing facilities and reducing costs. It is not believe that it is productive to study a control scheme now, for adoption in 5 to 10 years time.

To provide the necessary control and monitoring facilities for the electrical generation and transmission switchgear, we have included in the costs of the generators £3m; for duplicated cabling £5m, and for a control centre £4m. It is believed that this total amount, £12m, is adequate to provide full and reliable facilities.

2.8.5 General Comments

The scheme envisaged and costed is based upon the use of presently available equipment and ratings with the exception of the submarine cabling which historically is always in some respects new in design.

Experience and nominal calculations confirm the electrical arrangement is workable and would at the worst require only minor changes for it to be applied.

We foresee that the continual flexing of the 132 kV cables between the spine and the sea bed will provoke a major criticism. The design of the cable to withstand flexing must be based either upon the materials being sufficiently flexible or by mechanically restricting the flexing to a safe level. The material stresses due to flexing are proportional to the deflection and a way of restricting this is enclosing the cable in a thick walled, metal reinforced rubber or plastic tube, the bending radius of which is sufficiently large. Such tubes are now made and consideration shall be given to extending their characteristcs to meet the requirements of the use of normal construction cables, or cables only slightly different to normal, between the spine and the sea bed.

Figs 5 and 6 of Drawing No. 10072 show the arrangement of the bushings in a barrier in a typical 33 kV bushing. Fig 7 of Drawing No. 10072 is included to indicate the generator instrumentation and control envisaged.

The overall cost of the 1981 arrangement, which delivers 2000 MW to Skye, is less than the equivalent 1979 arrangement. The further period of study has reduced the cost by more than inflation in the intervening periods.

SECTION 3

STATE OF DEVELOPMENT

3. STATE OF DEVELOPMENT

3.1 Structures

3.1.1 Spine Design Philosophy

The spine is the main structural member, resisting deflection by wave action in small to medium seas in order to achieve efficient energy absorption by the ducks. In large seas, efficiency can be sacrificed in the interests of survival of the spine; the use of moment-controlled joints ensures the limitation of the bending moments to such values as may be resisted by a spine the size of which has been selected for other reasons, typically the required duck size for power absorption, and for buoyancy.

The critical element in the design of the spine is the need to survive, not excessively large moments, but the continual reversal of moments, or, more precisely, the continual cycling of the bending moment axis around the longitudinal spine axis as the spine section orbits in an elliptical path in the vertical plane, due to the passage of a wave. This cycling takes place approximately 90×10^6 times in a period of 25 years, of which over 50% is at the maximum design moment, i.e. at the selected value at which the spine joint will yield.

The critical limit state is therefore the fatigue life. Too little is known about the fatigue behaviour of concrete or concrete members, particularly at load repetitions in excess of 10^6 and such information as is available is from cyclical compression tests on small plain concrete cylinders tested at fairly high frequency, typically 1 - 6 Hz. It appears that the design rules of "Det norske Veritas" which have been incorporated in the Consultant's Draft Design Guidelines, are based on lower bound values of fatigue life derived from such tests. They are, of course, also intended for single commercially-vital, heavily populated structures. These rules have been applied in the present design of the spine, but the following matters have been considered as being significant:-

(i) Whether linear extrapolation of the S-log N line given by DnV into the $N = 10^8$ region is justified by test results.

(ii) Whether the use of lower bound values for plotting the S-log N line and in establishing a value for the Miner's Number will give an appropriate fatigue life for the wave energy installation as a whole.

(iii) Whether the concept of a characteristic fatigue life, namely one which only a small proportion of spine units will fail to meet, is appropriate to an unmanned installation comprising in excess of 400 units (in each of perhaps several"power stations").

If the characteristic fatigue life has been pitched at the level of the 95% confidence limit, this would imply about 20 units failing over a period of 25 years, which is a high degree of reliability in which is essentially a mechanical component of the power station.

(iv) It has seemed desirable to consider the possibility of carrying out a Level II Probabilistic Design (ref: CEB-FIP Model Code), namely to establish the failure rate of spine units which would be commercially acceptable and from the statistical spread of load and material characteristics to determine the necessary mean fatigue life for design.

Limited time and funding has made it impossible to follow through, fully, the consequences of these considerations. However, as well as checking the reference designs, in accordance with the Consultants Design Guidelines, the mean fatigue life of these designs has been estimated, i.e. the expected life with a 50% failure rate. To do this, the mean values of the various characteristics have been projected from the experimental results which are available. (See Appendix.)

The resulting lives are as follows:-

Life in accordance with rules:

= approx. 30 years

Mean life (50% failure rate):

= approx. 72,000 years

Without reference to commercial considerations it would, further, be possible to project the acceptable characteristic life (say, with a 5% failure rate), with assumptions being made about the standard deviations of the various design parameters, where they are not actually available. The analysis is, however, too time consuming for the present, and it is considered that a comparison between the outcome of the Design Guidelines and the mean life will be sufficient at this stage to show that Level II Probabilistic Design methods need to be further examined.

It is considered that at a later stage in development it will be necessary to carry out a finite element analysis of the spine to take account of all combinations of forces upon the spine, as well as secondary effects such as local bending moments near bulkheads and so forth. For the present, sufficient simple checks have been made of the major effects.

3.1.2 Duck Design Philosophy

The duck is required to carry the power take-off machinery in its sealed "pod", and to convert wave action into rotational oscillations of the duck, centred on the spine.

The wave forces on the outer envelope are transmitted into the bearings of the gyroscope flywheels until the torque limit imposed by electrical considerations is reached; above this limit, wave forces are transmitted to the sea to leeward, very little load being transmitted through the duck/spine bearing.

The duck body, of cellular form with curved interstitial walls, is well able to deal with the generally distributed forces of low intensity, and the locally imposed point or line loads from the power pod will be located at wall intersections where they can be quickly dispersed. The main structural design consideration is a requirement for the duck body to survive submergence to 30 metre depth in an extreme wave and the intersititial walls are designed to resist the differential pressure as reinforced concrete sections.

Racking of the whole duck body will occur from the possible obliquity of wavefronts acting on the 36 metres wide duck; the torsional moment caused is not accurately known but it is very likely that the degree of longitudinal prestress which is desirable for other reasons will be sufficient to prevent excessive principal tensile stresses arising.

3.1.3 Tank Testing

The techniques and results of tank testing, including the co-relation of narrow and wide tanks will be the subject of separate reports issued by Edinburgh University from time to time. It should be noted, however, that productivity is based on the testing of 1/140th scale models in sea states comprising 399 spectra for the 42m site and synthesised by John Crabb of I.O.S. corrected for a site in 100 m depth (see section 7.1.2). Extreme seas have been generated to simulate the 100 year wave and survival against such a sea state repeatedly established.

3.1.4 Narrow Tank to Wide Tank Comparison

Appended hereto.

3.1.4.

NARROW TANK TO WIDE TANK COMPARISON

Objectives

We wanted to check that the wide tank duck models worked properly and that there were no unexpected results from adding an extra dimension.

We wanted to check directional characteristics.

The models

Narrow tank models have moved ahead of the design which was selected for mass production for the wide tank. This meant we had no current results for comparison purposes. We went back to D0022, which was a variable width design used to measure gap sensitivity. We added end-plates to produce the same proportion of gap to duck as the wide tank equivalent. (This was .311 to 1.) The shape and proportions were identical but the back diameter was only 100mm.

We mounted this model on a *fixed* axis in the narrow tank and measured its efficiency as a function of frequency. The wave amplitude was held constant at .54cm. Efficiency was calculated on the basis of the power out divided by the power in the whole width of the tank. The results are plotted in the solid black curve in Fig. 1.

(Note that this model does not have the soft corners and bulbous front of the later designs. It does not exploit the compliant mounting effects which help in long waves. Its performance is therefore considerably below that used for 1981 output predictions.)

For the wide tank replication we used six ducks with 130mm back diameter. Their width was 305mm and they were pitched at intervals of 400mm. Their angular velocities were measured by a tacho generator on the end of a belt driven brush motor which supplied power take-off torque. This signal could be corroborated by the output from an integral fringe counting shaft encoder and is therefore very accurately known.

The torque calibration is less satisfactory because the duck has seven water immersed ball bearings for its location to the spine plus two more for belt pulleys.

After torque measurements in the water (which cancelled spine weight) and out of the water in vertical and horizontal attitudes we settled for vertical axis calibrations in air with a specially selected thrust race taking the vertical weight of the spine section. This method does not produce the losses in radial bearings caused by wave-loading and is therefore conservative.

Torque was calibrated by holding the duck body stationary and driving the spine inside it through a strain gauge torque cell and frictionless universal joints. Calibrations were repeated for several angles. Fig. 2 shows the amount of lost torque as a function of velocity for a group of ducks. The small amount of scatter allows a corrected torque signal to be computed from the known velocity.

The electronically controlled hinges of the spine sections were locked rigidly and the group of six were mounted on a rigid submerged frame. The hub depth was adjusted to the value corresponding to that of the narrow tank.

Wave measurement

A group of 10 conductivity compensated gauges were placed in a line along the direction of wave propagation at the position of each duck in turn. Measurements were made in the absence of the model, and then an average taken all readings.

Results

Wide tank results with corrected torque readings are superimposed on the narrow tank curve in Fig. 1. (Two period scales are necessary because of the size increase.) The adjustment maintains the same ratio of duck diameter to deep water wavelength. Efficiency is calculated from the mean output of all six ducks divided by the power in the total sea front occupied.

Fig. 3 shows the effect of moving the mounting frame to various angles to the original wave front. Also plotted are the values that would be predicted by a cosine law operating on the efficiency at 0°.

Conclusions

The cosine rule is a reasonable approximation but perhaps conservative for large angles.

Narrow tank regular wave fixed axis results can be transferred to the wide tank.

FIG 1 / COMPARISON OF NARROW & WIDE TANK EFFICIENCIES.

SOLID LINE = NARROW TANK PERFORMANCE

Δ = MEASURED WIDE TANK RESULTS.

EFFICIENCY IS AVERAGE OF 6 DUCKS IN WIDE TANK.

(SEE "WIDE TANK DUCK LOG", pp 272-278)

7-12-81 DCJ

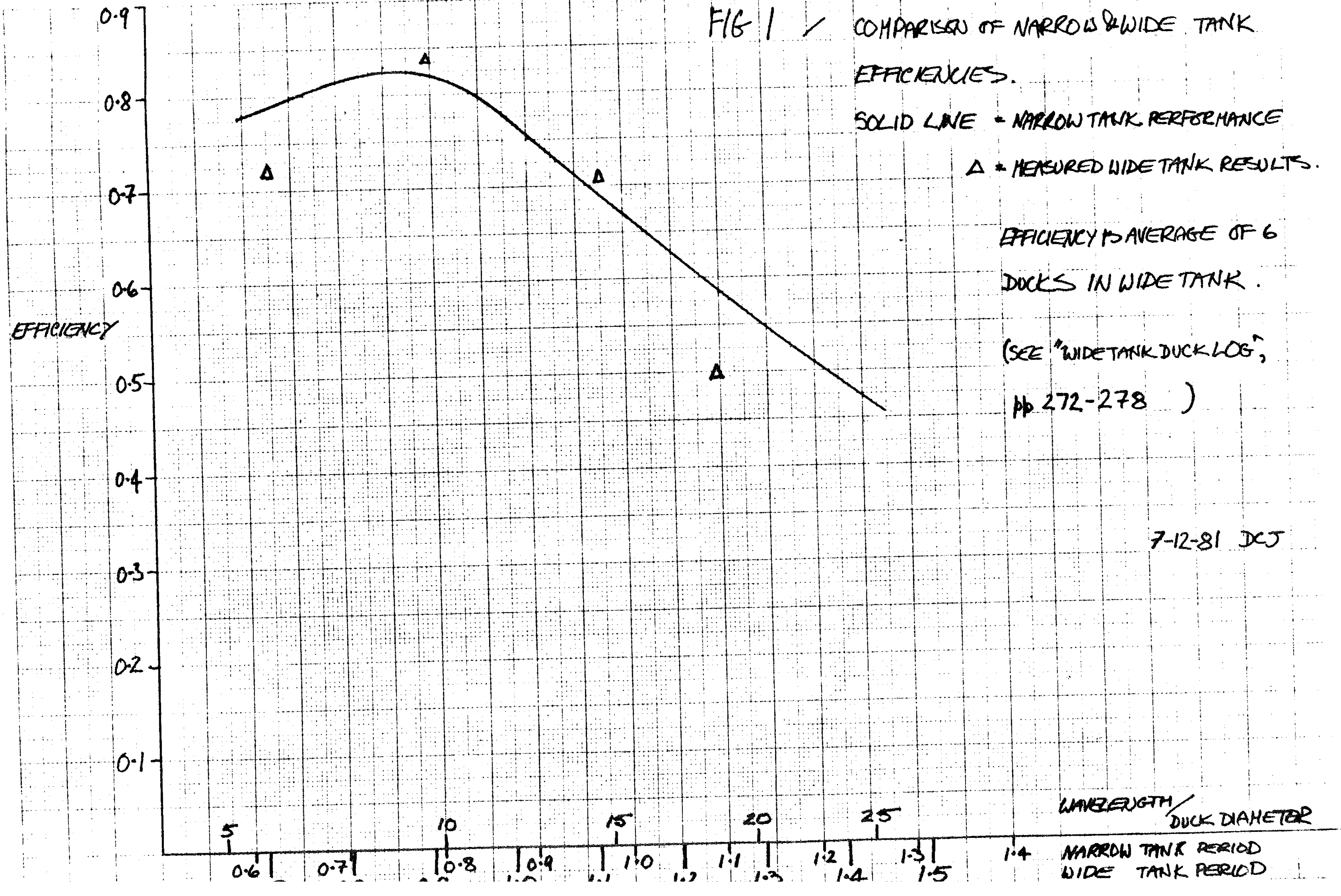
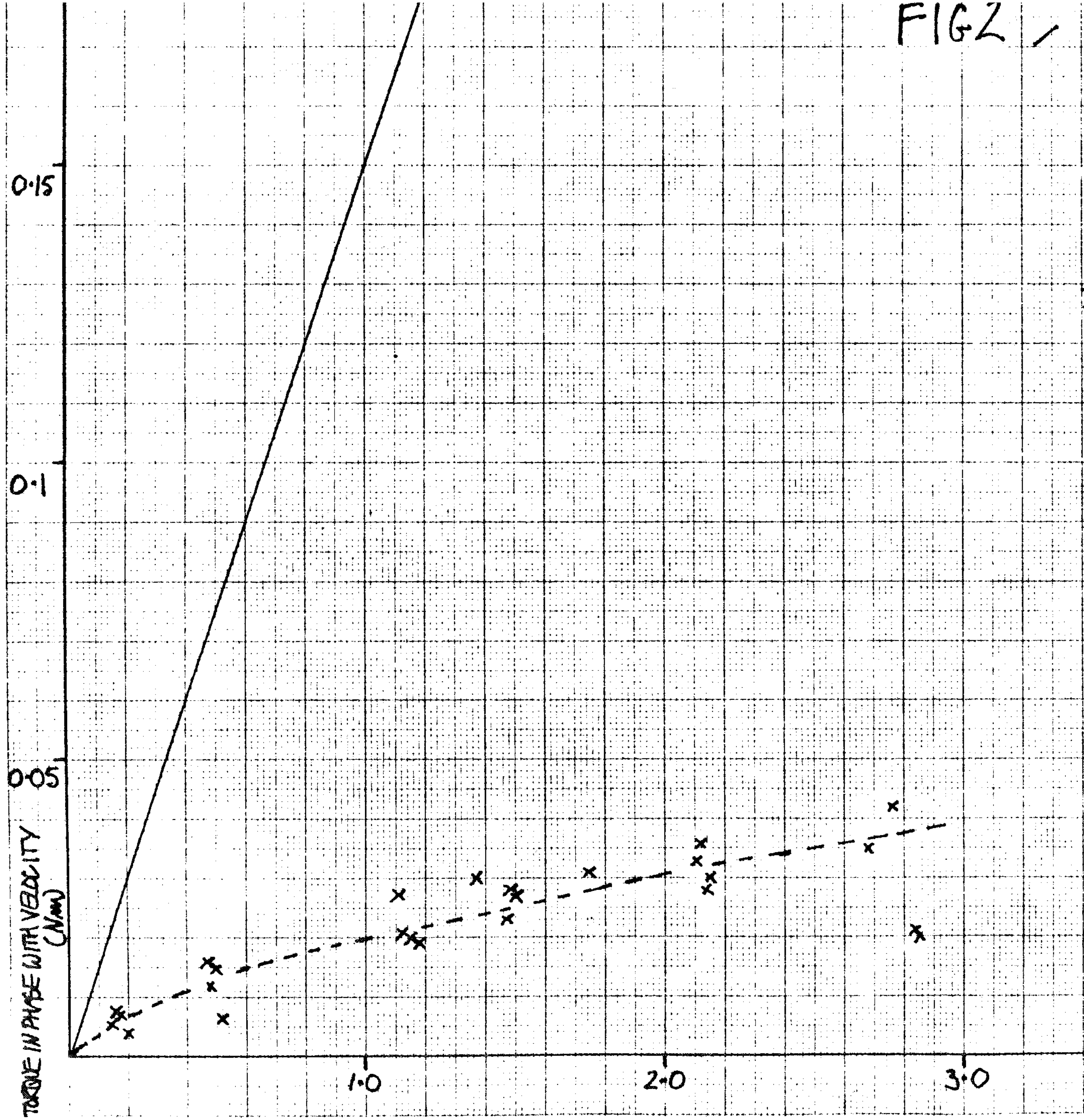


FIG 2 / WIDE TANK MODEL MECHANICAL LOSSES



SOLID LINE = DAMPING TORQUE @ DAMPING = ·15 Nm/RAD/SEC

x = MEASURED LOST TORQUE

DOTTED LINE = BEST LEAST SQUARES FIT TO DATA POINTS - - - -

$TQ_{LOST} = \cdot 0195 (\text{VELOCITY})^{\cdot 636}$

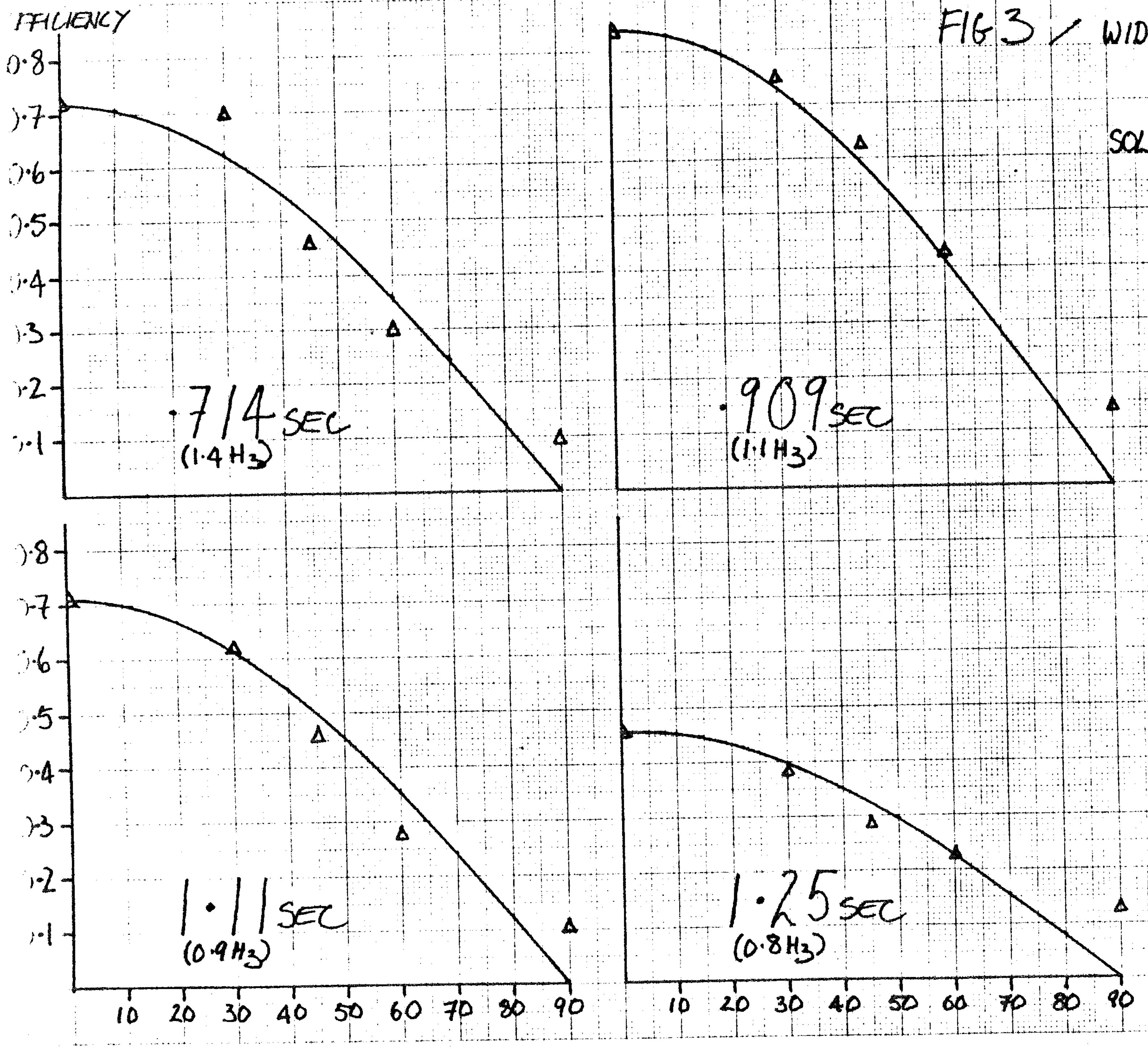
DATA FROM DUCKS

11 55 49 12

(SEE "WIDE TANK DUCK LOG", pp 268-270)

7-12-81 DCJ

(LOSSES MEASURED WITH DAMPING = ·15 Nm/RAD/SEC @ FREQ = 1 Hz)



ANGLE (DEGREES)

FIG 3 / WIDE TANK MODEL - - -
EFFICIENCY VS ANGLE

SOLID LINE = COS LAW PREDICTION

Δ = MEASURED VALUE

TESTS DONE WITH A GROUP of 6 DUCKS...
11, 55, 49, 12, 32, 03

(SEE "WIDE TANK DUCK LOG", pp 272-278)

7-12-81 DCJ

3.1.5 Extent of Design and Optimisation

The principal optimisation exercise which has been carried out is on the choice of spine diameter. The 1979 reference design employed a 10 m nominal diameter spine which had imposed undesirable buoyancy limitations on the duck body, and a low fatigue life on the spine. Optimisation of the energy conversion system has suggested an increase in diameter and this has been followed through in the structural design, two sizes of 12.5 m and 15.0 m diameter being examined. Of these the 12.5 m diameter is still somewhat too small for adequate fatigue life; the 15.0 m diameter is satisfactory from the point of view of fatigue, but the economical spine wall thickness became rather slender, when related to a constant weight per metre run, to be acceptable from considerations of temporary construction conditions and also ring bending due to hydrostatic pressure.

The diameter of 14 metres selected to suit recent power optimisation appears also to suit structural requirements, and lies between the two extremes of size examined.

The main requirement of flotation, overall strength and fatigue life have clearly been met, although many further checks on every possible element would be desirable.

3.2 Mechanical Details

3.2.1 Spine/Duck Bearings

Alternatives are under consideration. The state of development on rollers and chains needs no comment here. On magnetic bearings, however, things are not so positive. At paragraph 4.4 is a paper on the work carried out so far by the team at Edinburgh University. Appended is a copy of a letter indicating that Mullard, one of the biggest manufacturers of ceramic magnets in Europe, believe the proposed design to be feasible. Further development is obviously very necessary but well worth while when the end result is a long life, maintenance free bearing for this application. Quotations for magnets have also been received from Stackpole Corporation of America, indicating prices of about £500 per ton (less than half the figures quoted by Mullard).

3.2.2 Spine Joints

(Drawing No. 10040)
The spine joint comprises a Hooke's joint at the centre with 8 rams mounted around the periphery. The rams, which operate at 6000 psi, are standard commercial units modified only in that the seals and bearings are selected to give a long, trouble free life. The bearings at the extremes of the Hooke's joint are laminated steel and rubber designed by Andre Rubber Co. specifically for the life and duty to be encountered. The spherical seatings at each end of the ram are also of laminated rubber and steel construction based on designs which have been proven over the years.

No maintenance is necessary at these points. The joint movements are so small that the Hooke's joint bearings make only the equivalent of 2000 revs. per annum. The hydraulic circuits for the surge direction and the heave direction are kept separate, each circuit feeding into its own swashplate motor. Both motors are coupled to a further swashplate unit, the output of which is then finally coupled to a constant pressure main. The motor units concerned will be identical to those already used in the duck. A further swashplate motor will be coupled to a single flywheel used as a smoothing for a series of ducks (see Drawing Nos. 10081 and 10037).

3.3 Mooring

3.3.1 Design Philosophy

The philosophy behind the design of the moorings is to achieve a constant tension mooring by allowing variations in force to be absorbed by changing geometry. The sinker and float configuration (shown on Drawing No. 10090) achieves this objective whilst restraining the Spine/Duck assembly to remain within approximately two diameters of its original neutral position, even during the heaviest of seas.

3.3.2 TANK TESTING

FIRST RESULTS FROM SPINE MOORING TRIALS

(Based on 12 m dia. Spine)

Background

Narrow tank results indicated that the long term mean horizontal force on ducks amounted to about 25 tonnes per duck with a slight fall as wave amplitude increases. A gyro control method intended to nurse gyro bearings in large waves can reduce the value to about 10 tonnes in large waves. The first experiments in the wide tank show that these results can be carried over.

The 1979 reference mooring scheme was similar to that shown on Drawing No. 10090. It was designed with the following aims in mind:-

1. To exploit the advantages of long crest-spanning spines, such as shared duty, power accumulation, the avoidance of side-to-side collisions and extra damping.

2. To pay as much attention to keeping rode tension constant as to keeping it low.

3. To avoid the need for special bottom or depth requirements. (It turned out that we had to specify rather deep water but there is plenty of that.)

4. To achieve very long inspection intervals by insisting on mooring viability after a large fraction of rode failures.

The Model

Our spine is built up of rigid sections, 125mm diameter hinged at intervals of 400mm. Hinge characteristics can be modified electronically during the experiment. For these tests we used a hinge set to yield at a bending moment about equal to the failure point of full-scale concrete. It floats with a freeboard of about 5mm.

In an attempt to induce some of the predicted phenomena we removed alternate moorings using 6 sets for 42 spine sections.

We used the same weights and floats fore and aft. Weight and buoyancy were about 135 grms.

We varied the lengths of the lines and the separation of the anchor blocks.

We measured the tension in one of the central forward rodes close to the anchor block.

Unless otherwise stated, all the seas used the 10 second PM spectrum at 1/96 scale with a Mitsuyasu spread. We varied the nominal PM amplitude.

Increases in tension are shown positive upwards. The results are given in model units. To convert to full-scale units per duck you should multiply by 96 cubed and divide by 8. This works out roughly to a multiplication by 11 for tons per duck. To convert Hrms to Hsig multiply by 4.

Fig 1 shows the record for Hrms = .68m Te = 10 sec.

Conditions at South Uist will be like this for the majority of the time. The RMS Value of the deviation is 0.65% of the mean value. The maximum peak-to-peak variation over a time equivalent to sixteen minutes at full scale is 3.9% of the mean value.

Edinburgh University Wide Tank Experiment

half 10 sec pm mits 1/8 small floats
Date : 28-MAR-81 Time : 16:31:44

Fig 1
H_{rms} - .68m

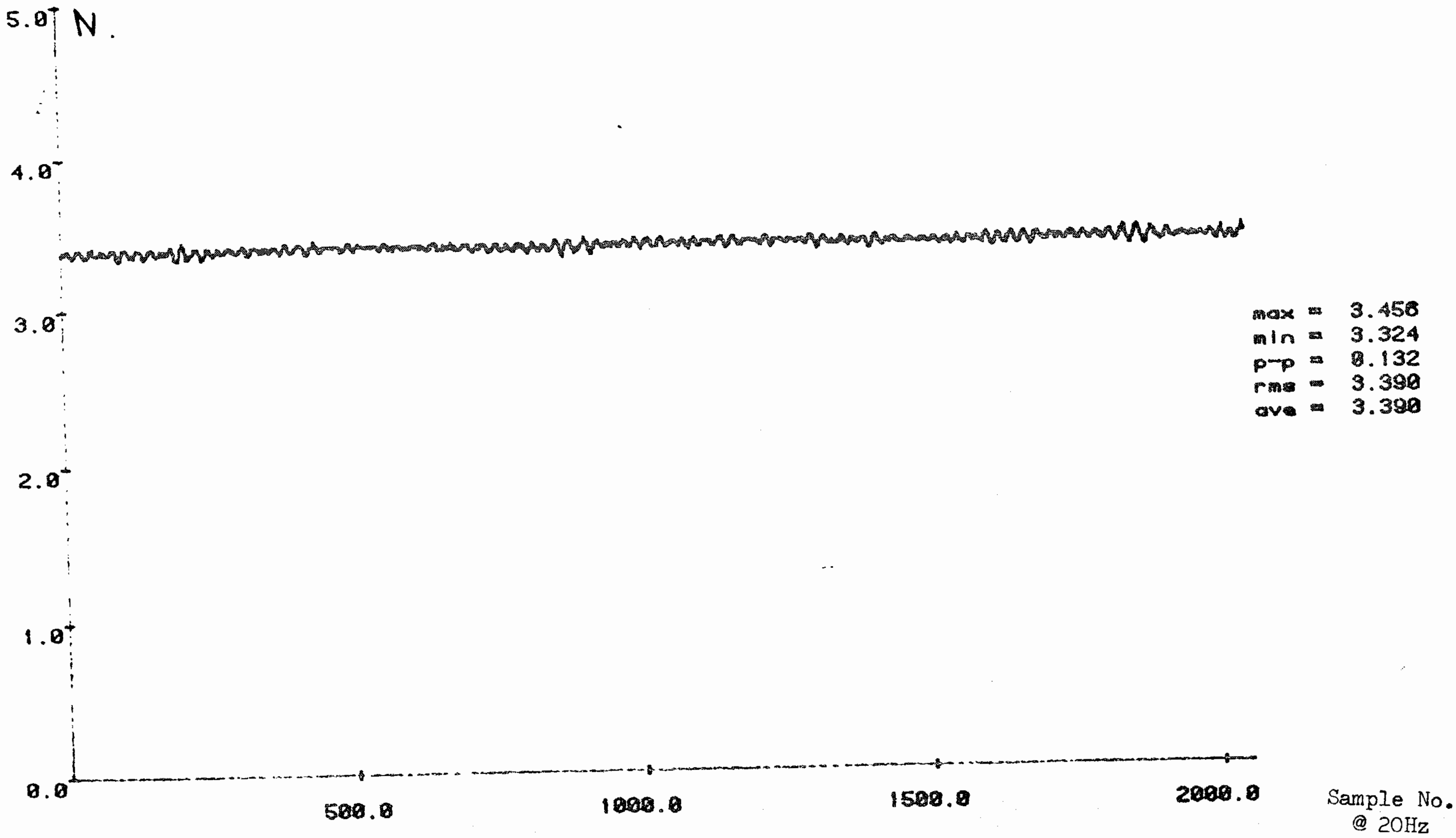


Fig 2 shows the same data at a large scale with the mean level removed. The variations look wave-like and there is some evidence of a low-frequency component.

Edinburgh University Wide Tank Experiment

half 10sec pm mits 1/8 small floats
Date : 28-MAR-81 Time : 16:28:40

Fig 2
H_{rms} - .68m

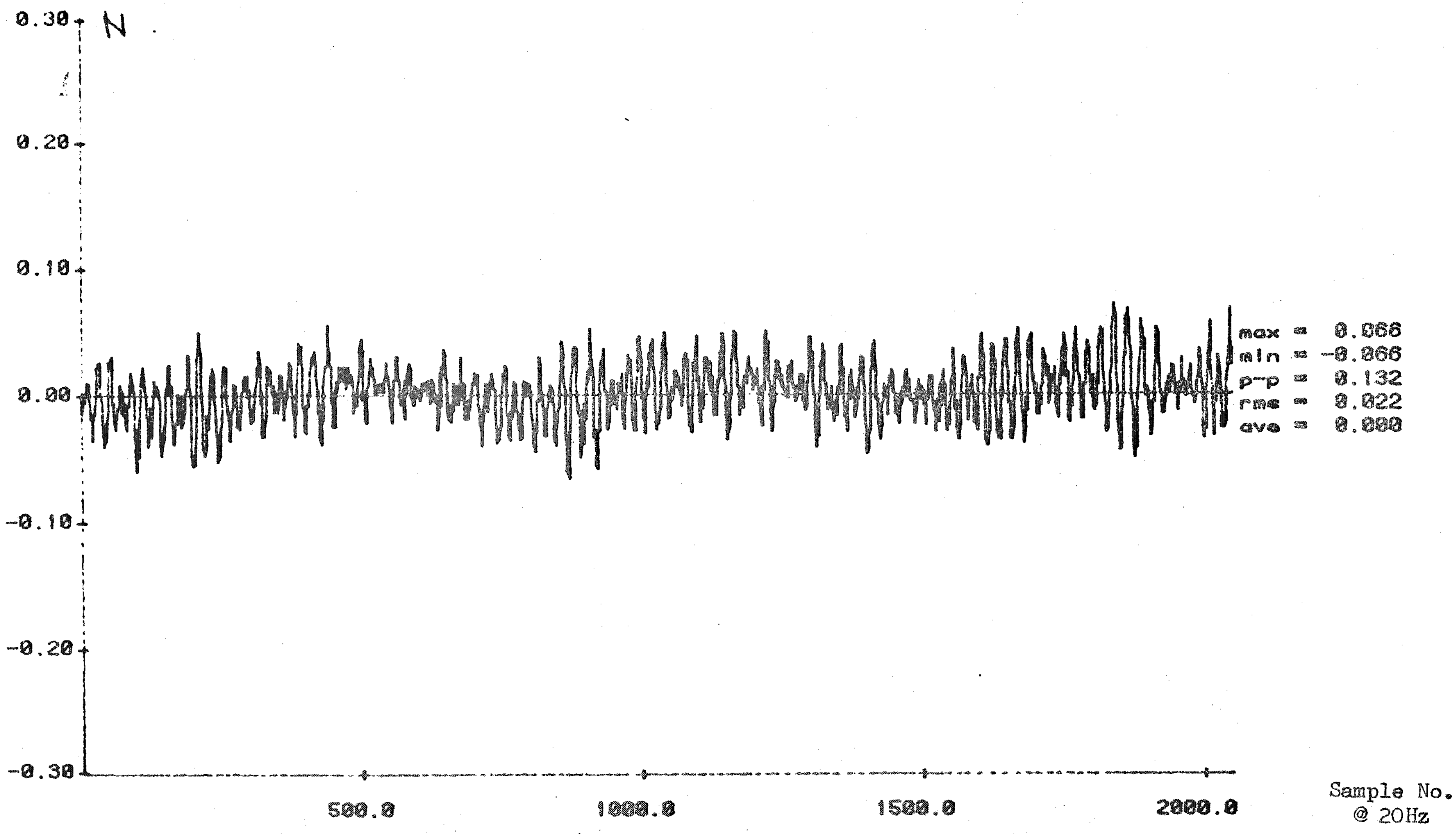
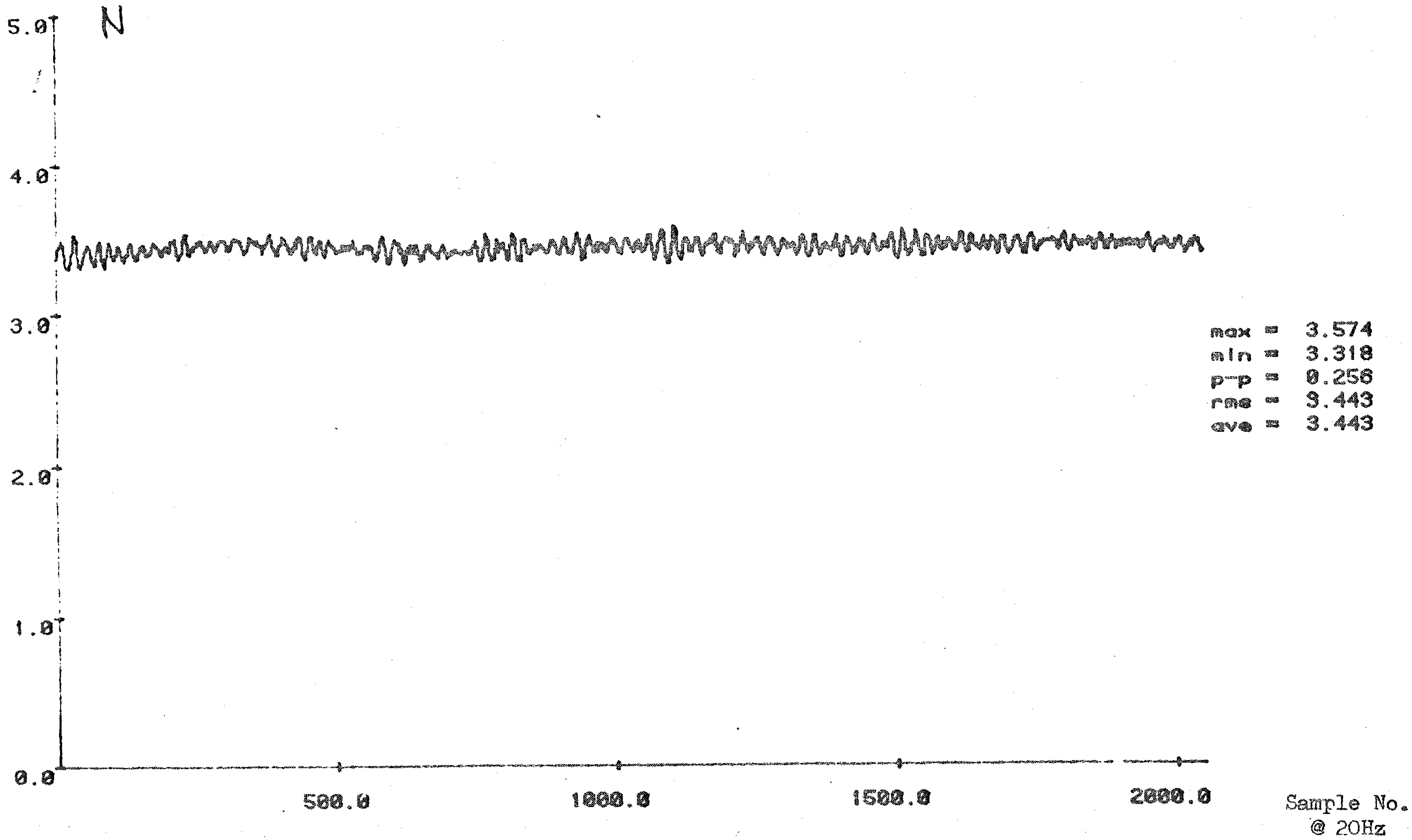


Fig 3 For this we doubled the wave amplitude to 1.36m at full scale. The peak-to-peak variation has increased by 1.93. There is a very small (1.5%) reduction in mean value but we need more experience with long-term transducer drifts before we can vouch for its significance.

Edinburgh University Wide Tank Experiment

one to eight small floats 10 sec pm mits
Date : 28-MAR-81 Time : 14:51:54

Fig 3
H_{rms} - 1.36m

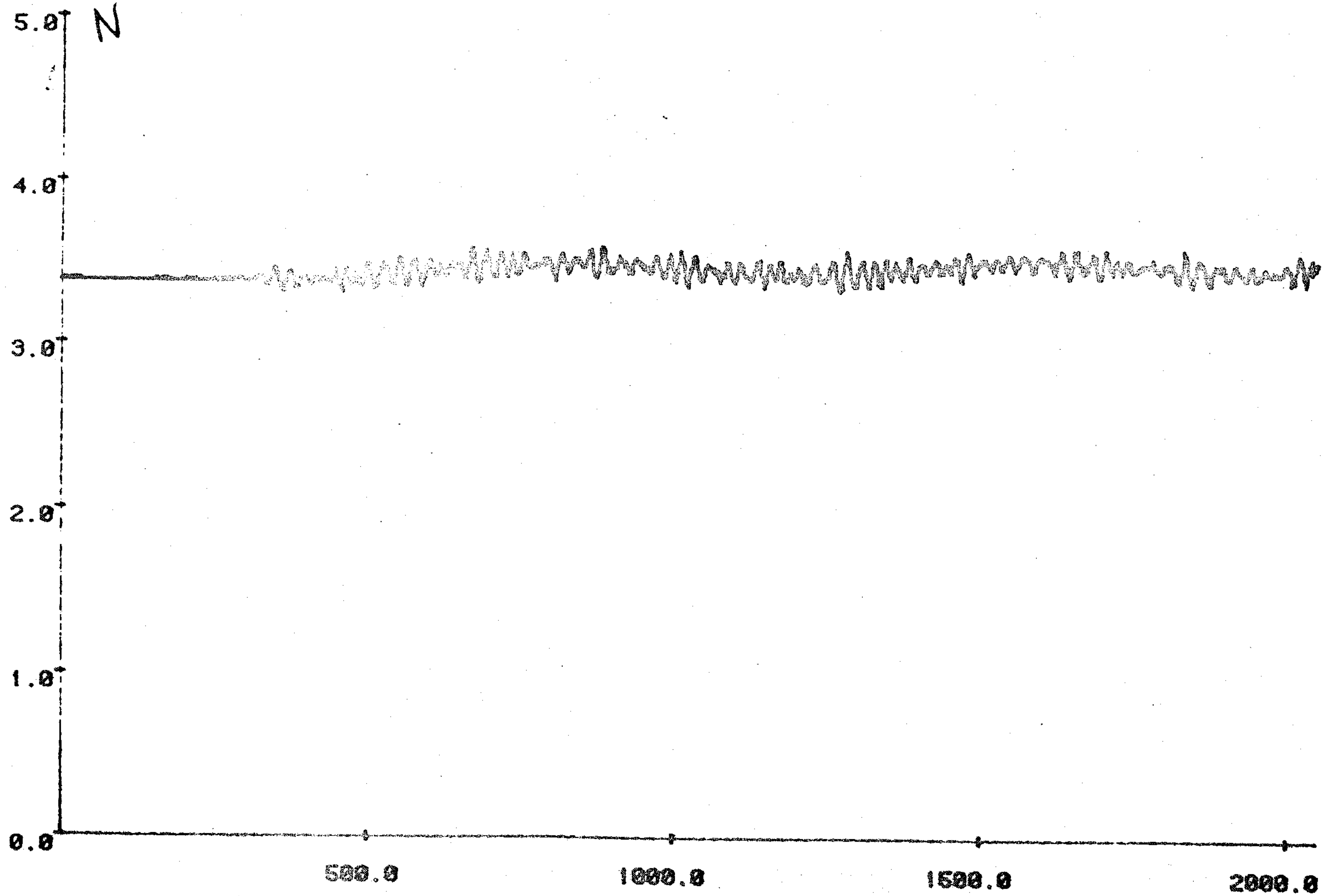


<u>Fig 4</u> For this we tried to induce low-frequency oscillations by a sudden switch on of the wave making system.

Edinburgh University Wide Tank Experiment

15 sec delay 10 sec pm 1/8 small floats
Date : 28-MAR-81   Time : 15:14:58

Fig 4
H_{rms} - 1.36m



max = 3.590
min = 3.308
p-p = 0.282
rms = 3.434
ave = 3.434

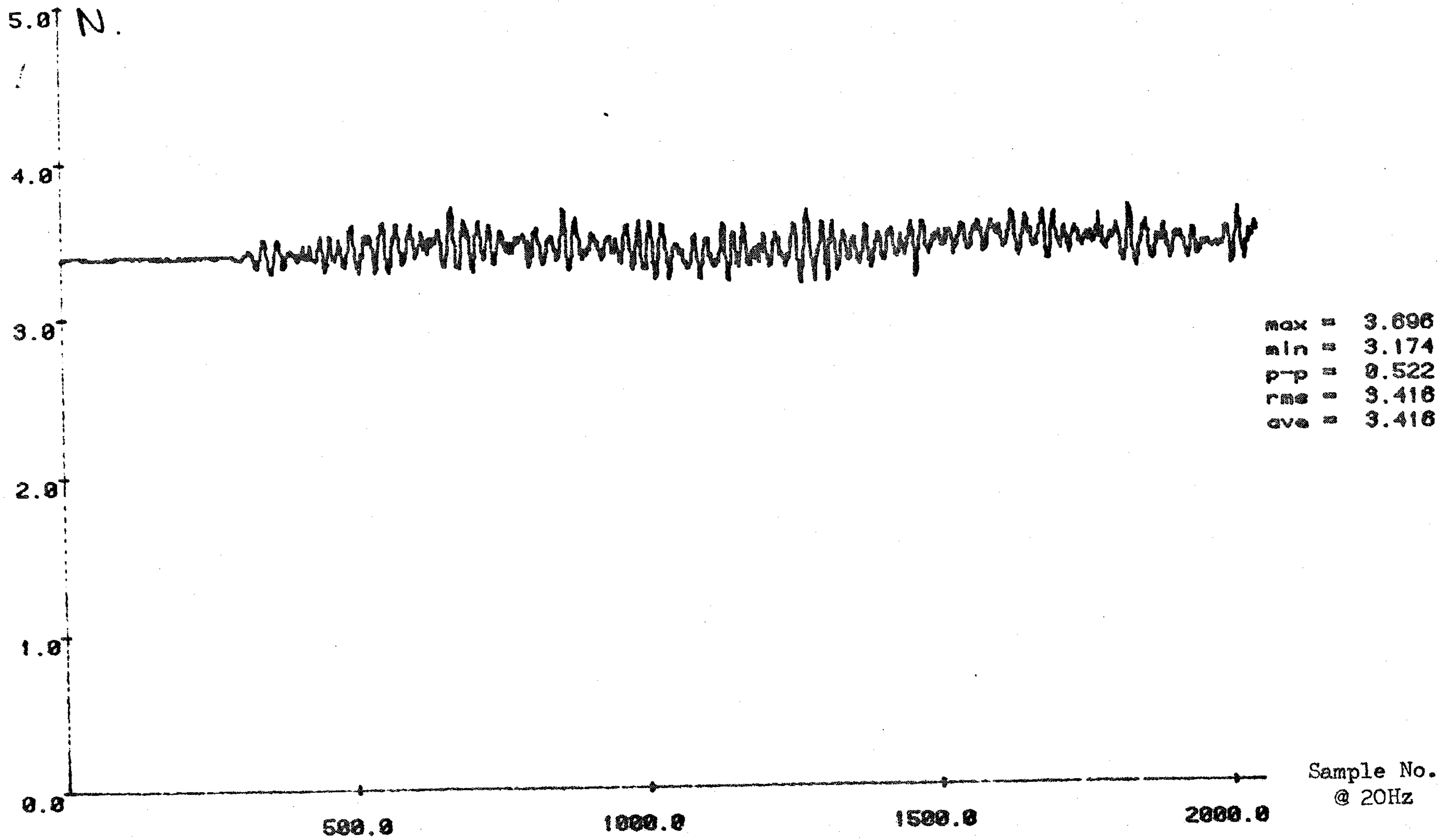
Sample No.
@ 20Hz

Fig 5 For this we did the same for a further doubling of the wave height to 2.72 metres RMS. This is very rough for South Uist and would have a power equivalent to 578 kw/m.

Edinburgh University Wide Tank Experiment

15 sec delay 2*10 sec pm 1/8 small floats
Date : 28-MAR-81 Time : 15:27:52

Fig 5
H_{rms} - 2.72m
Power - 5 78KW/m



max = 3.696
min = 3.174
p-p = 0.522
rms = 3.416
ave = 3.416

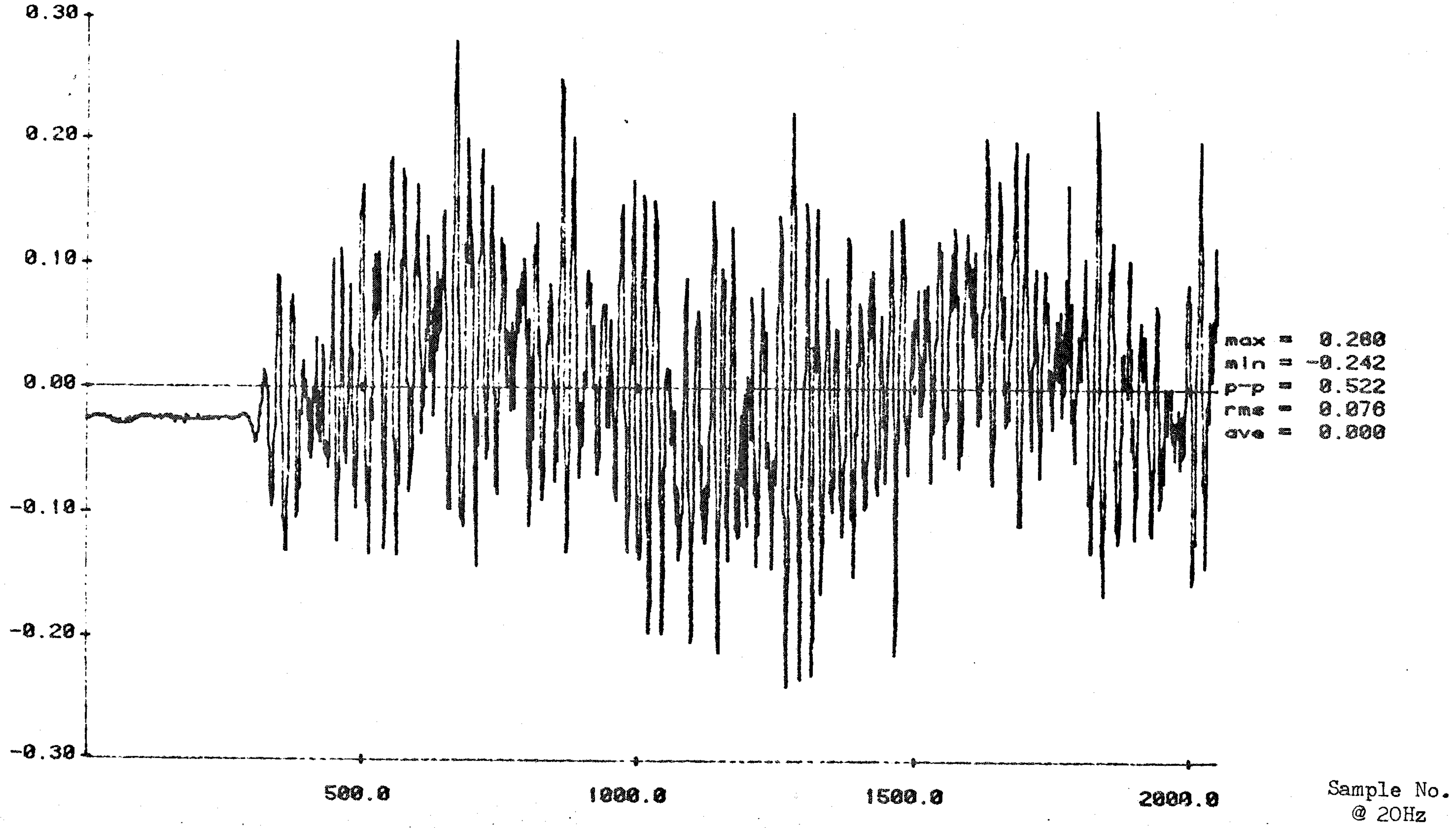
Fig 6 For this we removed the mean level and increased the sensitivity to the same as that used in Fig 2. For a four-fold increase in wave amplitude the peak-to-peak value has increased by 3.95. A small low-frequency response is evident.

Edinburgh University Wide Tank Experiment

15 sec delay 2*10 sec pm mits 1/8 small floats
Date : 28-MAR-81 Time : 15:31:59

Fig 6

H_{rms} - 2.72m

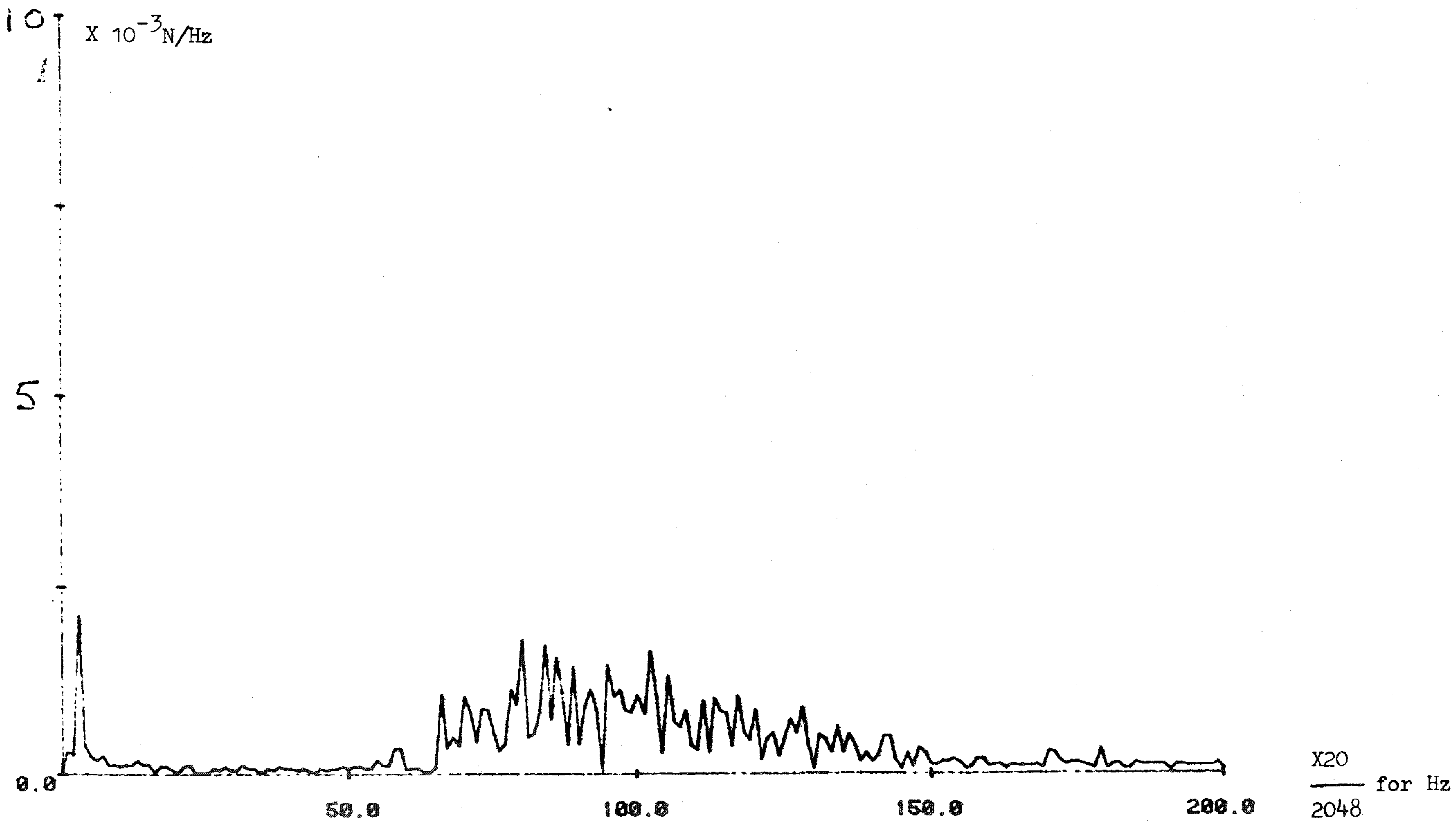


Figs 7, 8 & 9 For these we show the spectrum for the three levels of wave amplitude. The low-frequency component is present at about the place expected for a set-down effect. It is about the same size as the excitation of the wave part of the spectrum.

Edinburgh University Wide Tank Experiment

tension spectrum half 10 sec pm mits 1/8 small floats
Date : 28-MAR-81   Time : 16:18:01

Fig 7
H_{rms} - .68m

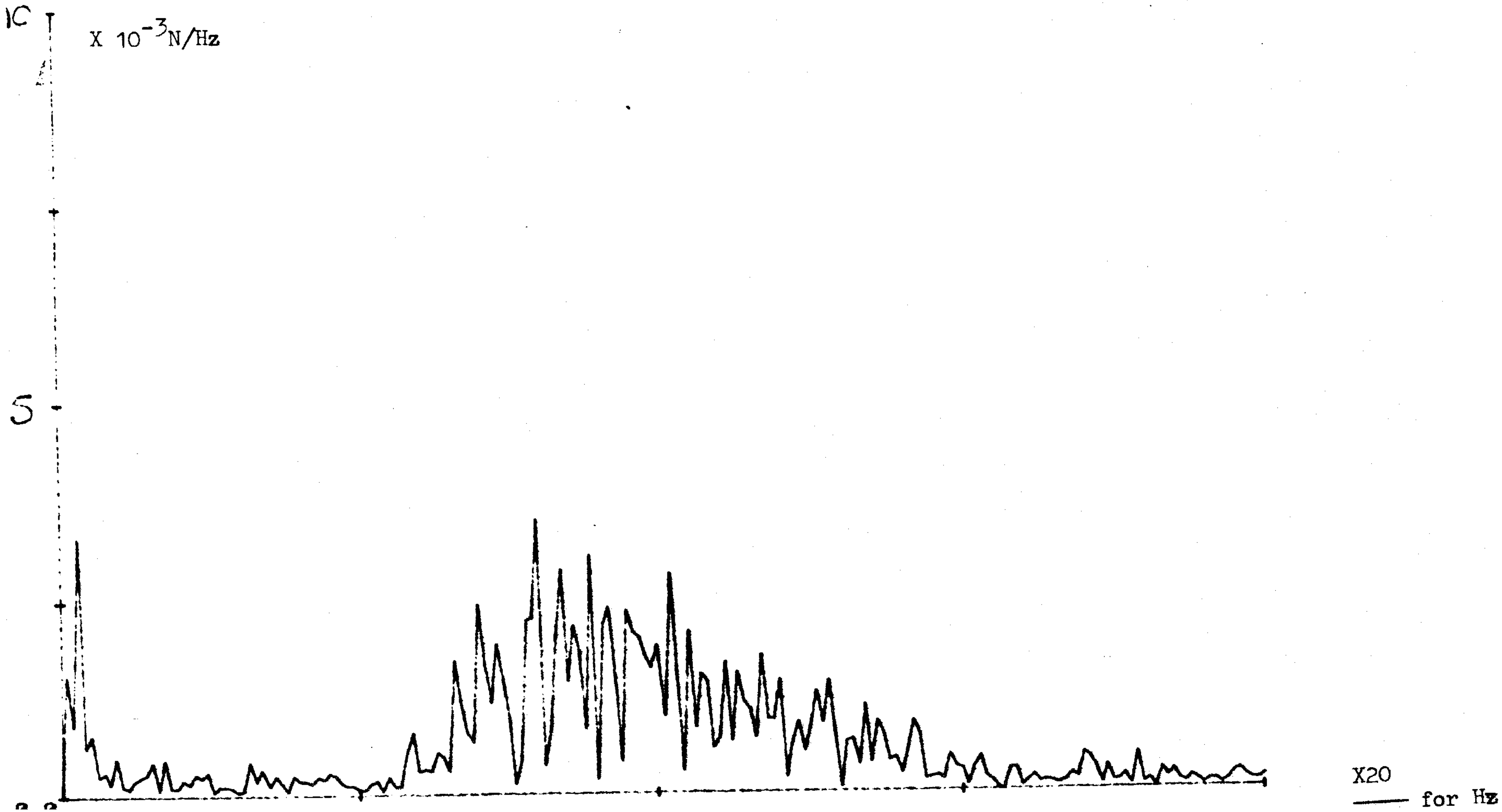


Edinburgh University Wide Tank Experiment

Fig 8

H_{rms} - 1.36m

spectrum tension 1*10 sec pm 1/8 mite small floats
Date : 28-MAR-81   Time : 16:11:54



Edinburgh University Wide Tank Experiment

tension spectrum 2*10 sec pm mite 1/8 small floats
Date : 28-MAR-81 Time : 16:04:06

Fig 9
H_{rms} - 2.72m

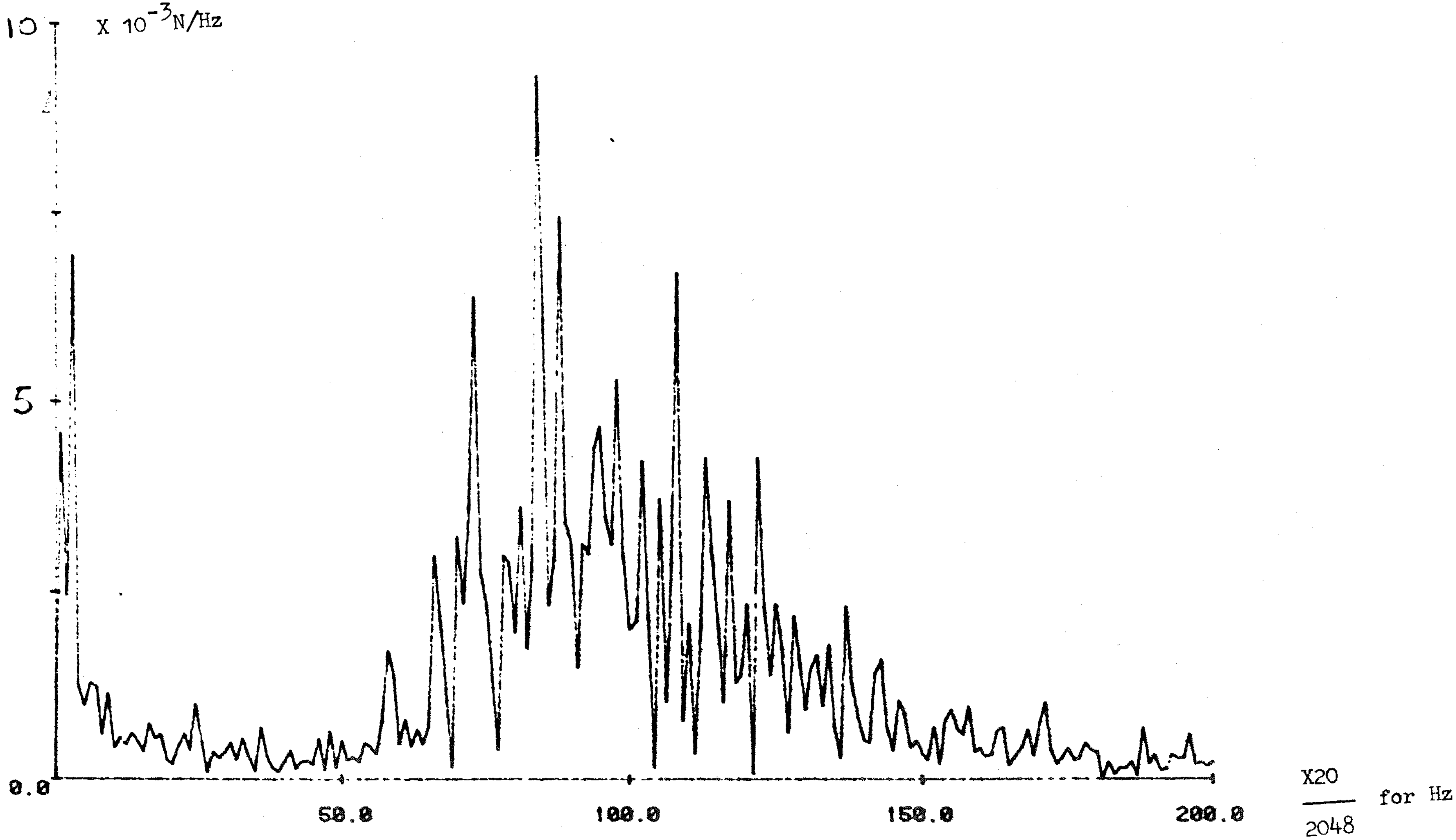


Fig 10 Here we attempted to show the effects of damping of the model in calm water by pulling it forward about three diameters with a boat hook, holding it and releasing it. A good deal of damping is present.

We believe that damping may be increased by the presence of waves, but have not so far been able to show this experimentally.

Edinburgh University Wide Tank Experiment

barge pole test 1 rdc
Date : 28-MAR-81 Time : 14:38:45

Fig 10
calm water recovery
after 3 diameter disturbance

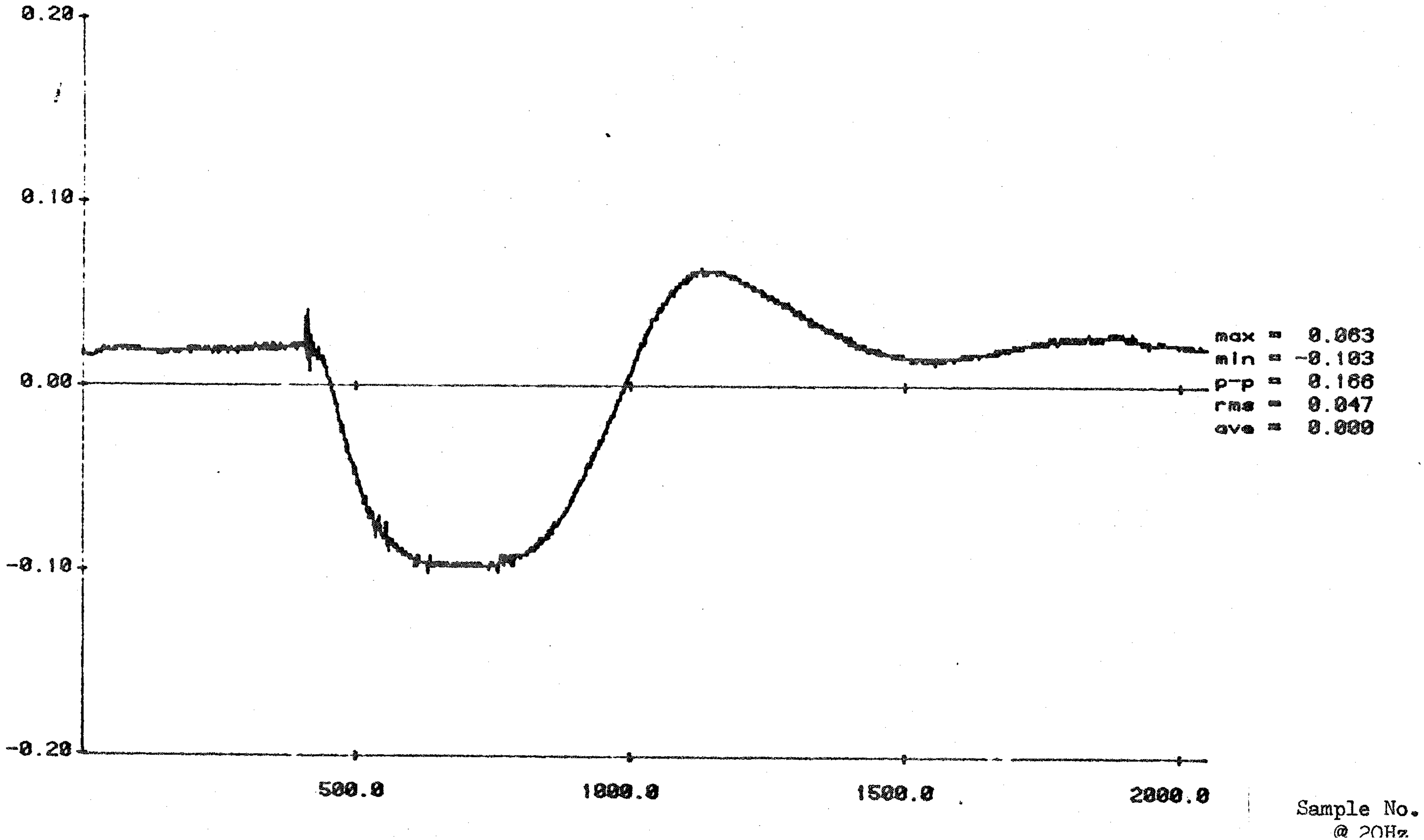
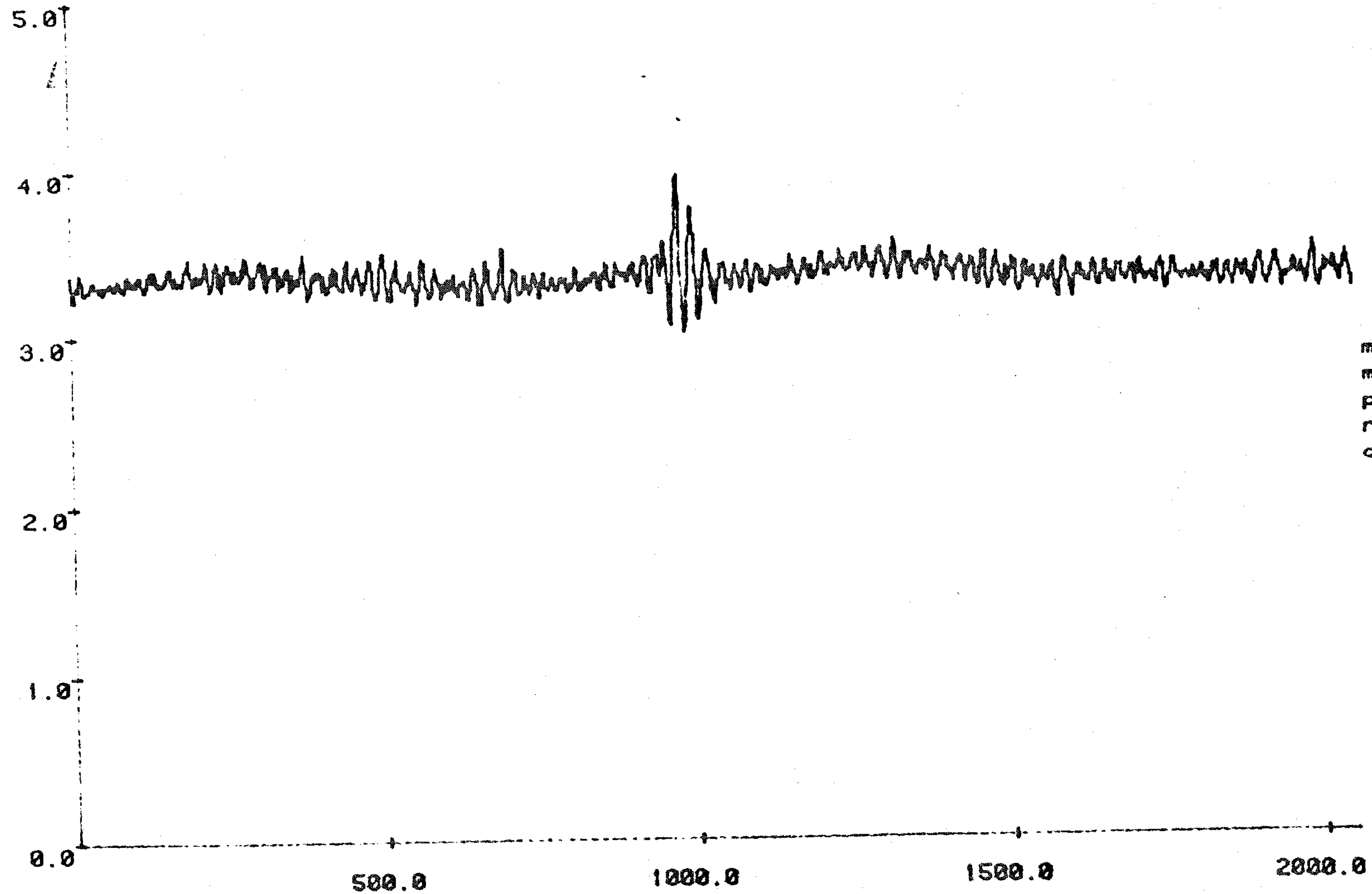


Fig 11 Here we arranged to generate an extreme freak wave with peak-to-peak/RMS ratio of 12.25 at the middle of the spine during the course of a 2.72 metre RMS sea. This would be equivalent to an amplitude of over 33 metres. The value exceeds the range of our gauges and is subject to confirmation.

Edinburgh University Wide Tank Experiment

tension 2*10 sec pm mits freak 12.25 pp/rms 1/8 small floats
Date : 28-MAR-81 Time : 17:03:53

Fig 11
Freak wave
- 33m during
2.72m H_{rms}



max = 3.956
min = 3.006
p-p = 0.950
rms = 3.345
ave = 3.344

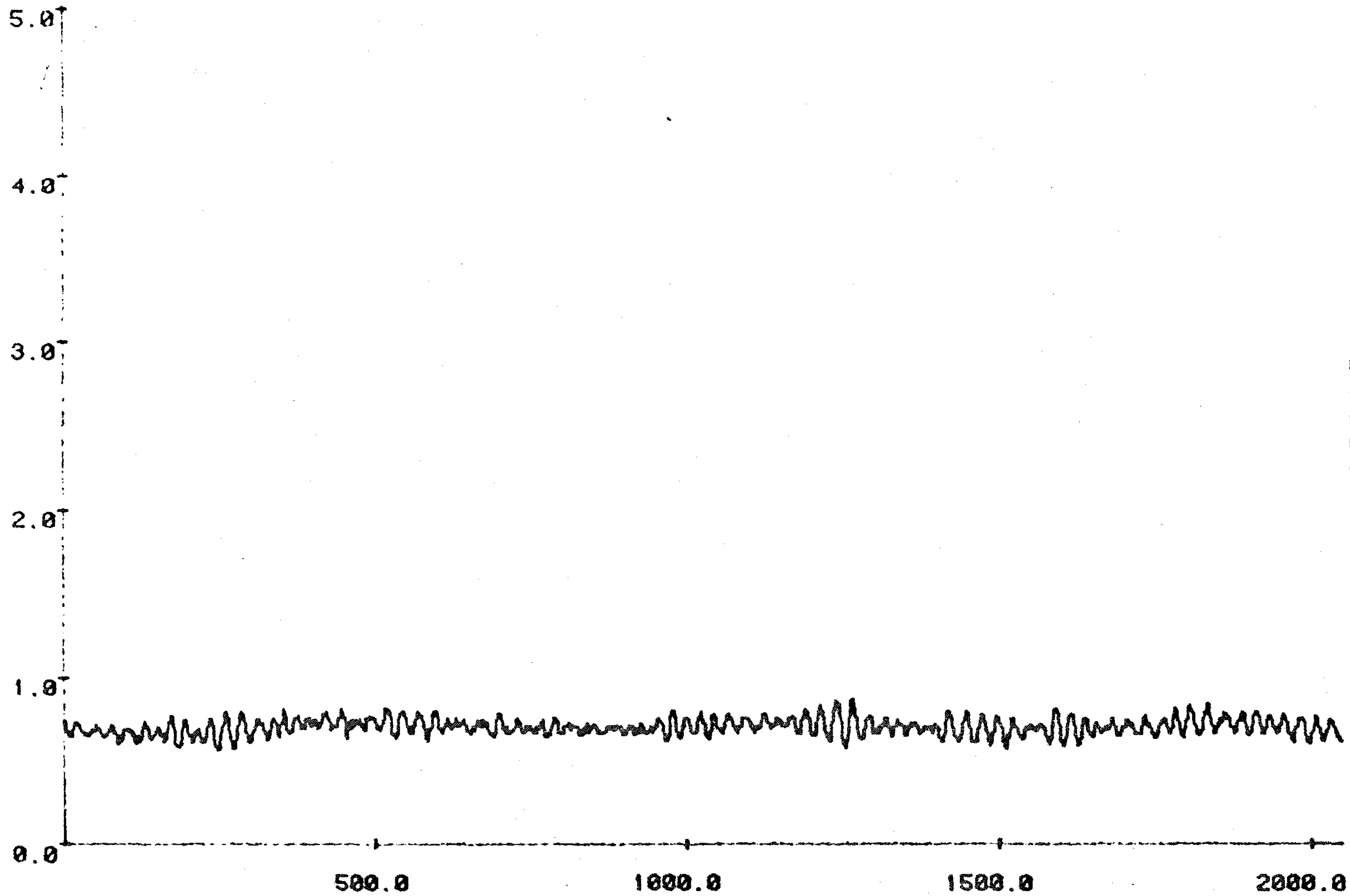
Fig 12 For this we re-arranged the mooring with its anchor blocks closer together and adjusted the line lengths to bring the front buoys to half tank depth. The wave amplitude is 1.36m, the same as for Fig 3.

Mean tension is reduced by a factor of nearly 5 to just under 8 tonnes per generic spine device, but peak-to-peak variations have increased by 22%. Clearly there is a good deal of freedom to trade one characteristic for another.

Edinburgh University Wide Tank Experiment

mk II config 10sec pm mits 1/8 small floats .8 .4.127
Date : 30-MAR-81 Time : 15:38:01

Fig 12
H_{rms} - 1.36m
(See Fig 3)



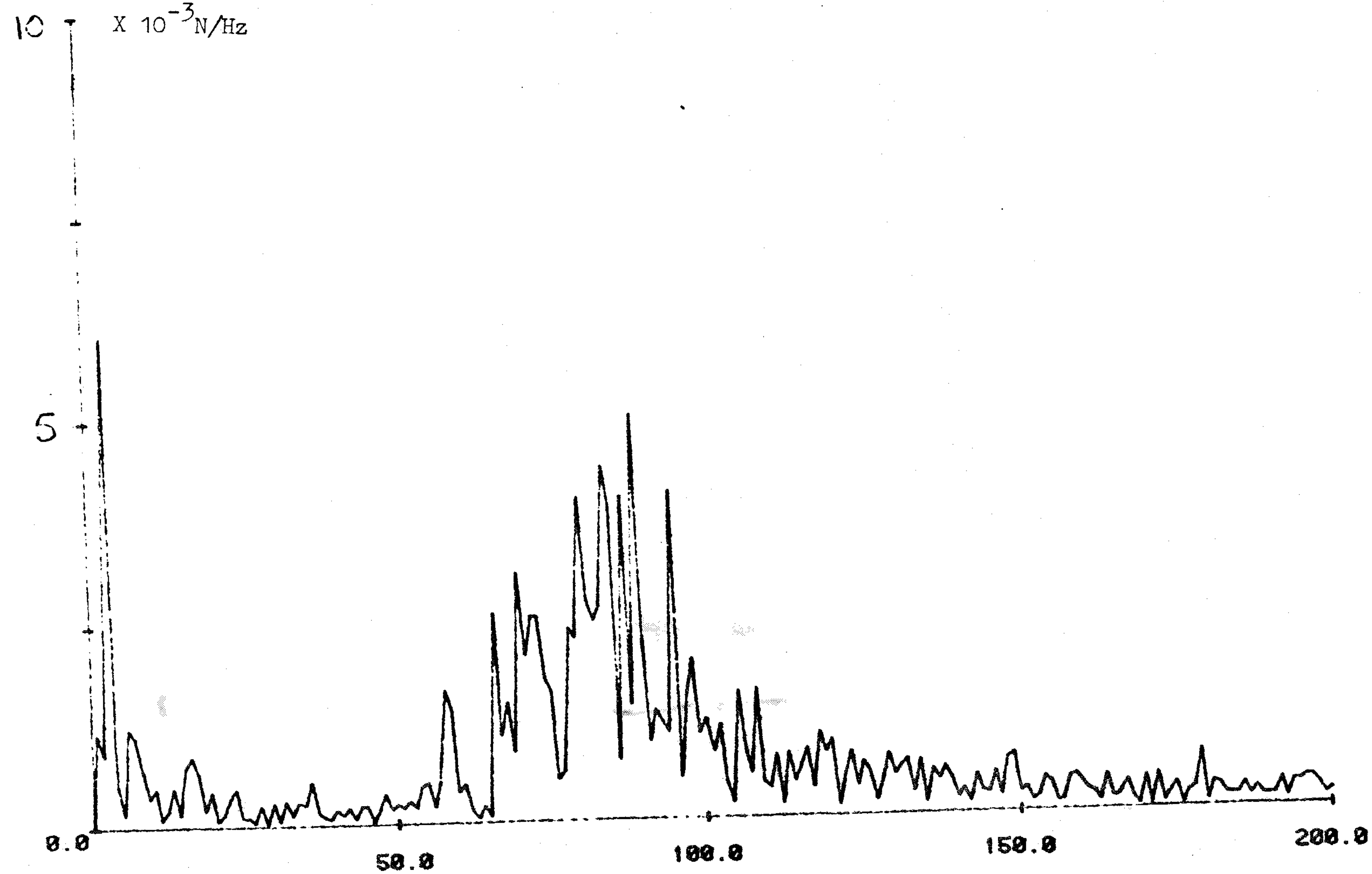
max = 0.864
min = 0.552
p-p = 0.312
rms = 0.699
ave = 0.697

Fig 13 shows the spectrum for the slacker configuration. The lower frequency components are more prominent and the region between points 100 and 150 is emptier than the corresponding region in fig 8. This comes as no surprise.

Edinburgh University Wide Tank Experiment

spectrum tension mkII 10 sec pm mits.8.4 1.27 small floats
Date : 30-MAR-81 Time : 15:41:20

Fig 13
H_{rms} - 1.36m



Conclusions

1. While we still need to do a good deal of testing with generic energy-absorbing devices mounted on the spine it is clear that our original optimism about very long compliant spines was not wrong.

2. The 8 tonnes per duck value is just a party trick and we intend to stick to 25 tonnes per duck for the time being.

3. Considerable reductions in damping at larger scales can be tolerated.

4. Information about the relative merits of low mooring forces amd constant mooring forces would be helpful for further work.

S. H. Salter

April 1981

3.3.3 Extent of Design

The design is complete except for detailed considerations of floats and sinkers.

3.4 Primary Power Absorber

3.4.1 Concept and Details

The aim of the primary power absorber is to achieve synchronous electricity output from the wave input with a maximum of efficiency up to the rated power. Accelerometers in the pressure vessel will detect the sea state and through micro-processors will select the number of ring cam pumps required to match the sea state. The output from the ring cam pumps will be fed to flywheels so that excess power is absorbed by the flywheels for the high part cycle while the flywheel gives up power during the low part of the cycle, thus achieving constant pressure and flow at the swashplate motor driving the alternator. Synchronous AC is produced. The speed variation for a 10 ft diameter flyheel is less than 2 RPM in 1500. A surplus of 11% of ring cam pumps ensures that any damaged cylinder can immediately be taken out of service and kept out of service without loss of nominal output. Hydrostatic bearings are used throught the entire mechanical system with the sole exception of the roller cam followers on the ring cams. This will ensure a long, maintenance free life; 25 years is the objective. We understand from EASAMS that a 5 year refurbishment period leads to acceptable economics. Maintenance costs are based on this assumption.

An essential part of the concept is to place all mechanical items of the main power chain into a canister in a low pressure inert gas so that all the equipment is running under ideal conditions and has no effects whatsoever from the ocean environment.

3.4.2 Testing

Each canister will be built and fully tested under controlled environmental conditions in a shore based establishment. It will then be depressurised, flushed and recharged with vacuum stripped oil and floated to the duck construction site.

3.5 Secondary Mechanical Conversion (Spine Flexible Joint)

3.5.1 Design Philosophy

The philosophy behind the design of the spine joint is to ensure a long, maintenance free life, together with a method of achieving a joint stiffness without incurring heavy energy losses. The joint moment comes from the output pressure of the rams, and for a large part of the time the rams are being moved by the sea to limit this moment. This hydraulic pressure is used via swashplate motors to generate electricity.

The system is separate from the primary power conversion and since the heave moment required is only half the surge moment, more power is available from the heave system than from the surge.

A number of ducks are coupled to a constant pressure main and that pressure is ultimately used to drive a hydraulic motor which in turn drives an alternator of a similar type to that in the duck power canister. A flywheel is again used to smooth out any variations in the supply of oil pressure.

3.6 Installation

It is common practice in the North Sea to tow out and position large structures with a high degree of accuracy. Examples of these procedures can be seen in the number of loading systems that have been installed in the short, but productive, life of the North Sea.

The procedure outlined in this report is considered to be the most practicable that could be adopted. It must be emphasised, however, that in order to establish the optimum solution for the tow out and installation of the moorings, a programme of modelling should be carried out.

Because of the similarity of mooring a string of ducks to the installation of moorings for an S.B.M. loading system it is considered that the mooring of the string of ducks would be well within the capabilities of North Sea installation procedures.

S E C T I O N   4

E N G I N E E R I N G   A P P R A I S A L

4. ENGINEERING APPRAISAL

4.1 Structure - General

The structure of both duck and spine are proposed in reinforced concrete with longitudinal prestress, and possibly some transverse prestress. In addition to providing for the main structural stresses, the prestress is seen as a means of preventing the incidence of cracking with its attendant corrosion risks, and, by eliminating cycling into the tensile stress zone, to reduce the effects of fatigue in concrete and reinforcement.

4.2 Structure - Design Conditions

For the Spine, the main design conditions to meet are the forces applied at the moment controlled joint. These are:

(i) The controlled moment of 20,000 tonne-metres on the surge and heave axes (resulting in 28,280 t.m. at 45°).

(ii) Direct axial tension or compression of 1000 tonnes due to chord shortening effect of wave action on the continuous spine.

(iii) 630 Tonnes shear due to maximum moment reversal between adjacent joints.

(iv) 20,000 Sin 10° tonne-metres torsion due to a total joint deflection of 10° under full moment (i.e. a 5° preset horizontal zig-zag angle plus a 5° maximum yield-deflection angle).

(vi) Wave action on the spine itself is considered to cause a maximum rise in bending moment at mid length of about 5000 tonne-metres.

In the transverse direction, hydrostatic forces will cause ring bending and compression in the cylindrical shell which will be partly relieved by the bulkheads. Such forces may cause significant effects in the temporary unballasted condition (during launching) and if accidental deep submergence of the spine occurs in service.

The main consideration in the design of the spine, once the strength requirements for the forces above have been met, and adequate buoyancy has been ensured, is a satisfactory fatigue life. This is discussed below.

For the Duck, the over-riding condition to meet is the necessary buoyancy in amount and location. External forces applied are the hydrostatic and wave slamming forces and in particular, the bearing forces applied by the power take-off. The latter are determined by the resistance torque (torque limit) of the gyroscope, and are controlled through the power take-off at 2250 tonne-metres (22.5×10^6 Nm).

4.3 Structure - Fatigue

For the spine, and to a lesser extent for the duck body, the necessary fatigue life to withstand the requisite number of load reversals determines section sizes, amount of prestress and so forth. These sizes and amounts give a static ultimate strength far in excess of the static load multiplied by the recommended load factors.

It has been necessary therefore to obtain the number of reversals of the forces, and in particular of the bending moment, in the planned life of 25 years. From the 399 wave spectra provided for a water depth of 42 m, modified spectra for a 100 m water depth have been derived by the Edinburgh team, yielding a histogram of RMS wave-height occurrences. Tests on a spine in the wide wave tank at Edinburgh give the RMS bending moment in surge and heave directions on the central portions of a continuous rigidly-jointed spine against RMS wave height.

A correction is necessary to the histogram of bending moment occurrences to take account of the yielding of joints at 20,000 tonne-metres in surge or heave. This correction is based on an estimate made by the Edinburgh team of the percentage of exceedances of the yield moment for every value of "non-yielding spine" bending moment. For spines near the ends of a string, the number of moment occurrences in excess of 20,000 tonne-metres will be higher but current experimental work has not yet given the final value.

Owing to the arrangement of moment-controlling hydraulic rams, the greatest bending moment occurs about an axis inclined at 45^{o} to the horizontal and therefore bending moment occurrences were obtained for this direction also from the heave and surge moments. Consequently it was possible to plot a histogram of extreme fibre stresses, from which, by use of an S - N curve and by application of Miners Rule, the fatigue life has been checked. (See Appendix.)

For spines near the ends of a string it may be sensible to accept a lower (but still commercially viable) fatigue life or alternatively the strength could be increased although this would introduce a non-standard element and affect interchangeability.

4.4 SPINE/DUCK BEARINGS

THE MAGNETIC SQUEEZE-FILM DUCK-TO-SPINE BEARING

DESIGN REQUIREMENTS

The design of the duck-to-spine bearing must take into account, and wherever possible turn to advantage, the following requirements.

1. The forces transmitted from duck to spine will amount to about one ton per metre length in an unloaded condition (due to mooring forces and ballasting errors) and 10 tons per metre at the yield threshold of the spine joints. Though seemingly large, if spread over the entire area between duck and spine, these forces would result in pressures lower than $1000Nm^{-2}$ (0.15 psi) and $10{,}000Nm^{-2}$ (1.5psi) respectively. The design upper limit of the bearing is ten times the yield value, ie $100{,}000Nm^{-2}$.

2. The gap between duck and spine will be filled with sea water, with the bearing operating under fully immersed conditions.

   (This arrangement results in the spine essentially 'floating' inside the duck - the buoyant force it then experiences is equal to the mass of water it displaces, multiplied by the local instantaneous value of its own acceleration, according to Archimedes' principle. In an unloaded condition this acceleration is simply g, the gravitational constant. However, any force imposed by the duck, and transmitted via the water in the bearing, will result in the spine feeling the equivalent of a change in the magnitude and direction of the pull of gravity. As this change will amount to less than ± 50% of g under the most severe conditions predicted, the importance of a water-filled bearing becomes clear).

3. The bearing requires to be radially very thin, in order to minimise the amount of spine diameter used up in accommodating it.

4. As both duck and spine represent very large concrete structures, neither will be built to tolerances of better than (at best) ± 15mm. Further deviations in their shape and size will be caused by the uneven loading of the waves, and the effects of differential thermal expansion.

5. The fraction of power lost at the bearing comes out at roughly twice the coefficient of friction for the surfaces involved. Even the best plain bearing materials, with coefficients as relatively low as 0.05, would result in a 10% output loss.

6. The target design life of the entire device, including the bearing, is 25 years.

The proposed bearing works using a combination of two separate principles squeeze films, and passive magnetic repulsion.

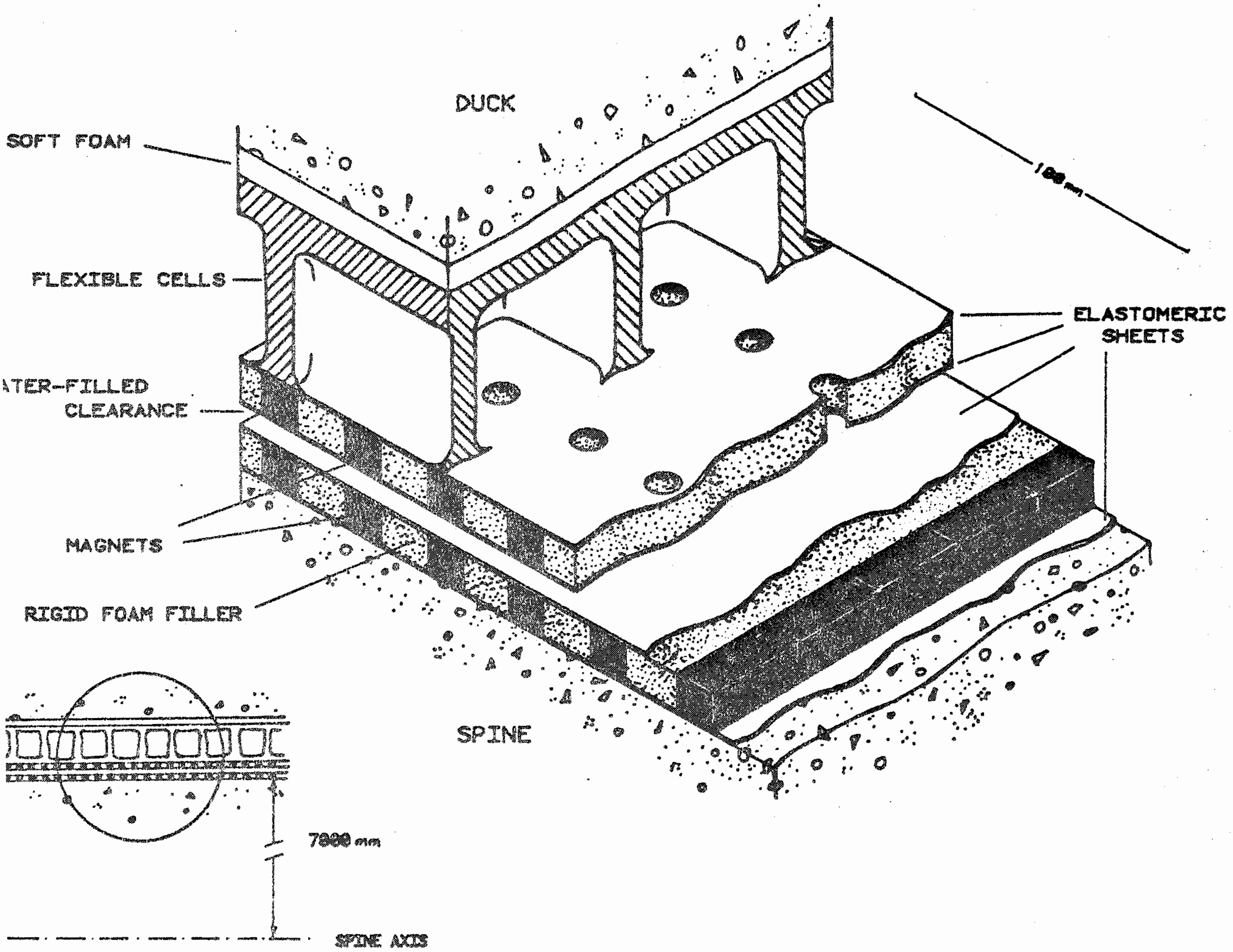
THE DESIGN

The principle of squeeze-films is commonly encountered in journal bearing or wherever two rapidly approaching surfaces are separated by a lubricating fluid. If their clearance is small, and the escape path for the fluid long, then the pressure built up between the surfaces may delay their contact for long periods.

Archibald has documented the relationships governing squeeze-films in terms of fluid pressure, volume flow rate, load sustaining times and convergenc speeds for several cases (1), and it can be shown that the convergence speed decreases roughly as the inverse cube of the clearance.

An illustration of the proposed bearing appears in Figure 1. It is designed so as to provide a very small, but finite, clearance between the outer spine surface and the inner annular surface of the duck, while still allowing for quite large relative displacements and/or stress-induced deformations of both duck and spine. This is accomplished by bonding to the surface of the duck a compressible 'slubber' layer which occupies almost the entire available volume between duck and spine.

The slubber bearing comprises a thick layer of flexible foam rubber or plastic cells with significant strength in compression (ie radial loading) but little in shear (axial loading), somewhat in the manner of 'Dunlopillo". The lower surface of the cell layer is in contact with a semi-stiff foam plastic layer in which are embedded small ceramic magnets; this layer is



GURE 1. SCHEMATIC CROSS-SECTION OF THE MAGNETIC SQUEEZE-FILM BEARING

sandwiched between two elastomeric sheets, each about 1mm thick, bonded to its upper and lower surfaces. The surface of the spine is overlaid with a similar magnetic sandwich in which the magnets are laid end-to-end, forming closed circumferential rings whose polarity is such that they present a continuous north (or south) pole on their upper surface.*

This is aligned with the magnets in the lower slubber layer which also form circumferential rings, but need not be continuous: the back of

*See also Appendix III.

the duck will be subject to smaller forces than the "beak", and in this area the slubber magnets will be separated by discrete circumferential gaps. The duck and spine magnetic rings are arranged in mutual repulsion.

The elastomer at the base of the slubber layer will consist of sheets c. 2m wide (axial dimension) applied from a roll, around the circumferential length of the bearing. Each sheet is pre-tensioned and held at both edges by rails attached to, and forming closed rings round, the *spine*. This is to maintain close alignment of the magnetic rings and ensure axial stability: this problem is discussed in a later section.

OPERATION OF THE BEARING

In the lower wall of each flexible cell will be one or more holes, continued through the magnet layer of the slubber bearing, and allowing water free access in or out of the cell. In the gap outside the slubber, water will be free to flow axially or circumferentially around the spine; inside it will be trapped, unable to flow in any direction other than out of the cell. We have created a 'boundary layer' of water, retaining its full hydrostatic properties, but with a restricted ability to flow.

The spring characteristics of the repelling magnets and the compressible cells will be chosen to complement one another, as shown in Figure 2.

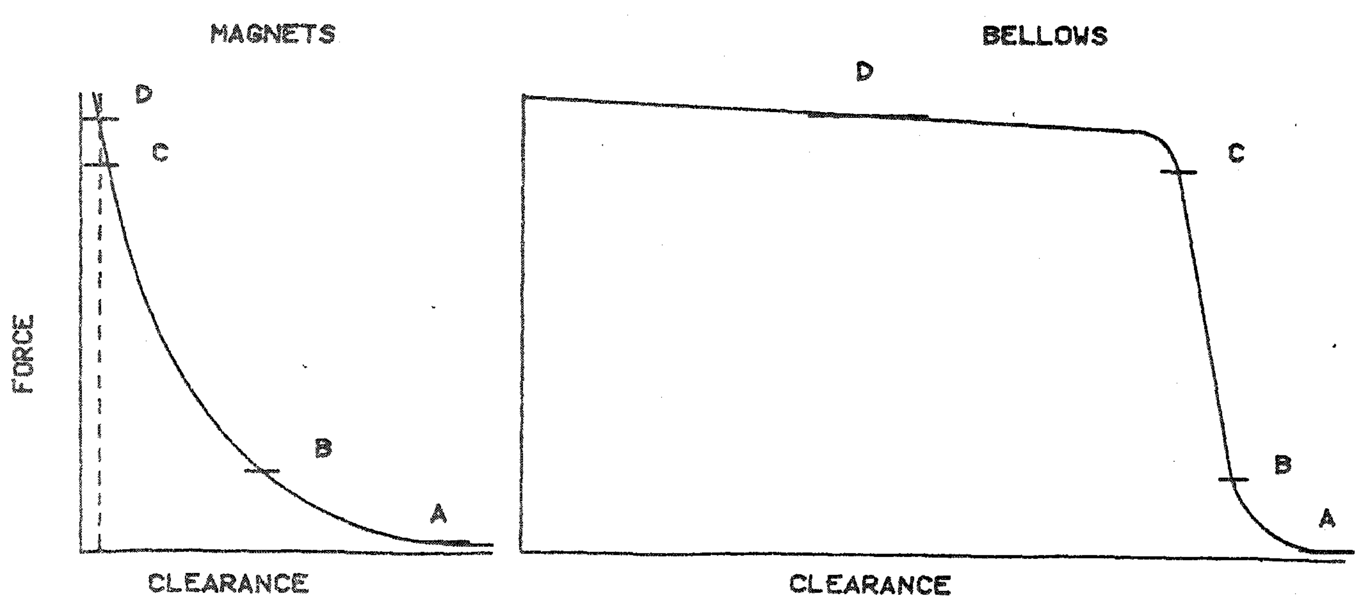


FIGURE 2. CLEARANCE VS FORCE CURVES FOR THE BEARING. THE FORCE FROM A TO B PRE-COMPRESSES THE BELLOWS. B MARKS THE STATIC POINT FOR DIRECT MOORING FORCES. AT C THE BELLOWS BEGIN TO CLOSE, FORCING WATER OUT THROUGH THE GAP, WHICH IS NOW VERY NARROW AND STILL CLOSING. THE FIRST SOLID-TO-SOLID CONTACT OCCURS AT D.

The load that can be sustained by the bearing will simply be the sum of the local pressure, plus the forces exerted by the magnets and bellows; the freedom of flow in and out of the cells ensures no pressure gradients are set up between the water inside, and that outside in the bearing clearance.

The small initial force increase from A to B will give rise to a limited, but significant, deflection of the bellows, accomplished by including in the cell layer a thin sheet of easily compressible foam (see Fig 1). At B this foam is fully compressed, and this marks the unloaded working point of the bearing, chosen from a knowledge of the mooring forces and ballasting errors. This "built-in squeeze" is to allow the magnet clearance to remain relatively small when subjected to negative loads, ie on the side of the duck diametrically opposite to the point of loading.

As the load increases, magnetic repulsion absorbs most of the rise from B to C with the gap closing accordingly; so far there is little change in the cell volume. Above point C the bellows begin to compress; the volume of the cells reduces and water is expelled through the holes in the lower sheet. Once released, the water may flow circumferentially or axially round the spine - the distances in either case are long compared with the clearance, and pressure gradients between adjacent cells are small. Flow is further impeded by the narrowing of the gap, and as volume flow rate is proportional to clearance cubed, this effect becomes increasingly important.

At D the two sides are about to contact, with the force limited to that required to deflect the bellows; these still have plenty of water left to discharge, but the clearance is now infinitesimally small. The spring characteristics of the bellows give a very flat curve at this point.

For the biggest waves likely to be encountered, this 'enhanced squeeze-film' bearing can sustain the loads for several hundreds of seconds; in practice the load reversal times will never exceed eight seconds. The magnetic component of the bearing will sustain smaller loads indefinitely, as well as automatically compensating for any deviations in spine diameter (even on a very local scale). By varying the circumferential pitch of the slubber magnets, the uneven nature of the load pattern can be exploited. Actual physical contact between the bearing surfaces should never occur.

THE MAGNETS

When two permanent magnets are arranged in repulsion, each comes under the influence of a demagnetising field, this being the vector sum of the magnet's own *self-demagnetising* field and the field created by the presence of the other. The property which determines the field strength around (and within) a permanent magnet, and the attractive or repulsive force it can exert, is its magnetisation J. This vector quantity, also known as the pole strength per unit area, is not constant, but varies with the demagnetising field experienced by the magnet. To further complicate matters, removal of the externally applied field in general restores the value of J to less than its former magnitude, depending on the hysteresis properties of the particular material used. In other words, permanent magnets in repulsion tend to permanently weaken each other.

For this and other reasons, the bearing will use anisotropic barium ferrite magnets, which belong to the class of "hard" magnetic materials: these have relatively low values of J, but can withstand extremely strong demagnetising fields with little loss of performance. This is illustrated in Figure 3.

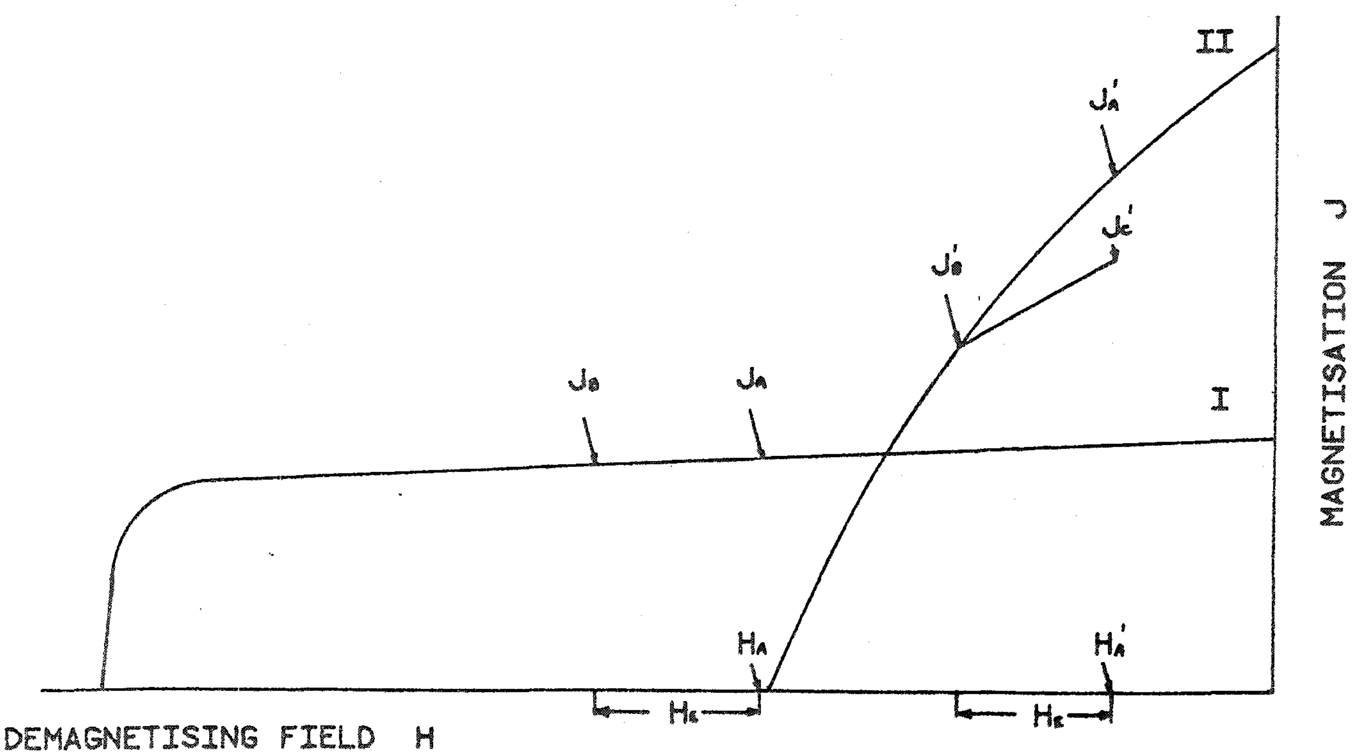


FIGURE 3. DEMAGNETISATION CURVES FOR PERMANENT MAGNET MATERIALS, (I) HARD (EG. ANISOTROPIC BARIUM FERRITE), AND (II) SOFT (MILD STEEL). J_A AND J_A' ARE THEIR VALUES OF MAGNETISATION IN SELF-DEMAGNETISING FIELDS OF H_A AND H_A' RESPECTIVELY. THE PRESENCE OF AN EXTERNAL FIELD OF MAGNITUDE H_E REDUCES THESE TO J_B AND J_B'. ON REMOVAL OF H_E, THE SOFT MATERIAL RECOILS ONLY TO J_C', SUFFERING A LOSS OF MAGNETISATION, WHILE THE HARD MATERIAL RECOVERS TO ITS ORIGINAL VALUE, J_A.

The magnets are required to have the spring characteristics shown in Figure 2(a), the desirable parameters being a high maxiumum force value combined with a sharp decline over the region CB - the latter feature is necessary when taking into account the behaviour at the opposite side of the spine. The magnetic forces must not cancel each other out. A computer was programmed to calculate the optimum magnet layout:, an explanation of the basis for these calculations is given, with some examples, in Appendix I.

The results indicated that opposing magnets with square cross sections of the same dimensions give the best force per unit weight at contact; with one magnet assumed infinitely long (the depth of the circumferential tracks around the spine is less than 0.2% of their radius of curvature), the repulsive force is then proportional to the length of the opposing magnet. Furthermore, the force per unit weight varies inversely as the linear dimensions (2).

The system chosen uses relatively small magnets of square cross-section, thus giving both a high force to weight ratio at small clearances and a rapid decrease in repulsion as the clearances become large compared with the magnetic dimensions. This allows the best use to be made of magnetic material. The decision whether to opt for a homopolar or heteropolar arrangement (see Figure 4) depends on the relative proportions of the axial pitch P and the magnetic dimensions.

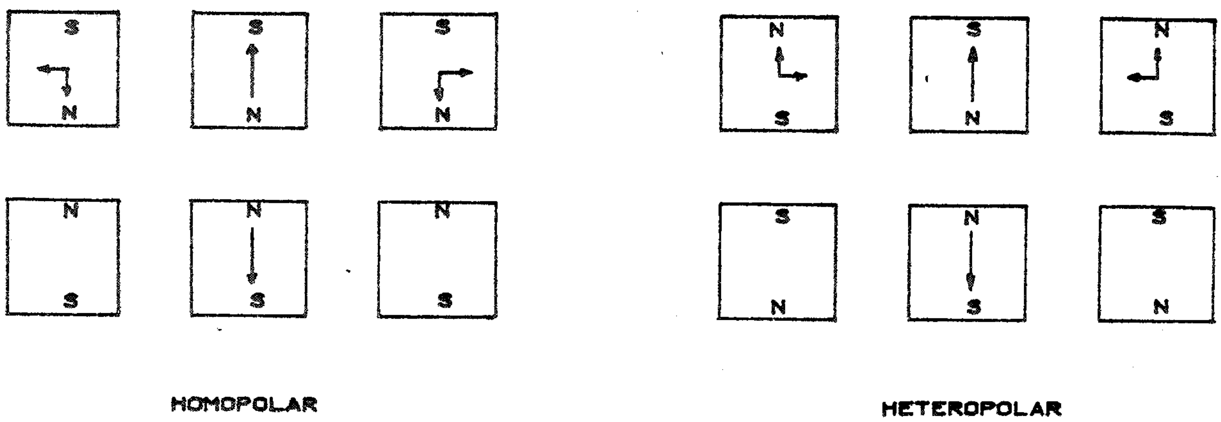
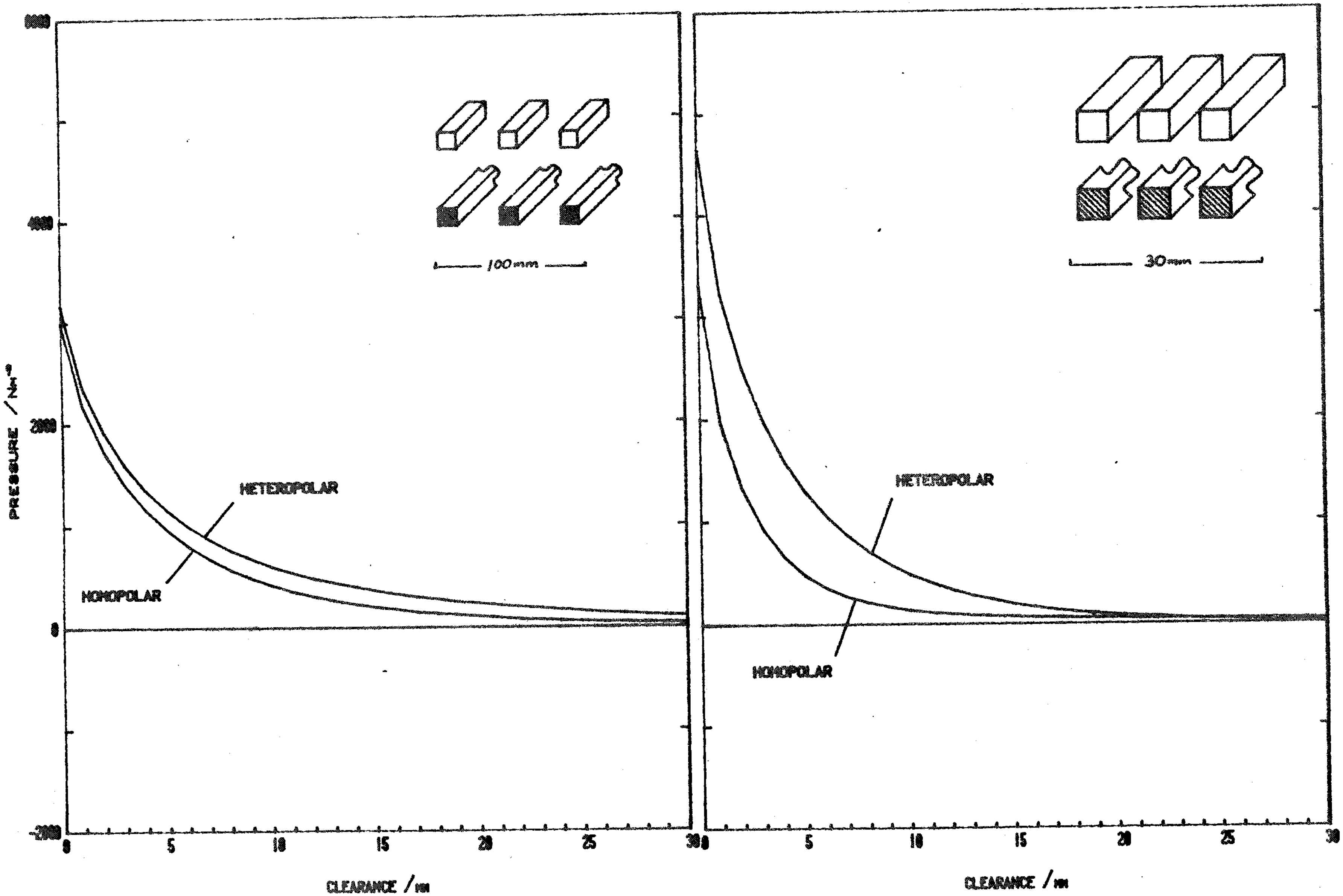


FIGURE 4. DIFFERENT ARRANGEMENTS FOR MAGNETIC REPULSION. THE ARROWS INDICATE THE FORCES PRODUCED BY THE INTERACTION OF EACH UPPER MAGNET WITH THE CENTRAL LOWER MAGNET. IN THE HOMOPOLAR CASE, THE OFF-CENTRE MAGNETS ARE ACTUALLY IN ATTRACTION WITH THE LOWER ONE, WHILE THE LATERAL FORCES CONTRIBUTE TO INCREASED STABILITY, TENDING TO RESIST ANY DEVIATIONS FROM THE ALIGNED POSITION SHOWN (INSTABILITY IS STILL INHERENT DUE TO THE INTERACTION OF THE CENTRAL PAIR).

IN THE HETEROPOLAR CASE, ALL THE MAGNETS ARE IN REPULSION, BUT NOW THE LATERAL FORCES ACT TO INCREASE THE INSTABILITY. SUCH AN ARRANGEMENT REPRESENTS THE MOST EFFICIENT USE OF MAGNETS IN REPULSION. EARNSHAW'S THEOREM DICTATES THAT THE BETTER THE REPULSION, THE MORE UNSTABLE IS THE SYSTEM IN SOME OTHER DIRECTION.



IGURE 5. COMPARISON OF HOMOPOLAR AND HETEROPOLAR MAGNETIC REPULSION SYSTEMS. IN THE FIRST CASE, THE LATERAL PITCH IS RELATIVELY LARGE, AND INTERACTIONS BETWEEN ADJACENT ROWS ARE SMALL: THERE IS LITTLE TO CHOOSE BETWEEN HOMOPOLAR AND HETEROPOLAR LAYOUTS. IN THE SECOND CASE, HOWEVER, THE ADVANTAGE OF A HETEROPOLAR SYSTEM AT SMALL PITCH VALUES IS SEEN. NOTE THAT ONLY PRESSURES ARE SHOWN HERE - THE DESTABILISING LATERAL FORCES ALSO INCREASE AS THE REPULSION DOES.

For small magnets at large pitch values, the two systems give similar force characteristics. This is because, perhaps unexpectedly, a magnet in the heteropolar system is actually in *repulsion* with the opposing magnets in adjacent rows. Similarly in the homopolar system, the interaction between adjacent rows leads to an attraction, thus decreasing the available lift. This effect can be explained by considering the angles involved (3), and is quantitively illustrated in Figure 5, depicting force *vs* clearance curves based on the mathematical model used in our calculations.

AXIAL STABILITY

Although permanent magnet repulsion systems can provide sufficient lift to support many times their own weight - in early experiments with ferrites, force-to-weight ratios of 30 were recorded (2) - they cannot do so in stable equilibrium: complete levitation by passive magnetic repulsion alone is impossible. Attempts to achieve stability by balancing attractive and repulsive forces likewise prove futile.

These are consequences of Earnshaw's theorem, originally formulated to explain the properties of ethereal particles, but equally applicable to magnetic poles (4); a short account of Earnshaw's theorem is contained in Appendix II.

The penalty, then, for making a bearing stronger in repulsion is to increase its instability in some other direction, with the need to make greater provision for lateral constraint. This is illustrated in Figure 6, in a comparison of calculated pressures and axial forces at constant clearance, but varying offset.

In practice, the magnets' alignment will be threatened by such factors as the relative axial displacements of duck and spine, 'banana shaped' deformations arising from stresses imposed on the concrete due to creep, shrinkage or thermal expansion, and deviations in the parallelism of the magnetic tracks. The ideal system for maintaining lateral stability would somehow mutually align the duck and spine magnets regardless of the relative movements or deformations of the two structures.

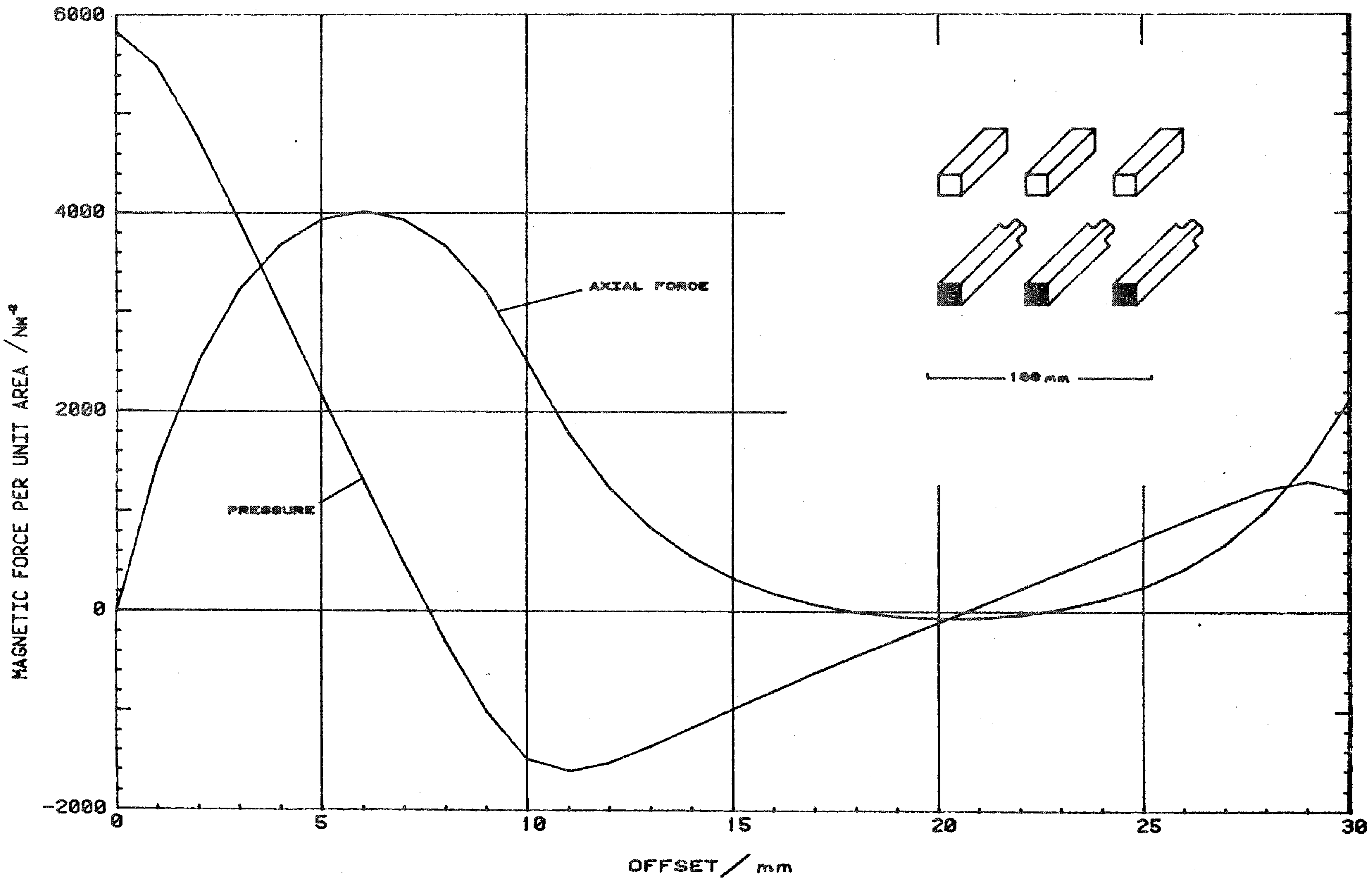


FIGURE 6. COMPARISON OF CALCULATED RADIAL (PRESSURE) AND AXIAL FORCES FOR THE MAGNET ARRANGEMENT SHOWN, AT A CONSTANT CLEARANCE OF 1mm, BUT VARYING AXIAL OFFSET. NOTE THE INITIALLY STEEP RISE IN AXIAL FORCE FOR SMALL OFFSET VALUES, AND THE REGION OF NEGATIVE PRESSURE, IN WHICH THE MAGNETS HAVE GONE INTO ATTRACTION. THE DIAGRAM SHOWS THE MAGNETS IN THE ZERO OFFSET POSITION - THE CLEARANCE IS EXAGGERATED FOR CLARITY.

The proposed arrangement uses rolling bearings and self-aligning magnets, and is illustrated in Figure 7. It works by essentially de-coupling the axial motion of the slubber magnets from the duck, and coupling it instead to the spine.

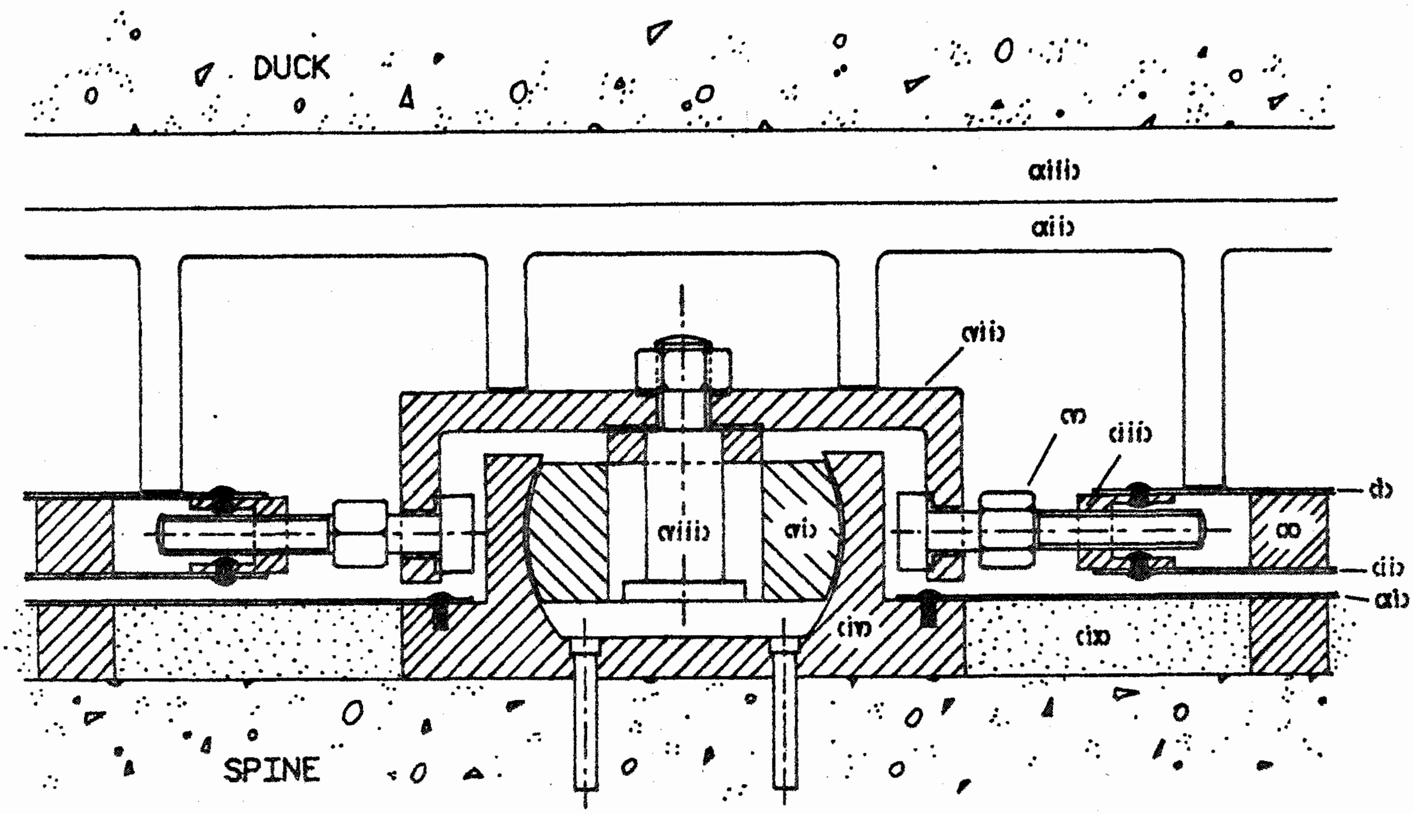


FIGURE 7. MAGNETIC SQUEEZE-FILM BEARING AXIAL ALIGNMENT DESIGN

(i, ii)	ELASTOMERIC SHEETS	(viii)	ROLLER SPINDLE
(iii)	SHEET-EDGE BATONS	(ix)	FILLER COMPOUND
(iv)	CIRCUMFERENTIAL RAILS	(x)	SLUBBER MAGNETS
(v)	SHEET TENSIONING SCREWS	(xi)	ELASTOMERIC SHEET
(vi)	ROLLING WHEEL	(xii)	FOAM PLASTIC BELLOWS
(vii)	U-SHAPED CHANNEL	(xiii)	SOFT FOAM LAYER

The elastomeric sheets (i) and (ii) terminate at either edge on batons (iii), while anchored around the spine are circumferential rails (iv) with axial pitch corresponding to the sheet width. Sheet tensioning screws (v) connect the batons to an inverted U-shaped channel (vii) positioned above the rail. Rubber wheels (vi) are mounted on spindles (viii) attached at intervals along the length of the channel. The wheels can make contact with the inner surfaces of the spine rail, and by doing so restrict the axial movement (but *not* the circumferential movement) of the slubber magnets (x).

In operation the slubber layer will glide over the spine with no contact occuring other than at the rolling bearings. Any potential offset of the magnets is limited to that caused by the shear force transmitted by the bellows layer, or by local misalignment of the rails. The first of these will be small, as the foam cell sheet is not actually attached to the upper surface of the magnet 'sandwich', and has little strength in shear anyway. Local deviations in the parallelism of the rails will be compensated for by the action of the sheets stretched between them. These are pre-tensioned and will stretch or relax in accordance with the pitch deviations of the rails; as a result, the moving magnets' pitch changes too but *remains a constant fraction of the rail pitch*. The principle is that of a 'lazy tongs': dividing unequal lengths into equal numbers of sub-lengths.

To ensure correct working of this technique, the spine magnets must also be pitched to allow for rail deviations incurred both during construction, and as a result of dynamic strain (due to thermal expansion, creep, shrinkage or bending).

The circumferential spine rails will be laid parallel to within a fair tolerance - the alignment of upper and lower magnets must be more accurate. To achieve this, the instrument used during construction to lay the spine magnets will operate on the same "lazy tongs" principle as the slubber alignmen system: the difference being that the spine magnets will be *permanently* pitched to compensate for non-parallelism, while the slubber magnets will be continuously, dynamically, compensating.

Side forces on a rail will only arise when an increase in the sheet tension (of the duck magnetic sandwich) on one side is not matched on the other; non-parallelism of the rails will have this effect, but the deviations will be small and average out over short distances. Because they are connected in series, pre-tensioning the sheets will only exert lateral force on those rails at the outermost ends of the bearing. These are a special case and will be strengthened accordingly, as will those for two rows inside them, as a safety measure.

Although the sources of dynamic strain mentioned above will almost certainly deform the spine in a non-uniform manner, the total differential strain will not exceed about 0.03% (10mm over the length of the spine); the deformation will be manifest as a banana bending, giving rise to small changes in the standing tension of the elastomeric sheets. Changes in the rail pitch will likewise be small, while the all-important magnet alignment will not be affected at all.

SURVIVAL OF THE BEARING

To survive a target design life of 25 years, the bearing must be resistant to:-

(i) wear and abrasion

(ii) corrosion or other chemical degradation

(iii) the effects of marine fouling

Wear and Abrasion

In normal operation, the bearing surfaces will not touch. The elastomeric sheets will nevertheless have high abrasion and scratch resistance as a safety measure; a synthetic-fibre reinforced polyurethane is one contender for this application, combining good wearing properties with an elastic modulus in the required range. The only points of contact will be between the rollers and the raceway (comprising the spine rails and sheet-edge runners). Selection of materials for these components will depend on the magnitude of the forces encountered: likely choices are hard plastic/rubber for the rollers and glass-reinforced plastic (G.R.P.) for the raceway.

Chemical Degradation

The presence of seawater in the bearing clearance dictates that the bearing components should not corrode, oxidise, dissolve or denature. Plastics may be chosen accordingly, and their lifespan will be greatly increased by the absence of three important factors known to accelerate degradation, namely, light, high temperature and marked temperature variation - the bearing operates in complete darkness in sea temperatures averaging $10^{o}C$, with an annual temperature range of less than $6C^{o}$.

The magnetic material contains iron and barium in their most highly oxidised states (barium ferrite is rust in all but name) and can last indefinitely in a marine environment. Similarly attractive are the qualities of G.R.P. materials such as isophthalic polyester resin reinforced with unidirectional glass fibre. The Royal Navy is currently operating vessels in which such materials have replaced steel in the hull construction.

Marine Fouling

Marine fouling poses the greatest threat to the bearing's operation. Although certain algal slime would act as super-lubricants, the presence of hard shelled species on the bearing surface would produce surface contact, increase friction, and cause permanent damage. An anti-fouling treatment is required which completely prevents both the initial settlement and subsequent growth of such marine organisms.

The fouling problem has been comprehensively summarised by Picken and FitzGerald of the Scottish Marine Biological Association (5). They record that *continuous* low-level chlorination in concentrations as low as 0.02 parts per million has proved an effective deterrent against the settlement and growth of hard fouling species. Chlorine may be used as a biocide at much higher concentrations than this without detriment to the environment.

Two of the most suitable methods of administering chlorine are (a) in the form of concentrated sodium hypochlorite solution, stored in tanks on board the duck string and diluted *in situ* and (b) continuous generation by electrolysing seawater. Hypochlorite introduces handling problems and has a limited shelf-life, requiring periodic replenishment. Electrolysis, used in conjunction with some method of preventing the build-up of by-products (e.g. calcareous deposits), represents a very attractive method for continuous low-level chlorination. Electricity is readily available, and the amount of chlorine required can be found knowing the volume and retention half-life of the water in the bearing, and the active life of the biocide. Preliminary calculations suggest a very small chlorine requirement, equivalent to about 8 grammes of the biocide continuously active in the bearing.

Protection must also be afforded against the entry of foreign matter such as seaweed and flotsam into the open ends of the bearing: for this purpose a suitable impediment is provided. The only paths for water to flow into or out from, the squeeze-film bearing exist at the outer edges of the duck (where rolling bearings supply the means of its gross axial location on the spine). A partial seal exists between the duck edge and the raised spine flange at this point, maintained by a stiffened rubber skirt acting like a draught excluder. Water will have difficulty flowing past this impedance, but more significantly, will have little incentive to do so: any wave load which causes a build-up of water pressure inside the squeeze-film bearing will give rise to a similar pressure increase on exposed parts of the spine in that vicinity. With only small pressure gradients across the "seal" there is little tendency for water to leak in or out. This in turn helps maintain the chlorine levels inside the squeeze-film bearing, while simultaneously preventing the inflow of oxygen and organic nutrients.

A final word can be said regarding the life expectancy of the elastomeric sheets which constitute the bearing inner surfaces. In 25 years they will travel relative to each other the equivalent of 125,000 miles at an average speed of about 3 mph. They will operate in conditions of low, almost constant, temperature and complete darkness. Loading will be zero, except in conditions as yet unobserved in nature (when it will be very small), and continuous water lubrication is assured.

A parallel might be drawn with the rubber tyres on a roadgoing vehicle - these are subject to some degree of sliding friction and generally operate on rough, dry surfaces rather than truly rolling on smooth wet ones. Even allowing for the harsher conditions, in countries where severe speed restrictions apply - e.g. 20 mph in Bermuda - and car tyres are consequently spared the worst rigours of cornering and braking, they generally last for as long as the car does. Indeed in Bermuda, where there is no second-hand car market, this may entail decades of continuous use. Better quantitative comparisons will be forthcoming once somebody devises a way of aquaplaning a solid-tyred car at 3 mph in complete darkness non-stop for 25 years.

APPENDIX I

THEORETICAL CALCULATION OF THE REPULSIVE FORCES BETWEEN PERMANENT MAGNETS

This problem may be approached from several directions employing different mathematical models; McCaig has described the commonest of these and discussed the relative merits of each (3). The permanent magnet property pertinent to repulsion calculations is J, the magnetic polarisation or magnetisation vector (some inconsistency surrounds this point, J also being referred to as the intrinsic flux density, but in this treatment SI units are used throughout, and J will be called the magnetisation with unit the tesla (T)).

Calculations may be greatly simplified if, for the magnetic material in question, the value of J is assumed uniform throughout its volume. This is justified when dealing with hard magnetic materials - such as the anisotropic barium ferrite used in the squeeze-film bearing - whose demagnetisation curves approach the ideal "square loop" (see page 6, this report). Although some correction must be applied to allow for the demagnetising influence of repelling magnets on each other, a reasonable first approximation may be obtained in this manner.

The model employed represents permanent magnets as uniform solenoids of rectangular cross-section, their magnetisation giving rise to a surface current density vector $\vec{K}$, where:-

$$\vec{K} = (\vec{J}/\mu_0) \times \vec{n} \qquad \text{(i)}$$

μ_0 is the magnetisation constant ($4\pi \times 10^{-7}$ in SI units) and $\vec{n}$ the unit vector normal to the appropriate face. This model is described by Tsui *et al* (6), and has also been employed by Borcherts in a theoretical analysis of permanent magnet levitation for track vehicles (7).

Two magnets arranged in repulsion are shown in Figure 8, together with a representation of the equivalent solenoids. The net attraction or repulsion is found by integrating the force contributions due to interactions between current conducting elements in the two solenoids. The incremental force $\vec{dF}$ between two such elements $\vec{dl}$ (lower) and $\vec{dl}'$ (upper), carrying

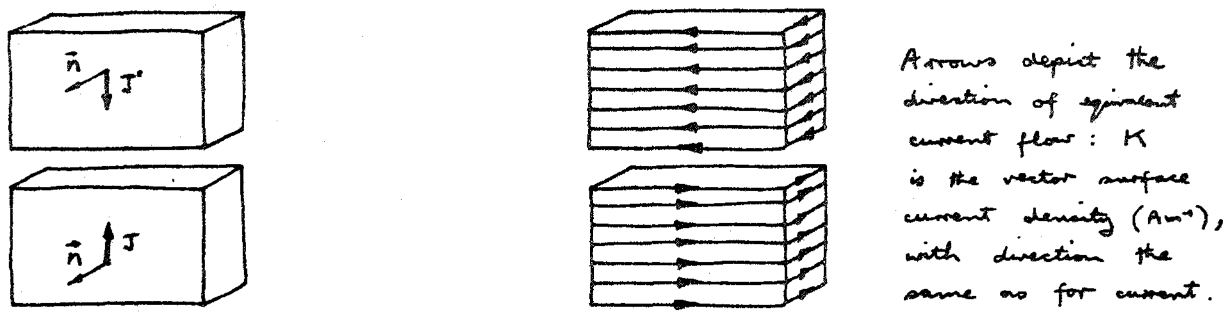


Figure 8. Repulsion magnets and equivalent solenoids.

currents of i and i' respectively is given by:-

$$\vec{dF} = \frac{\mu_0}{4\pi} \frac{i'\vec{dl}' \times (i\vec{dl} \times \vec{r})}{|\vec{r}|^3} \qquad \text{(ii)}$$

in which $\vec{r}$ is the position vector from $\vec{dl}$ to $\vec{dl}'$ (8).

This leads to an expression for the total force exerted on one complete current loop by another:-

$$\vec{F} = \oint\oint \frac{\mu_0 i i' \vec{r} (\vec{dl} \cdot \vec{dl}')}{4\pi |\vec{r}|^3} \qquad \text{(iii)}$$

to be integrated with respect to $\vec{dl}$ and $\vec{dl}'$ around the two closed circuits.

If the currents i and i' are now substituted with current *sheet densities* K and K', then the incremental lengths dl and dl' must be replaced by incremental *areas*. This is shown ln Figure 9 in an illustration of the interaction between two such area elements, dA and dA'. Note that dA may be replaced with $dxdy$, and dA' with $dx'dy'$.

Where the currents flow antiparallel, the resulting magnetic force is repulsive (eg. between sides A and A'); where they flow parallel (eg. sides A and C'), it is attractive. Currents flowing at right angles (eg. sides A and B'), do not interact. The total force F_T may be expressed in terms of the interfacial forces:-

$$F_T = \underbrace{F_{AA'} + F_{BB'} + F_{CC'} + F_{DD'}}_{\text{repulsion}} - \underbrace{(F_{AC'} + F_{BD'} + F_{CA'} + F_{DB'})}_{\text{attraction}} \qquad \text{(iv)}$$

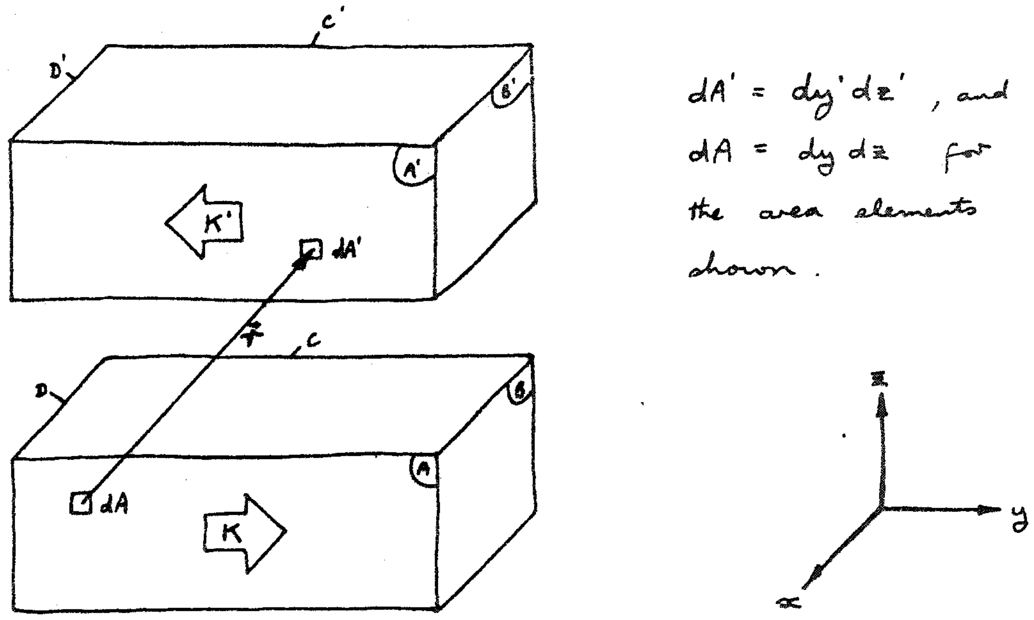
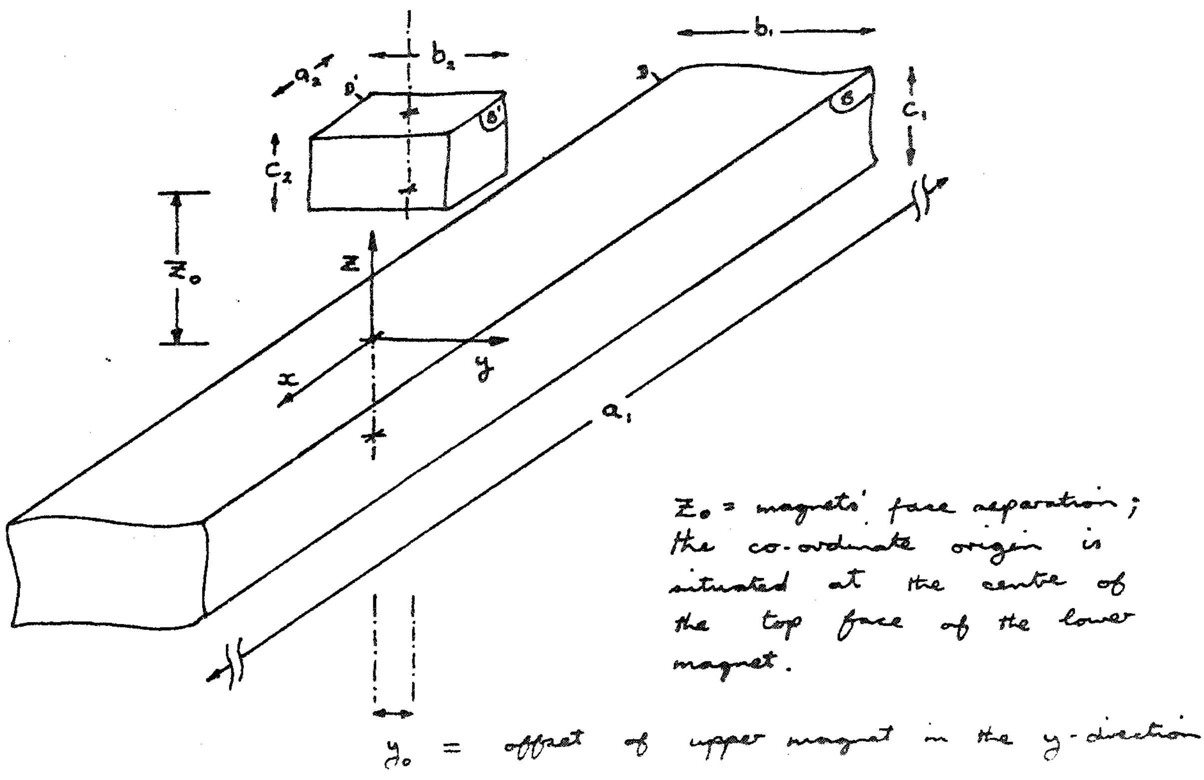


Figure 9. Interaction between area elements of solenoids.

In the case of a small magnet in levitation above a very long magnetic track, the lengths of sides B and D of the lower magnet being assumed much greater than B' and D' of the upper one, the repulsion forces $F_{AA'}$ and $F_{CC'}$ and the attractions $F_{AC'}$ and $F_{CA'}$ become negligible, and may be omitted from equation (iv). We now have:-

$$F_T = F_{BB'} + F_{DD'} - (F_{BD'} + F_{DB'}) \qquad \text{(v)}$$

This case is illustrated below in Figure 10.



From equation (iii) a general expression can be obtained for the force between two interacting sides P (lower) and Q' (upper) in this modified case:-

$$F_{PQ'} = \frac{\mu_0 K K'}{4\pi} \int_{z_0}^{z_0 + c_2} \int_{-c_1}^{0} \int_{-\frac{a_2}{2}}^{\frac{a_2}{2}} \int_{-\frac{a_1}{2}}^{\frac{a_1}{2}} \frac{\vec{r}_{PQ'}}{|\vec{r}_{PQ'}|^3} \, dx \, dx' \, dz \, dz' \qquad \text{(vi)}$$

From equation (i), K and K' may be replaced by J/μ_0 and J'/μ_0 ; in this calculation both magnets are assumed to be composed of the same material, so the constant term becomes $J^2/4\pi\mu_0$ (viii).

Equation (vi) may be used to represent each of the terms in expression (v) in turn, the limits of integration being the same in all four cases. The general position vector $\vec{r}_{PQ'}$ must be replaced in each case by $\vec{r}_{BB'}$, $\vec{r}_{DD'}$, $\vec{r}_{BD'}$, $\vec{r}_{BD'}$, or $\vec{r}_{DB'}$ as appropriate. Furthermore, each term $F_{PQ'}$ must be separately evaluated *twice* in order to obtain both the horizontal and vertical components. Although the horizontal components cancel when the magnets are in perfect lateral alignment, it is required to find their magnitude when the upper magnet becomes offset in the y-direction. If the general position vector $\vec{r}_{PQ'}$ is given by:-

$$\vec{r}_{PQ} = (x' - x)\vec{n}_x + b\vec{n}_y + (z' - z)\vec{n}_z \qquad \text{(ix)}$$

Where $\vec{n}_x$, $\vec{n}_y$ and $\vec{n}_z$ are the unit vectors along the reference axes, then the values of b , the y-component of each position vector, will be:-

position vector	value of b	
$\vec{r}_{BB'}$	$y_0 + \frac{b_2}{2} - \frac{b_1}{2}$	
$\vec{r}_{DD'}$	$y_0 - \frac{b_2}{2} + \frac{b_1}{2}$	(x-xiii)
$\vec{r}_{BD'}$	$y_0 - \frac{b_2}{2} - \frac{b_1}{2}$	
$\vec{r}_{DB'}$	$y_0 + \frac{b_2}{2} + \frac{b_1}{2}$	

where y_0 is the lateral (y-direction) offset of the upper magnet.

From equation (vi) the vertical force component $F_{pq'}$ for the general case is given by:-

$$F_{p\phi'}(z) = \frac{J^2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \int_{-c_1}^{0} \int_{-\frac{a_2}{2}}^{\frac{a_2}{2}} \int_{-\frac{a_1}{2}}^{\frac{a_1}{2}} \frac{(z'-z)\,\vec{n}_z \quad dx\,dx'\,dz\,dz'}{\left[(x'-x)^2 + b^2 + (z'-z)^2\right]^{3/2}} \qquad \text{(xiv)}$$

and the horizontal force component:-

$$F_{p\phi'}(y) = \frac{J^2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \int_{-c_1}^{0} \int_{-\frac{a_2}{2}}^{\frac{a_2}{2}} \int_{-\frac{a_1}{2}}^{\frac{a_1}{2}} \frac{b\,\vec{n}_y \quad dx\,dx'\,dz\,dz'}{\left[(x'-x)^2 + b^2 + (z'-z)^2\right]^{\frac{3}{2}}} \qquad \text{(xv)}$$

Dealing firstly with $F_{pq'}(z)$, and integrating with respect to x (lower magnet length) gives:-

$$F_{p\phi'}(z) = \frac{J^2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \int_{-c_1}^{0} \int_{-\frac{a_2}{2}}^{\frac{a_2}{2}} \frac{(z'-z)}{\left[(z'-z)^2+b^2\right]} \left\{ \frac{(x'+\frac{a_1}{2})}{\left[(z'-z)^2 + b^2 + (x'+\frac{a_1}{2})^2\right]^{\frac{1}{2}}} - \frac{(x'-\frac{a_1}{2})}{\left[(z'-z)^2 + b^2 + (x'-\frac{a_1}{2})^2\right]^{\frac{1}{2}}} \right\} dx'\,dz\,dz' \qquad \text{(xvi)}$$

If the lower magnet length a_1 is now extended to infinity to simulate a track, an expression can be obtained for the limiting force $\lim\limits_{a_1 \to \infty} F_{pq'}(z)$:-

$$\lim_{a_1 \to \infty} F_{p\phi'}(z) = \frac{J^2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \int_{-c_1}^{0} \int_{-\frac{a_2}{2}}^{\frac{a_2}{2}} \frac{2\,(z'-z)}{\left[(z'-z)^2 + b^2\right]}\, dx'\,dz\,dz' \qquad \text{(xvii)}$$

Now integrating with respect to x' (upper magnet length) gives:-

$$\lim_{a_1 \to \infty} F_{pq'}(z) = \frac{J^2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \int_{-c_1}^{0} \frac{2a_2 (z'-z)}{[(z'-z)^2 + b^2]} \, dz \, dz'$$

(xviii)

Integrating with respect to z (lower magnet depth):-

$$\lim_{a_1 \to \infty} F_{pq'}(z) = \frac{J^2 a_2}{4\pi\mu_0} \int_{z_0}^{z_0+c_2} \ln\left[(z'+c_1)^2 + b^2\right] - \ln\left[z'^2 + b^2\right] dz'$$

(xix)

Finally integrating with respect to z' (upper magnet depth):-

$$\begin{aligned}\lim_{a_1 \to \infty} F_{pq'}(z) = \frac{J^2 a_2}{4\pi\mu_0} \Big\{ & (z_0 + c_1 + c_2) \ln\left[(z_0+c_1+c_2)^2 + b^2\right] \\ & - (z_0 + c_1) \ln\left[(z_0+c_1)^2 + b^2\right] \\ & - (z_0 + c_2) \ln\left[(z_0+c_2)^2 + b^2\right] + z_0 \ln\left[z_0^2 + b^2\right] \\ & + 2b\Big[\tan^{-1}\left(\frac{z_0+c_1+c_2}{b}\right) - \tan^{-1}\left(\frac{z_0+c_1}{b}\right) \\ & - \tan^{-1}\left(\frac{z_0+c_2}{b}\right) + \tan^{-1}\frac{z_0}{b}\Big]\Big\}\end{aligned}$$

(xx)

The corresponding expression for the sideways force component between two faces is found by taking the limit of $F_{pq'}(y)$ (equation (xv)) as $a_1 \to \infty$.

This gives the fully integrated expression:-

$$\lim_{a_1 \to \infty} F_{pq'}(y) = \frac{J^2 a_2}{2\pi\mu_0} \left\{ (z_0+c_1+c_2)\tan^{-1}\frac{(z_0+c_1+c_2)}{b} - (z_0+c_1)\tan^{-1}\frac{(z_0+c_1)}{b} - (z_0+c_2)\tan^{-1}\frac{(z_0+c_2)}{b} + z_0\tan^{-1}\frac{z_0}{b} + \frac{b}{2}\ln\left[\frac{[b^2+(z_0+c_1)^2][b^2+(z_0+c_2)^2]}{[b^2+(z_0+c_1+c_2)^2][b^2+z_0^2]}\right]\right\}$$

(xxi)

By summing the appropriate four forces in expression (v), the total vertica or horizontal force exerted on the upper magnet by the track can be found, for any given combination of clearance z_0 and lateral offset y_0. Because the track is assumed to be infinitely long, there is no x-component of force.

A microcomputer was programmed to calculate the forces between a track of variable input dimensions (b_1 and c_1) but infinite length, and a supported block of variable dimensions (a_2 , b_2, and c_2) at any chosen values of y_0 and z_0. J was assumed equal for both magnets, the value to be selected for the particular magnetic material under study.

By suitably modifying the program, it was possible to calculate the forces between an array of parallel tracks and supported magnets: the calculation had to take into account every interaction between a track and an upper magnet. The rails' lateral pitch was assumed uniform, with all rails having the same cross-sectional dimensions; the supported magnets were likewise assumed to be identical. A further adaptation involved the introduction of a longitudinal pitch value for the upper magnets, assuming many to be supported above each track. The forces on a supported block can be seen from expressions (xx) and (xxi) to be directly proportional to its length a_2, hence the losses incurred b increasing the longitudinal pitch can easily be calculated. Furthermore, by taking into account the lateral pitch of the tracks as well, the total levitatio force available from a given *area* of track (including spaces) - ie. the pressure the system will bear - may be calculated. Results showing pressure *vs* clearance and axial force per unit area *vs* clearance can be seen in Figures 5 and 6 (main text), and overleaf in Figure 11.

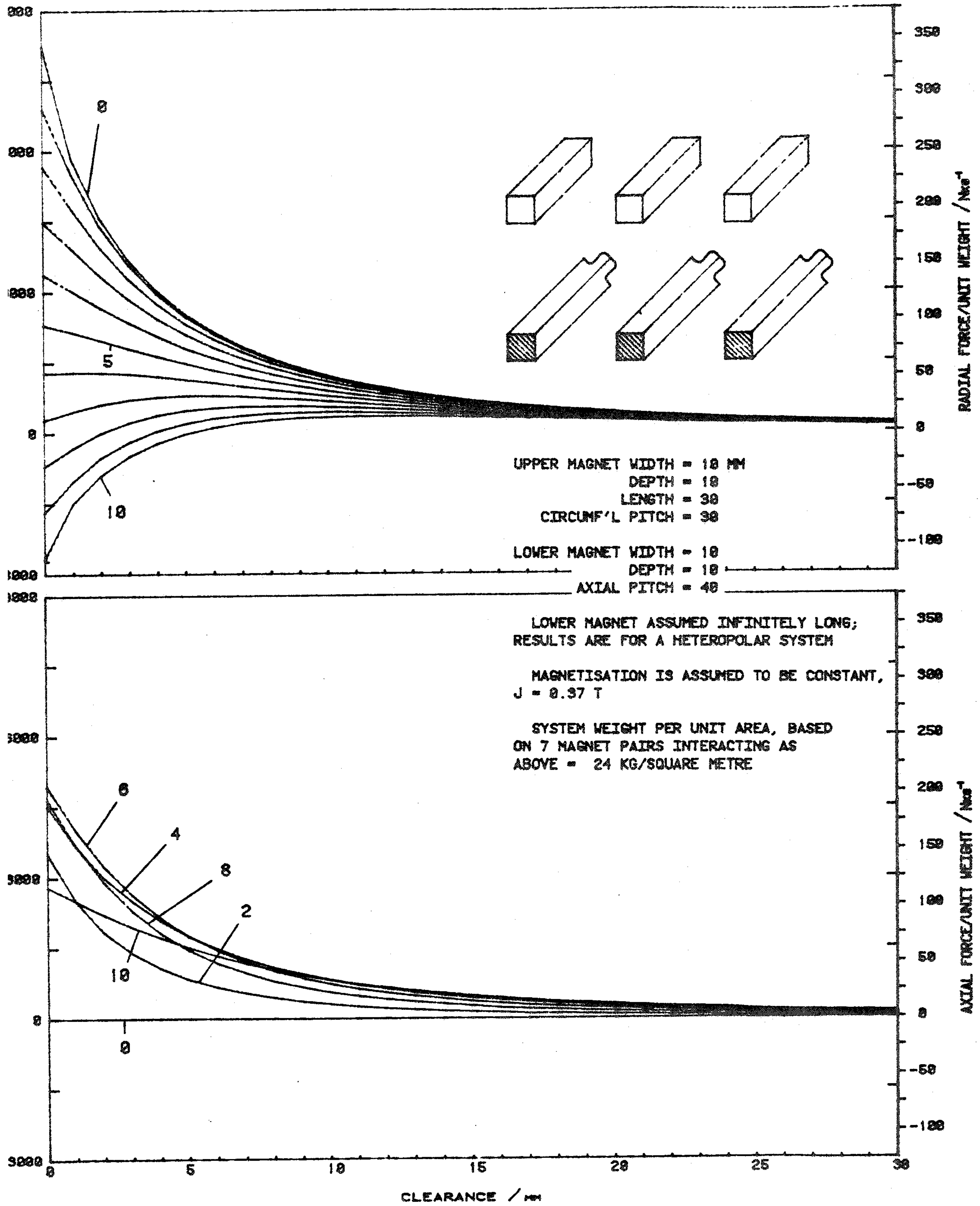


FIGURE 11. MAGNETIC TRACK CHARACTERISTICS CALCULATED USING THE METHOD DESCRIBED IN APPENDIX I. EACH CURVE IS OBTAINED AT A CONSTANT OFFSET VALUE, IN THE RANGE 0-10MM.

The results obtained in this mathematical treatment represent the *maximum* available forces for the systems described. Losses must be expected, due to the demagnetising fields mutually exerted by repulsion magnets in close proximit The magnetisation (J) of each is a function of the demagnetising field it experiences, while the field it exerts depends on its magnetisation: the problem is cyclical and most commonly solved using iterative techniques. Tsui *et al* (6) employed such a method with considerable success in a comparison of repulsio using various different permanent magnet materials. They also found that barium ferrites, though not the most powerful repulsion magnets, gave forces onl slightly less than the maximum calculated values, even after forced contact (whence are encountered the maximum demagnetising fields).

APPENDIX II

SUMMARY OF EARNSHAW'S THEOREM*

Complete levitation by passive magnetic repulsion would require that a point magnetic pole, or collective arrangement of poles (in the form of a permanent magnet), be suspended at a position of minimum potential energy. If a magnetic field were to exist at a point fulfilling this condition then, denoting as V the scalar magnetic potential of the pole(s), and using rectangular co-ordinate notation, the following must be true:-

$$\frac{dV}{dx} = \frac{dV}{dy} = \frac{dV}{dz} = 0$$

A further condition for a potential minimum is that the second derivative of V be *positive*.

The continuous nature of magnetic field lines in free space is implied in Laplace's equation:-

$$\frac{d^2V}{dx^2} + \frac{d^2V}{dy^2} + \frac{d^2V}{dz^2} = 0$$

in which at least one term must be positive and one negative - hence the three second derivatives of V cannot all be positive, with the result that the pole(s) must be unstable with respect to displacement in at least one direction.

Furthermore, in Earnshaw's words, "The equation $d_f^2V + d_g^2V + d_h^2V = 0$, from which the instability arises, holds equally for attraction and repulsion. It may be observed also that the instability cannot be removed by *arrangement*; for though the values of d_f^2V, d_g^2V, d_h^2V depend upon the arrangement of the particles, the fact that one at least must be positive and one negative depends only upon the equation $d_f^2V + d_g^2V + d_h^2V = 0$, which is true for every arrangement. And consequently, whether the particles be arranged in cubical forms, or in any other manner, there will always exist a direction of instability".

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*The original version may be found in Reference 4.

APPENDIX III

SECURING THE SPINE MAGNETS

The method of attachment of the ferrite magnetic blocks to the spine surface is of some importance and deserves further comment. These magnets must exhibit no tendency to work free during the lifetime of the bearing.

The upper (duck) magnets, sandwiched between two elastomeric sheets at the base of the slubber layer, are well secured: one possibility is that a similar sandwich - without any holes through it - be overlaid on the spine, as in Figure 1.

Another arrangement - shown below in Figure 12 - would involve spine magnets with a shallow channel along the length of their upper surface; a thin belt made of synthetic fabric is recessed in this, and is tightly wound around the spine circumference, restraining the magnets.

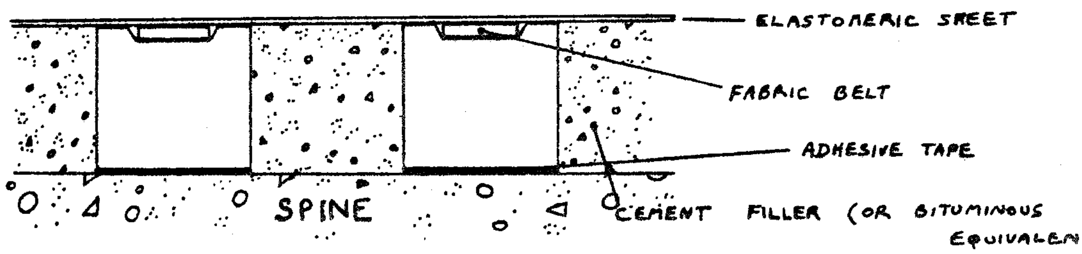

Figure 12. Method of attachment of spine magnets.

Further hindrances to their possible detachment are provided by the elastomeric sheet stretched over their upper surface, and the double sided adhesive tape on the surface of the spine itself. The repulsion forces encountered in operation will tend to *increase* the adhesion of the magnets to the spine, while a suitable filler compound - most probably cement based - will prevent any potential axial dislocation.

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(7) R.H. Borcherts, "Mathematical Analysis of Permanent Magnet Suspension Systems", *J. Appl. Physics*, Vol. 42, No. 4, March 1971.

(8) A.F. Kip, *Fundamentals of Electricity and Magnetism*, 2nd edn., Chapter 8, McGraw-Hill Kogakusha, Tokyo, 1969.

4.5 Spine Joints

4.5.1 Hookes Joint

The preferred design for this unit comprises water lubricated laminated spherical steel/rubber bearings of the type produced by Andre Rubber Co. Such bearings, based on existing technology, can be relied upon for a long maintenance-free life.

An alternative basis of the Hookes joint is a series of 4 x 1250 mm diameter bore SKF spherical plain bearings of the steel /PTFE composite type, type designation GEC 1250 fsa. These bearings have been specifically designed to be maintenance free and SKF quote cases of their use in very arduous conditions such as a steel works where they have given a life of 20 years or more without any attention whatsoever.

In the case of the duck their duty is not nearly so hazardous in that they are not exposed to any hostile environment.

It should be noted that this joint will be made only in calm water or ashore. The connection at sea is the fixed joint flange at mid-spine (see Drawing No. 10102).

4.5.1 (Continued).

Design Equations for Control of Spine Joint

The system equations are as follows:

M	=	Joint moment
A	=	Total ram area
L	=	Ram spacing
P_1	=	Ram pressure
P_2	=	Main pressure
θ	=	Angle of 1st swash plate
V_1	=	Displacement per revolution per radian swash
ϕ	=	Angle of 2nd swash plate
V_2	=	Displacement per revolution per radian swach
Q_1	=	Ram flow rate
Q_2	=	Output flow rate
$\dot{\alpha}$	=	Joint angular velocity
ω	=	Shaft speed in radians/sec
T	=	Shaft torque

$$M = P_1 AL = \frac{P_2\ \phi V_2 AL}{\theta V_1}$$

$$T = \frac{P_1 \theta V_1}{2\pi} = \frac{P_2\ \phi V_2}{2\pi}$$

$$Q_1 = \dot{\alpha} AL = V_1 \theta \omega$$

$$\omega = \frac{\dot{\alpha} AL}{V_1 \theta} = \frac{Q_2}{V_2 \phi}$$

$$\therefore\ Q_2 = \frac{\dot{\alpha} ALV_2 \phi}{V_1 \theta}$$

4.5.2 Rams

The rams needed are 610 mm bore at 6000 lbs per square inch with a 2.2 m stroke. These rams fall well within the normal commercial range of a number of firms. We have in fact selected a Bradford Cylinder design and a quotation for a ram with a suitable life is appended hereto.

4.5.3 Power Take-Off

In the original report on this Wave Energy Device dated November 1979, it was suggested that three aspects of the power take-off required further examination:-

1. The Flywheel/Gyroscope
2. Swashplate Hydraulic Motor
3. The Ring Cam Pumps

All environmental problems on all the above equipment are solved by virtue of the enclosure within the power canister at low atmospheric pressure using inert gas to insulate the entire system from the hazardous, salty conditions of the surroundings

4.5.3 (i) Flywheel

This is the subject of a separate contract being undertaken by John Laing for the Department of Energy and a full report will be issued in due course; however work has proceeded sufficiently far to be able to state that a flywheel of this configuration is feasible.

(ii) Swashplate Motor

(Drawing No. 10083 refers)

The Robert Clerk design of pump/motor embodies many novel features directed at achieving long life and high efficiency with a high speed, rapid response unit. Towler Hydraulics of Leeds have had an opportunity of assessing the potential of the design while also running a half-size unit. They state that, with sufficient funding, there is no reason why a suitable unit should not be developed within two years. This view is borne out by a separate report by Commercial Hydraulic Limited of Hucclecote commissioned by ETSU direct.

Efficiency calculations for the Robert Clerk motor are appended hereto.

4.5.3 (ii)

PUMP/MOTOR EFFICIENCY CALCULATION

Hydraulic Losses

Notation:
P = pressure.
S = leakage perimeter or one dimension of a shear area.
ε = clearance.
μ = absolute viscosity.
L = leakage path length or a second dimension of a shear area.
V = shear velocity.
Q = chamber volume.
K = bulk modulus.
n = operations per second.

In high pressure oil systems fluid flow velocities are low, laminar flow can usually be assumed and fairly accurate loss calculations can be made. They fall into three classes:

1. Leakage. Energy is dissipated as fluid flows from a high pressure source through the clearances round the pistons, valves, port plates etc.

$$\text{Leakage power} = \frac{P^2 S \varepsilon^3}{12\mu L}$$

We need to keep clearances small while viscosity and peakage paths are large.

2. Shear. Energy is dissipated as a result of the shear force and relative velocity between two adjacent moving areas.

$$\text{Shear power} = \frac{SL\mu V^2}{\varepsilon}$$

We need viscocity and areas low while clearances are large in conflict with the requirements for low leakage.

3. Compressibility. Energy is stored as a result of the finite bulk modulus of the hydraulic fluid. This energy may be recovered by intelligent design but is sometimes destructively dissipated when

a chamber is connected to the wrong pressure.

$$\text{Compressibility power} = \frac{QP^2n}{2K}$$

We need to keep chamber volumes small and try to use a high bulk modulus oil. Vaccuum stripping helps. Proper timing control can eliminate the compressibility losses except at low swash angles in axial piston units where the control angles become impractically large.

It usually turns out that the lowest losses are achieved if leakage and shear losses are made equal. The value of viscosity is uncertain because of its temperature dependence. But at the point where shear and leakage are equal, moderate variations in viscosity have little effect because the two loss changes cancel one another.

The most important single factor is clearance because of its cube law effect on leakage. All the skills of the designer should go into getting the clearance correct despite the distortions that can occur when the units are under pressure. By making our system pressure proportional to gyro spin-speed, we maintain the equality of leakage and shear over the entire range of sea states. Apart from the constant shear loss of the axial piston unit which drives the electrical generator and the generator's own iron losses, we will then achieve a constant conversion efficiency and avoid the disproportionately high losses which occurred at low power levels in the 1979 reference design.

Detailed losses are summarised on the following pages.

ROBERT CLERK 9/55 AXIAL PUMP/MOTOR

Number of cylinders	= 9
Piston/cylinder bore	= 55 m/m
Max. swash-plate tilt	= 24^{o} angle either way of centre
Max. piston stroke	= 151 m/m
Max. stroke displacement	= 358.9 cm^3 = 21.892 cu ins
Max. pump/motor displacement	= 3.23 litres = 197 cu ins
Max. delivery at 1500 rpm @ 0 psi	= 80.75 litres/sec = 1067 GPM
Max. theoretical power @ 3200 psi diff	= 1783 KW = 2391 HP
Residual cylinder volume @ TDC	= 27.41% of max. displacement (6 cu ins)
Working fluid absolute viscosity	= 21 centistoke @ op. temp. (3 microreyns)
Working fluid compressibility	= 1.477% @ 3250 psi
TDC "no-loss" timing angle	= 7 $1/4^{o}$ @ 3250 psi x max. stroke (24^{o} swash)
BDC anti-noise timing angle	= 15^{o} @ 3250 psi x max. stroke (24^{o} swash)
TDC "no-loss" timing angle	= 21^{o} @ 3250 psi x quarter stroke (6^{o} swash)
BDC anti-noise timing angle	= 26^{o} @ 3250 psi x quarter stroke (6^{o} swash)
"No-stroke" uncorrectable loss	= 1.14% @ 3250 psi x zero stroke ($\pm$ 3^{o} swash)

Timing varies automatically with pressure and stroke.

Stroke is controlled by modular programmable hydro-computer integrating multiple inputs to determine pressure/displacement/speed outputs.

Timing is controlled by secondary hydro-computer on tilt-box, integrating system pressure and counter-torque pressure in tilt box lobe motor to determine timing angle. Timing angle pressure signal is fed back to prime hydro computer where, by integration with system pressure, a feed-back signal of stroke-displacement is provided.

9/55 PUMP/MOTOR VOLUMETRIC AND SHEAR LOSSES (INTERIM)

@ 1500 RPM @ 3500 psi @ 3 microreyn (21 c/s)

Item	Nominal Clearance	Leakage c.i.p.s.	Leakage H. Power	Shear H. Power	Development Potential
Portface	.0005"	1.17	0.62	2.94	+ 8%
Contra-thrust	.001"	3.60	1.91	1.49	
Port-plate journal	.001"	1.26	0.67	0.56	
Piston Cylinder	.001"	1.3	0.69	3.09	
Swash thrust	.0007"	3.94	2.10	4.89	-50%
Spherical journal	.0014"	2.22	1.18	1.00	
Big ends	.001"	1.29	0.68	0.01	
Little ends	.0002"	-	-	0.03	
Spider Joints	.001"	1.82	0.97	9.12	-35%
Swash rocker	.001"	2.32	1.23	-	
Timing thrust	.0005"	1.17	0.62	-	
Timing torque	.001"	?	?	-	
Boost leakage (50 psi)	-	0.38	0.0035	-	
Controls requirement (600 psi @ 49.5% ε)	-	3.33	0.82	-	
Boost input (50 psi @ 12% ε)			1.50	-	
		23.80	12.99 HP	23.13 HP	

Dry interior scavenged by Boost suction, therefore no loss.

Total losses 36.12 HP = 1.5% of Theoretical Power (cf 2.2% in 9/25 P/M).

Compressibility of Pump delivery recoverable in like-Motor.

9/55 PUMP/MOTOR: 197 cipr 1500 RPM 3200 psi a diff = 2391 HP (THEOR.)

LOSS EXTRAPOLATION

Volumetric leakage does NOT vary with dimension EXCEPT insofar as CLEARANCE may alter.

Tested leakage of 9/25 pump was 0.6 GPM per 1000 psi
Therefore 1.92 GPM @ 3200 psi = 4.3 HP = 0.18% of 9/55 pump
* Assuming 50% clearance increase = 14.51 HP = 0.61%

Rotary Shear Losses (RSL) vary as $\text{Size}^4 \times \text{Speed}^2 \times \text{Clearance}^{-1} \times 3$.
Speed is the inverse of size so that Rotary Shear Losses are proportional to Size^2

9/25 pump RSL 2.95 HP (calc) 3.2 HP (test) @ 3000 RPM
* Therefore 9/55 pump RSL 11.52 HP (calc) 12.5 HP (test) @ 1500 RPM

Stroke Shear Losses very as $\text{Size}^2 \times \text{Stroke}^2 \times \text{Speed}^2 \times \text{Clearance}^{-1}$
Stroke varies with Size, but inverse to speed.
Therefore, SSL vary as Size^2

* Piston/Cylinder SSL $= \frac{0.637}{1.5} \times 2.2^2 = 2.06$ HP @ 1500 RPM

* Big End SSL (calc x 9) $= \frac{0.144}{1.5} \times 9 = 0.86$ HP @ 24° & 1500 RPM

Stroke Friction Lossses vary as $\text{Size}^3 \times$ Stroke Angle x Speed x Pressure
SFL vary as $\text{Size}^3 \times$ Speed x Pressure
* Little End and Spider Joints SFL = 9.12 HP @ 1500 RPM @ 3200 psi

TOTAL LOSSES: (Excl. Compressibility and Control)

Leakage @ + 50% Clearance	= 14.51 HP	= 10.82 kW	= 0.61%
Rotary Shear Losses	= 12.5 HP	= 9.33 kW	= 0.52%
Piston/Cycl. Shear Losses	= 2.06 HP	= 1.54 kW	= 0.086%
Big End Shear Losses	= 0.86 HP	= 0.64 kW	= 0.036%
Little End and Spider Joint F.L.	= 9.12 HP	= 6.8 kW	= 0.381%
	39.05 HP	29.13 kW	1.63%

RING CAM PUMP (DIRECT DAMPING)

Cam ring is basically single-planar (0^{o} Latitude) about an axis concentric with that of the cylindrical enclosing canister.

Its O/D is increased to 4500 and I/D to 4250 with symmetrical opposed camming faces (axial) 125 rolling width. It will have 27 lobes per face (vs 21 previously) each having a min. crest curvature of 384 mm. It will have an integral outward annular flange 40 mm thick (symmetrical about the radial plane) x 4750 for bushings for 50 mm pins to the torque-link mountings.

The camming surfaces all project to the point where the radial plane of symmetry meets the longitudinal axis. So too do the rolling surfaces of the follower rollers which have been increased to 200 median (vs 185 previous) with the bearing bore now 160 (vs 150) x 135 long x 32 off per side.

The cyclic max line stress at rolling contact is now 7880 lb/in (vs 4635) but the rolling velocity is now 5.38 ft/sec maximum cyclic peak (vs 18.4). The increase of acceleration loading due to the larger diameter piston (70 mm vs 50) and conrod is negligible in the present context. The power strokes per cylinder in 25 years reduces dramatically to 115.2×10^{6} (vs 239×10^{6} previously) and the power strokes per cam lobe even more so to 4.5×10^{6} (vs 11.38×10^{6} previously).

The 70 mm pistons x 127 mm stroke provide a displacement of 488.75 cc (29.83 cu in) and the residual cylinder head and sleeve volume is 352.4 cc (21.5 cu in) so that the effective displacement, as reduced by compressibility recovery is depleted by 1.06% at 3250 psi, reducing proportionately with pressure. In addition the working fluid is compressed 1.47% as delivered, but in neither case is this an energy loss. In any case there is 11% redundancy of ring-cam pumping elements, above the axial motor acceptance at rated power output.

The cam rings are tangent-link mounted in the enclosing canister and the pumping elements, ring mains and distributing connections are mounted on the gyro outer cage which will have high gyroscopic rigidity in space so that the duck canister and cam ring will have a to-and-fro rotation about the gyro cage at a maximum angular velocity of 0.75 rad/sec.

For the total displacement of 844.6 litre/rev the maximum delivery will therefore be 100.87 litre/sec for each cam ring, or 201.74 litres/sec per gyro or 403.5 litre/sec per gyro pair corresponding to max. overload output which at 3250 psi equates to 9054 kW (Th) - 2.53% compressibility less 19.4 kW subtractive losses, giving 1761.1 kW nett overload output to the synchronous drive axial motor.

The system boost pressure (50 psi max) is not subtractive from the overload output, but must be subtracted from the theoretical mean power (1811kW-1.06%) to equal the wave input power (1792) -27kW per gyro pair as it is operative in a camming-motor mode in the same sense as the duck nod torque.

The torque of the 8 ring cam pumps in a Duck at 3250 psi equates to a sea-state torque limit of 0.5 MNm/m over the 45 metres effective length of each duck.

The pair of ring cam pumps on each gyro delivers to the pair of axial motors driving the gyro and in addition supplies one-half the requirement of the separately mounted axial motor driving the synchronous generator, but only at peak wave power. As the wave approaches the null power crest or trough, the hydro computer built into each gyro motor reduces stroke displacement of the gyro-motors so as to maintain a precise pressure directly related to the then obtaining gyro speed. As the wave input reduces to one-fifth of maximum, the gyro stroke displacement go over-centre into the pumping mode, so that when the wave power subsides to null, each gyro-motor will have reached a limit of one-quarter stroke-displacement in pumping mode, drawing on energy stored in the gyro, before the stroke cycles back across dead-centre up to a maximum motoring stroke displacement, the whole cycle occupying ca. 5 seconds or half the wave period.

The gyro-motors have automatic port-timing control which ensures that they recover the compressibility energy in the ring-cam pump delivery except for a small loss when passing over centre from motoring to pumping. The generator drive motor is fully corrected over its entire operating range.

The ring-cam pumping losses at max. front end (overload) input of 90 kW/m, amount to 19.01 kW per gyro.

GYRO-LOSSES @ 1500 RPM @ 3250 psi SYSTEM

Gyro load following bearing: leakage 2350 psi 86% effic.	=	3.36	kW
Gyro load rate 4 pockets: leakage 2350 psi 86% effic.	=	1.23	kW
Gyro transverse load 4 pockets: leakage 1300 psi 70% effic.	=	0.555	kW
		5.145	kW
Gyro load, rate and transverse journal: shear 1500 RPM	=	9.51	kW
Gyro thrust (one-way) bearing: leakage 600 psi 49.5% effic.	=	0.52	kW
Gyro thrust (one-way) bearing: shear 1500 RPM	=	1.96	kW
Total losses per one-end bearings	=	17.14	kW
Total bearing losses per Gyro	=	34.28	kW
Gyro-drive Axial Pump/Motors (2) cycling (1 thru - 1/4)	=	32.55	kW
Gyro-drive Axial Pump/Motors (2) compressibility remanence	=	3.96	kW
Gyro windage and centrifugal filter recovery	=	1.28	kW
On-Gyro eductor losses included in above	=	-	kW
Gyro-cage precession swivel bearings (2) losses	=	0.68	kW
Gyro gimbal-cage support bearings (2) losses	=	0.21	kW
Gimbal-cage Eductor losses inclusive	=	-	kW
Ring cam pumps (2) losses inclusive (see below)	=	19.01	kW
Total inclusive losses per Gyro at max. power	=	91.97	kW
Percentage loss on 2025 kW input	=	4.54	%

RING-CAM PUMP LOSSES PER GYRO AT 3250 psi

Piston/cylinder leakage (128)	=	1.78	kW
Piston/cylinder shear	=	0.8	kW
Spherical hydrostatic big end leakage (128)	=	3.86	kW
Spherical hydrostatic big end shear	=	0.0	kW
Spherical little end leakage and shear	=	0.0	kW
Cam-follower roller leakage (128)	=	3.74	kW
Cam-follower roller shear	=	2.6	kW
Cam-follower lever pivot leakage (128)	=	3.125	kW
Cam-follower lever pivot shear	=	0.0044	kW
Cam/roller losses	=	3.1	kW
Total Ring-Cams per Gyro	=	19.01	kW

SUMMARY

2 Gyro assemblages @ 91.97 kW	=	183.94	kW
Generator drive motor	=	26.95	kW
Pipe losses	=	1.12	kW
		212.01	kW
Front end input - ½ duck (max.)	=	2025.00	kW
Conversion efficiency (excl. electrics)	=	89.53	%

(iii) Ring Cam Pumps

The ring cam pumps are the prime generators of high pressure in the hydraulic fluid. The cylinders are based on those of the Swashplate pump/motor unit. Each cylinder is provided with a pair of poppet type valves, the inlet one being fitted with a magnetically assisted operating coil which is energised to "enable" the particular cylinder to pump. This makes for a 'fail safe' arrangement. (Drawing Nos. 10060 and 10061 refer). Each pumping cylinder is operated by a generously proportioned roller through a 2:1 velocity ratio linkage. This roller runs on a large ring cam of a type similar to that used by McTaggart Scott of Edinburgh, on motors supplied to the Royal Navy amongst others, whose advice has been sought throughout the design.

(iv) Eductor Pumps

In the interests of eliminating all possible moving parts, the eductor method is used to establish the various pressures required throughout the system for satisfactory operation. A paper "An Infinite Life Multi-Output Ancillary Pump" is appended hereto.

4.5.3. (iv) Contd../

AN INFINITE LIFE MULTI-OUTPUT ANCILLARY PUMP

SUMMARY

The requirement for an ancillary pumping system, providing multiple pressure outputs plus sump scavenging in a vacuum environment, and having "infinite" life, absolute reliability and zero maintenance, could only be met by utilising the eductor or jet pump principle.

Adoption of a multi-stage feed-back self-enveloping design, backed by an easily understood mathematical design method, meets all requirements with the added advantage of minimal bulk and low manufacturing costs.

I N D E X

LIST OF ILLUSTRATIONS

1 PREFACE

The attempt to provide 25 year maintenance-free reliability of the gyro-hydraulic power conversion system adopted for harnessing wave energy resources by means of nodding ducks, affects not only the high-pressure hydraulic pumping equipments, but makes equally high demands on all ancillaries and subsystems.

This is especially true of the ancillary pressure supplies for bearings, hydraulic controls, circuit loss make-up and boost, and for sump scavenging in the vacuum environment. This note describes the solution adopted after an assessment of every possible alternative configuration.

2 "SOLID STATE" EDUCTOR PUMPING SYSTEMS

Where ancillary pumping systems are handling only small amounts of power efficiency becomes less accountable than life, reliability and maintenance.

Eductor pumps, the principle of which should be basically familiar to most of us from our early Chemistry lab experience of tap-water powered evacuators, have no moving parts whatsoever and are virtually unchokable, hence their use as sand and slurry pumps and for handling bilge and sewage.

If to these characteristics we can add minimum bulk, a single hydraulic power input, multiple outputs and a scavenging intake, and unmachined die-cast manufacture, who is going to argue about a few percentage points of overall system efficiency.

2.1 Principle of the Eductor or Jet Pump

The term eductor or jet pump* describes a pump having no moving parts and utilising fluids in motion under controlled conditions. Specifically, motive power is provided by a high pressure stream of fluid directed through

* A. M. JUMPETER: Jet Pumps: Pump Handbook: McGraw-Hill

a nozzle designed to produce the highest possible velocity. The resultant jet of high-velocity fluid creates a low-pressure area surrounding it causing the secondary or suction fluid to be induced and accelerated in a convergent tract to join with and be entrained by the jet before more intimate mixing in the throat tract.

Ideally there is an exchange of momentum in the process producing a uniformly mixed stream travelling at a high velocity intermediate to the motive and induced flow velocities. A divergent diffuser tract is shaped to reduce the throat velocity gradually and convert the momentum energy back into pressure at the discharge end with as little loss as possible.

The four basic parts of an eductor are the nozzle, the inducer/entrainment tract, the throat, and the diffuser divergence (see Fig 1).

If two eductorrs are staged in series, efficiency can be improved substantially by transferring the suction intake to the second stage inducer and feeding the first stage inducer by a measure of recirculation from the second stage outlet. However, the recirculation connection should be as short as possible, with as large a bore or section area as possible to minimise the flow transference losses.

MULTI-STAGING EDUCTOR PUMPS

Instead of operating in parallel to provide multiple output services, eductors can be staged in series. But this becomes unwieldly with lots of inter-connections and worsening of efficiencies due to the multitude of long flow paths.

By adopting a "coaxial" envelopment configuration in which each pressure stage is enclosed by the next lower pressure stage which provides its recirculatory "induced" flow, as well as a service outlet connection and the nozzle input powering the encircling next lower stage, the outermost stage induces directly from the sump in which the entire assembly is immersed so that the depth of head is added to the small residual vacuum to ensure induction scavenging of the sump fluid in a vacuum environment.

The efficiency of eductors is governed by the flow mixing losses engendered by combining the high velocity nozzle and low velocity induced flows, together with the boundary layer losses which increase exponentially with velocity. Provided the deflection angles of the divergent pressure recovery sections are not great enough to cause cavitatious eddying, the losses in this section will be restricted to the boundary losses.

One advantage of designing a multi-stage eductor as a "coaxial" envelopment is that the early stages which usually have high mixing velocities can operate with minimum diameters and therefore lower boundary flow losses, whereas the later stages with lower velocities can benefit in their flow mixing by the diametrically extended flow surfaces which can be drastically shortened to reduce the boundary flow losses.

4 A POSITIVE DESIGN METHOD FOR MULTI-STAGE EDUCTORS

If we accept that the "coaxial" arrangement with successively reversed flows in short tracts, can provide an optimum design basis with negligible boundry flow losses other than those which are a part of the mixing and pressure recovery functions, it is possible to proceed with the various aspects of design and performance assuming that a detailed specification of the desired outputs, both in terms of pressure and delivery appropriate to system ancillary services has been drawn up.

4.1 Flow Diagram

A diagramatic representation of three-stage eductor pump (Fig 2) shows the specified pressures (P suffix) and outputs (O suffix) and the unknown inputs (N suffix and recirculatory flows (R suffix).

2 Calculation Areas

The calculations will be divided into six areas,

(1) Fluid velocities

(2) Flow rates

(3) Induced flow convergence acceleration

(4) Diffuser divergence decelerations

(5) Non-contributory boundary flow losses and instabilities

(6) Efficiency/life/cost evaluation

In these calculations, fluid pressures (P suffix) are lbs/sq ins; flows (O suffix, N suffix and R suffix) are cu in/sec; velocities (V suffix) are ft/sec; gravity acceleration (g) and other accelerations (A suffix) are ft/sec^2; fluid density (w) is 56lb/cu ft for hydraulic oil, viscosity is neglected as being effectively irrelevant for hydraulic oils.

2.1 Fluid flow velocities

Although referred to as "flow velocities", these are in no way related to flow rates, being entirely a function of combinations of pressure influenced accelerations of the fluid with density and gravity corrections which for our purposes are assumed constant.

Thus: $$V\ nozzle = \sqrt{\frac{288g(Pn - Pr)}{w}} = 12.87\sqrt{Pn-Pr}$$

$$V\ throat = \sqrt{\frac{288g(Po - Pr)}{w}} = 12.87\sqrt{Pn-Pr}$$

But Pr is recirculated from next lower stage and Po is identical with nozzle pressure of following stage therefore

$$\text{for the first stage} \quad Vn1 = 12.87\sqrt{Pn - P2}$$

$$\text{and} \quad Vt1 = 12.87\sqrt{P1 - P2}$$

$$\text{for the second stage} \quad Vn2 = 12.87\sqrt{P1 - P3}$$

$$\text{and} \quad Vt2 = 12.87\sqrt{P2 - P3}$$

$$\text{for the third stage} \quad Vn3 = 12.87\sqrt{P2 - P4}$$

$$\text{and} \quad Vt3 - 12.87\sqrt{P3 - P4}$$

where Pn is the pressure from the primary power source and P4 is the low pressure infeed or scavenging head: also Vnx is the stage nozzle velocity (ft/sec) and Vtx is the stage throat velocity (ft/sec).

The number of stages need not be limited to the three stages of the example which is indicative of the applications for which it appears to offer the only solution.

4.2.2 Fluid flow rates

As the stage output flow rates (O_1, O_2, O_3) at pressures P_1, P_2, P_3 are specified, it is only necessary to ascertain the inputs, N_1 from the primary hydraulic power source with a pressure Pn, and N4 from the low pressure infeed or sump scavenge at pressure P_4, before determining the recirculatory and diffuser flows.

THE FIRST STAGE RECIRCULATORY INFLOW $(R_1) = \left(\frac{V_{n_1}}{V_{t_1}} - 1\right)N_1$

$$= N_1\left(\sqrt{\frac{P_n - P_2}{P_1 - P_2}} - 1\right)$$

FIRST STAGE THROAT THROUGH FLOW $(T_1) = N_1 + R_1$.

$$= N_1\sqrt{\frac{P_n - P_2}{P_1 - P_2}}$$

ND STAGE RECIRCULATORY INFLOW $(R_2) = N_2\left(\frac{V_{n2}}{V_{t2}} - 1\right)$

$$= N_2\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right)$$

$$= \left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right)$$

D STAGE THROAT THROUGHFLOW $(T_2) = \left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}}$

D STAGE NOZZLE FLOW $(N_3) = T_2 - O_2 - R_1$

$$= \left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}} - O_2 - N_1\left(\frac{P_n-P_2}{P_1-P_2} - 1\right)$$

D STAGE INDUCED INFLOW $(R_3) = N_3\left(\frac{V_{n3}}{V_{t3}} - 1\right)$

$$= N_3\left(\sqrt{\frac{P_2-P_4}{P_3-P_4}} - 1\right)$$

$$= \left\{\left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}} - N_1\left(\sqrt{\frac{P_n-P_2}{P_1-P_2}} - 1\right) - O_2\right\}\left(\sqrt{\frac{P_2-P_4}{P_3-P_4}} - 1\right)$$

D STAGE THROUGHFLOW $(T_3) = \left\{\left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}} - N_1\left(\sqrt{\frac{P_n-P_2}{P_1-P_2}} - 1\right) - O_2\right\}\sqrt{\frac{P_2-P_4}{P_3-P_4}}$

STAGE RECIRCULATORY $(R_2) = T_3 - O_3$

$$= \left\{\left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}} - N_1\left(\sqrt{\frac{P_n-P_2}{P_1-P_2}} - 1\right) - O_2\right\}\sqrt{\frac{P_2-P_4}{P_3-P_4}} - O_3$$

$$\therefore R_2 = \left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right) = \left\{\left(N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}} - O_1\right)\sqrt{\frac{P_1-P_3}{P_2-P_3}} - N_1\left(\sqrt{\frac{P_n-P_2}{P_1-P_2}} - 1\right) - O_2\right\}\sqrt{\frac{P_2-P_4}{P_3-P_4}} - O_3$$

$$\therefore N_1\sqrt{\frac{P_n-P_2}{P_1-P_2}}\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right) - O_1\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right) = N_1\left\{\sqrt{\frac{P_n-P_2}{P_1-P_2}}\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}} - 1\right) + 1\right\}\sqrt{\frac{P_2-P_4}{P_3-P_4}}$$

$$- \left(O_1\sqrt{\frac{P_1-P_3}{P_2-P_3}} + O_2\right)\sqrt{\frac{P_2-P_4}{P_3-P_4}} - O_3$$

$$\therefore N_1 = \frac{O_1\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}}\sqrt{\frac{P_2-P_4}{P_3-P_4}} - \sqrt{\frac{P_1-P_3}{P_2-P_3}} + 1\right) + O_2\sqrt{\frac{P_2-P_4}{P_3-P_4}} + O_3}{\sqrt{\frac{P_n-P_2}{P_1-P_2}}\left(\sqrt{\frac{P_1-P_3}{P_2-P_3}}\sqrt{\frac{P_2-P_4}{P_3-P_4}} - \sqrt{\frac{P_1-P_3}{P_2-P_3}} - \sqrt{\frac{P_2-P_4}{P_3-P_4}} + 1\right) + \sqrt{\frac{P_2-P_4}{P_3-P_4}}}$$

4.2.3 Convergence accelerations

Having deduced all the flow rates, these have to have each flow accelerated to a specific velocity, the nozzle or injected flows being less critical than the recirculatory or induced flows.

Nozzle areas, whether jet bore or annular, are obtained from the nozzle flow rate (cu in/sec) divided by the nozzle velocity (ins/sec), and convergence can be quite blunt. However, the nozzle length is more often determined by the demands of the surrounding induced flow and by the designers natural tendency to balance up the overall lengths to optimise production.

The acceleration of the induced flow is largely dependent upon the forcing pressure of the stage output from which it is derived. Where this forcing pressure is high the flow acceleration can be high and the convergence accomplished over a short tract length. However, if the induced flow to the last stage is from a very low pressure source or worse still, from a sump subject to a near absolute vacuum ambience, the induced flow forcing head is almost entirely determined by the depth of sump fluid above the very carefully designed eductor intake.

The tract convergence then becomes critical with only a small part of the flow acceleration taking place in the induction tract, and most of the work being done by the nozzle flow velocity energy transfer. This is assisted by the large diameter annular nozzle projecting a jet sheet at an angle outward from the true axis such that the sine of the angle times nozzle flow rate times nozzle velocity squared approximately equals the sine of the inflow angle times the induced flow times the square of the terminal convergence velocity before meeting the nozzle flow.

4.2.4 Divergence decelerations

The large diameters of the lower stages allow of minimal length parallel mixing tracts sized for convenience rather than for precise calculated lengths. However, the divergent diffuser tracts are critical in relation to the initial flow breakaway angle which can be later increased as the expanding flow reduces in velocity, until the ratio of the tract section area at the limit of divergence to that at the commencement of divergence is equal to the inverse velocity ratio, terminating at less than 15ft/sec corresponding to the stage offtake pipe flow velocity.

Theoretically a case can be made for an exponentially increasing divergence flare from the parallel throat to the limit of divergence, but for all practical purposes a two segment divergence is not measurably less efficient, particularly if the breakaway angle junctions are "softened" after manufacture and inspection.

4.3 An Example of 3-Stage Eductor Design

Fig 3 shows an enveloping 3-stage eductor pump providing three outputs of: - 1st stage 10 c.i.p.s. at 1600 p.s.i. for gyro cage bearings; 2nd stage 5 c.i.p.s. at 400 p.s.i. for controls; 3rd stage 110.5 c.i.p.s. at 60 p.s.i. for circuit boost, heat exchange and signalling.

The systems pressure powering the eductor 1st stage nozzle is 3150 p.s.i., and the sump scavenge inflow has only 14" head of fluid (½p.s.i.) to force induction and convergence acceleration from the sump vacuum environment.

By substitution in the equations of 7.2.2, it will be found that the primary input into the 1st stage nozzle will be 31.57 c.i.p.s. and the nozzle velocity 675 ft/sec: which is more than twice the 3rd stage nozzle velocity of 258 ft/sec which must mix energy with a scavenge flow of 94 c.i.p.s. having a terminal convergence velocity of only 8.5ft/sec when it meets the nozzle flow.

As this scavenge inflow terminal momentum is several orders of magnitude less than the smallest worthwhile nozzle deflection momentum, the outward deflection angle of the nozzle sheet will be limited to the boundary flow breakaway angle corresponding to the nozzle velocity.

The overall axial length of the enveloping eductor is determined by the 1st and 3rd stages, and the cost in terms of efficiency loss in compromising the 2nd stage nozzle/inducer length is negligible.

The efficiency of the first stage is 72%, of the 2nd stage 37%, and of the 3rd stage 20%: The overall powered pumping efficiency is 25%, with the extra advantage of scavenging 94 c.i.p.s. from the near vacuum sump.

It is important that there should be no possible risk of any orifice becoming clogged. The central nozzle is .071" diam. The second stage annular clearance is .009" over a 3" peripheral length. The third stage clearance is .040" over 5½" periphery. These dimentions provide a generous safety margin.

5 ONE OR A MILLION - EASE OF PRODUCTION

With the design made easy, and the main body and enveloped eductors simple lathe-work, "one-offs" are no problem except for the end-cover offtakes.

At the other production extreme, all the eductor elements are plain die-castings with the exception of the end-cover offtakes which in any case are particular to application. Assembly also is straightforward and lends itself to automatic handling.

The materials of manufacture will almost certainly be diecast Aluminium alloys, (excepting the high pressure input nozzle and the three tie-bolts). They will be polished and anodised at all the internal tract surfaces to minimise boundary losses, and they may most conveniently be centred one within the other by light alloy rivet heads in triplicate. The unidirectional thrust on the two inner concentric elements can be located on small triplicated castellations at the counterthrust end.

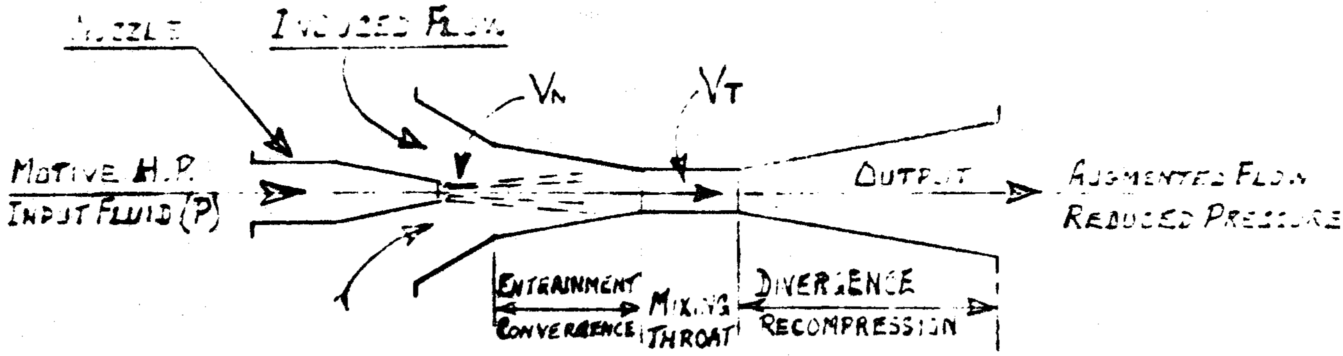
6 CONCLUSIONS

It has been possible to design to a proven formula a recirculatory multi-stage eductor ancillary pumping system powered from the primary pressure source and providing three intermediate pressure services together with sump scavenging from near absolute vacuum.

Calculation of the flow rates, accelerations and velocities has been reduced to simple formulae, as have the recirculatory indices and overall efficiencies which include not only the pumping efficiencies but also the driving efficiencies and the gratuitous scavenging of the vacuous sump.

There is no requirement for maintenance, reliability is absolute, and life infinite.

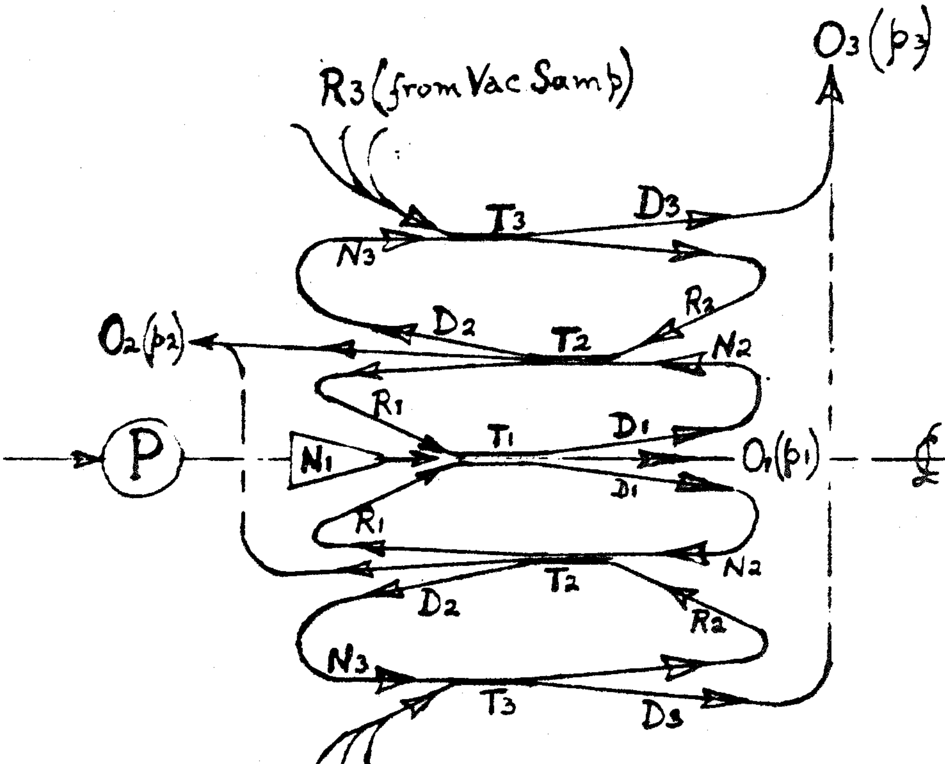
- o O o -



V_N = Nozzle Velocity $\quad\quad$ V_T = Throat Velocity

Motive and Output Flow Velocities are Negligible (ca 15 ft/s)

Fig 1. Eductor Pumping Principle



P = motive pressure
p = output pressure
N = nozzles
R = recycle/induction
T = mixing throat
D = diffuser divergen[ce]
O = output delivery
℄ axis of symmetry

suffix = stage numb[er]

NB Stages 2 & 3: nozzles are annular

Fig 2 Multi-Stage Eductor Flow Diagram

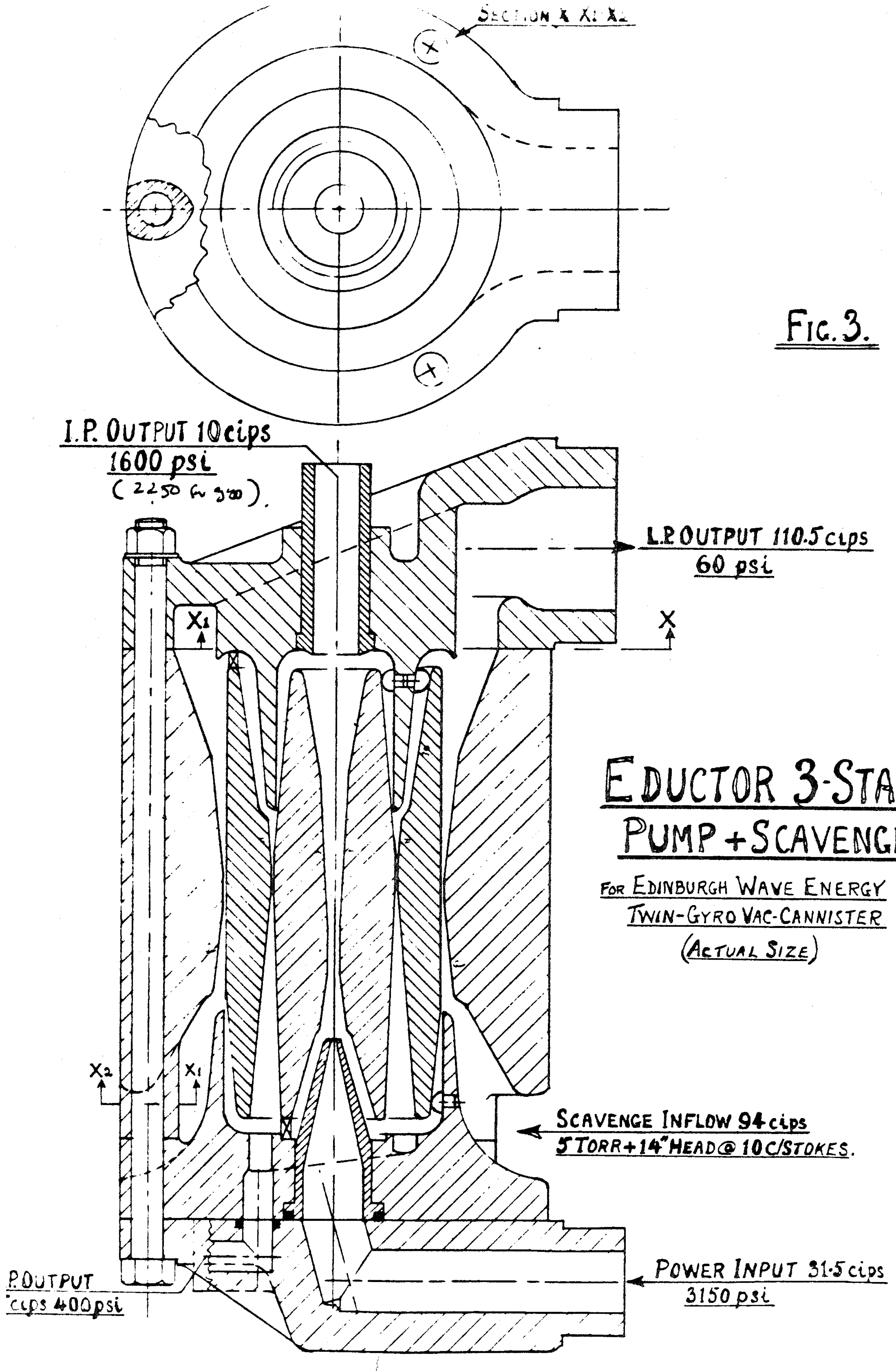


FIG. 3.

EDUCTOR 3-STAGE PUMP + SCAVENGE.

FOR EDINBURGH WAVE ENERGY TWIN-GYRO VAC-CANNISTER

(ACTUAL SIZE)

4.5.4 Control Mechanisms

The whole field of Control Engineering has rapidly advancing techniques developing with new devices appearing almost daily. A satisfactory method of control could obviously be devised, interfacing electronics, hydraulics and mechanical interfaces using today's techniques. The team feels, however, that detail design of such aspects of the duck power conversion should not be undertaken too early.

4.5.5 Condition Monitoring

This comes under two sub-headings:-

a) the sea condition, and
b) the condition of individual components.

For a), the duck will be fitted with suitable accelerometers with electronic analogue or digital output fed to a mini-computer so that all the necessary changes to operation are automatic. The condition of each of the vulnerable components in the power chain will be continually monitored with automatic removal from service if faulty.

For b), essential information will be fed to an on-board computer with a suitable, probably duplicated, link ashore. This link could be fibre-optics or radio; the former being preferred as extra cores in the power cables to minimise interference. Multiplexing of signals would be employed to keep the number of such links down.

The computer will be interrogated by the shore station as a matter of routine, with a priority-override on any catastrophic changes in condition. Such techniques are in use today. Tomorrow's equipment will be very reliable.

4.5.6 Installation

The spine and the duck are to be constructed separately and mated in shallow water. The string will be built up into a 1 Km length for tow out to the installation site.

The string will be joined together by the use of 3 Hydraulic Latches equally spaced around the circumference of the mating flange. Each section will have the male latches at one end and female latches at the other end, so that if during the life of the device it is required to remove one section, the latches can be hydraulically released, the spine driven apart, and the section towed out for repair or maintenance as required.

All the sections are to be towed out complete with individual moorings stowed, ready to be deployed at the installation site.

4.5.7 Inshore Assembly Procedure

(i) The spine and the duck are to be towed to a sheltered shallow water site.

(ii) The spine and the duck are to be positioned by the tugs so that they are in the correct orientation for the mate.

(iii) A diver will be deployed from the Dive support Vessel and take the line that has been previously attached to the duck retaining strap, and connect it to a line from the winch temporarily mounted on the duck. When the two lines have been joined the winch will haul in the line pulling the retaining strap with it.

(iv) When the strap is in position it will be secured on a duck and the winch will be removed for use on another duck and spine.

(v) The end flange of each segment will be precast with three equally spaced recesses, each able to accept a 36" dia. hydraulically actuated latching mechanism. On the flanges that house the female half of the hydraulic latches, the flange will be equipped with a precast wear tube to accommodate the pull-in line for assembly and installation. Each segment will also be equipped with a pull-in winch. The winch will be located directly beneath an access hatch so that it can be removed and re-used in another section.

(vii) The flange with the female half of the latch will have a pull-in line, pre-threaded through its wear tube.

The flange with the male half of the latch will be equipped with a short pull-in line.

A diver will be deployed from the Dive Support Vessel to joint the two pull-in lines together.

(vii) The pull-in lines will be put under tension as the two sections come together and the hydraulic latches dock. The latch hydraulics will be actuated from a remote station on top of the spine.

(viii) A facility is to be provided to dewater the void space between the spines prior to opening the watertight bulkheads.

(ix) When all 10 segments have been joined to form a 1 Km string the tow will commence.

4.5.8 Towing Procedure

Vessels/towing-gear/survey equipment will consist of:-

1 deep sea tug 22000 IHP 176 ton bollard pull.

1 assisting deep sea tug 8000 IHP 96 ton bollard pull.

1 escort/survey vessel

All three vessels will be equipped with a navigation package for positioning the ducks.

Towing gear to be carried on the ocean going tugs:-

1 towing lug 500t breaking strength
1 x 60m long pennant 500t breaking load 3" dia.
3 100 t shackles
1 double towing nylon 60 - 100m long 16" circumference.

The tugs will carry a complete back-up in event of an emergency.

Other equipment will consist of:-

1 air tugger winch to handle pennant wires
1 pick-up line for transfer of emergency pennant from the tow
1 x 1" dia. wire for the recovery of broken lines from the deck to the outboard end of the pennant.
Navigation lights to comply with maritime regulations

(i) The string will be equipped with a suitable towing or pushing attachment. The position of the attachment will be determined by model testing.

(ii) The draft of the tow will be adjusted to suit the requirements of the towing contractors.

(iii) A complete survey of the mooring location and the towing route is to be carried out prior to the commencement of the tow out of the first string of ducks.

(iv) From the calculation of the drag on the duck and spine assembly, the maximum towing speed has been estimated at 2 - 3 Knots with a 100t bollard pull.

On arrival at the installation site the towing configuration will be as follows:-

- the main tug on the main tow line
- the escort vessel and assisting tug on the stern pennant lines.

(v) The first string of ducks to be towed out will be moored.

(vi) A surface swimmer will be deployed to connect the pre-threaded line from the female flange to the line from the male flange.

(vii) Hydraulic hoses will be connected to the remote hydraulic panel on the spine.

(viii) The pull-in lines will be put under tension and the two segments will be pulled together.

The assisting tug will remain on the stern line to control the angle of the pull-in.

As the hydraulic latches mate the hydraulics will be activated thus locking the latching mechanisms.

(ix) The assisting tug will disconnect the stern line.

The void space between the segments will be dewatered before opening the watertight bulkheads to make the electrical connections.

4.6 Moorings

4.6.1 Rodes and End Connections

Because of the low and constant tensions in the rodes brought about by the system adopted for the duck, no special technology is required to ensure a long lasting installation. Parafil, or PVC coated steel wire rodes will be used with chain link end connections as recommended by British Ropes.

4.6.2 Sea Bed Anchorages

The loads permit the use of 'clump' anchors, weighing about 120 tonnes (downstream and 400 tonnes upstream). There will be two such anchors per duck, one upstream and one downstream. The use of such anchors is almost independent of the sea-bed conditions; however, where circumstances permit a modified Bruce anchor will be considered.

4.6.3 Other Details

The mooring system envisaged uses additional buoys and sinkers. These will be of conventional manufacture although the team are examining the possibility of using concrete as the main material of manufacture. Care will be taken with the attachment system to avoid excess bending in the rodes.

4.6.4 Installation of the Moorings

The moorings are to be stowed on the string of ducks prior to the tow out.

The moorings will consist of a 60 mm, PVC clad, grease packed wire rope, a clump weight, subsea buoy and sinker. A Parafil alternative will also be considered.

The moorings will be made up to the required lengths and be shackled to the mooring point on the spine. The clump weight will be lowered to the seabed by the crane on the Dive Support Vessel. An R.O.V. survey following deployment will be required.

SECTION 5

RELIABILITY AND LOSSES DUE TO BREAKDOWN

5. RELIABILITY AND LOSSES DUE TO BREAKDOWN

5.1 On Board Systems

5.1.1 Mechanical Components

All the mechanical components within the power canister have, probably, better working conditions than almost any other engineering installation. They operate in a sealed non-corrosive, non-oxidising, low pressure environment specifically to ensure long life. All bearings are to be of the hydrostatic type with the sole exception of the line contact between the cam follower rollers and the ring cam surface. Here the follower roller is generously proportioned so as to keep Hertzian stresses low enough to ensure less than 10% failure over 100 years. (About 1/6th of the stress used by British Rail.)

In the case of the ring cam pump it should be noted that the number of pumping cylinders provided is some 11% over the peak requirement. The control system works by "enabling" the cylinders; any damaged cylinder or mechanism being automatically left inoperable. It should also be noted that even the catastrophic failure of one complete power canister will only reduce the output of the station by 0.1% of rated power - a figure easily made up by a small increase in output of healthy units.

The team has rejected pumps with moving parts where possible and are proposing eductor pumps for establishing the various flows and pressures required (see 4.5.3 (iv) page 149, and drawing numbers 10080 and 10081).

Failure of a section of the compliant spine joint (e.g. the failure of a hydraulic ram) will allow the joint to swing freely. Since both sides of each joint are separately moored, only a loss of efficiency will be suffered by the affected ducks. The unlikely failure of a Hookes joint would be more serious in that it would allow the spine to actually part, whereupon heavy seas could increase damage by battering the two halves together. This can largely be avoided by splaying the end moorings, thus applying a slight tension to the string. Wide tank testing continues to investigate this aspect.

5.1.2 Electrical Components

The electrical AC generators have been carefully selected with generously proportioned hydrodynamic bearings.

The expected life of transformers and switchgear, both of which have also been carefully selected, is indicated in table 2 of paragraph 6.3. A maintenance period of 10 years is indicated in all cases but the teams are confident that this will be improved upon.

Transmission cables are included in the table.

5.1.3 Electronic Controls

The modern technology of solid state circuitry and switching will be fully exploited in the detail design of the control system. The reliability of such equipment is improving from year to year and the team is confident that all problems can be solved.

5.2. Other Plant

The on-shore plant, other than the maintenance facility, will consist of a control and data logging station and a transmission system. The latter is dealt with under 6.3 table 2. The former will comprise a data logger with perhaps 100,000 channels giving 100 pieces of information per duck. The team anticipate using fibre-optic links built into the undersea cables backed up by radio link. All the system will make use of the most modern and reliable equipment available at the time of design.

5.3 Failure Rates

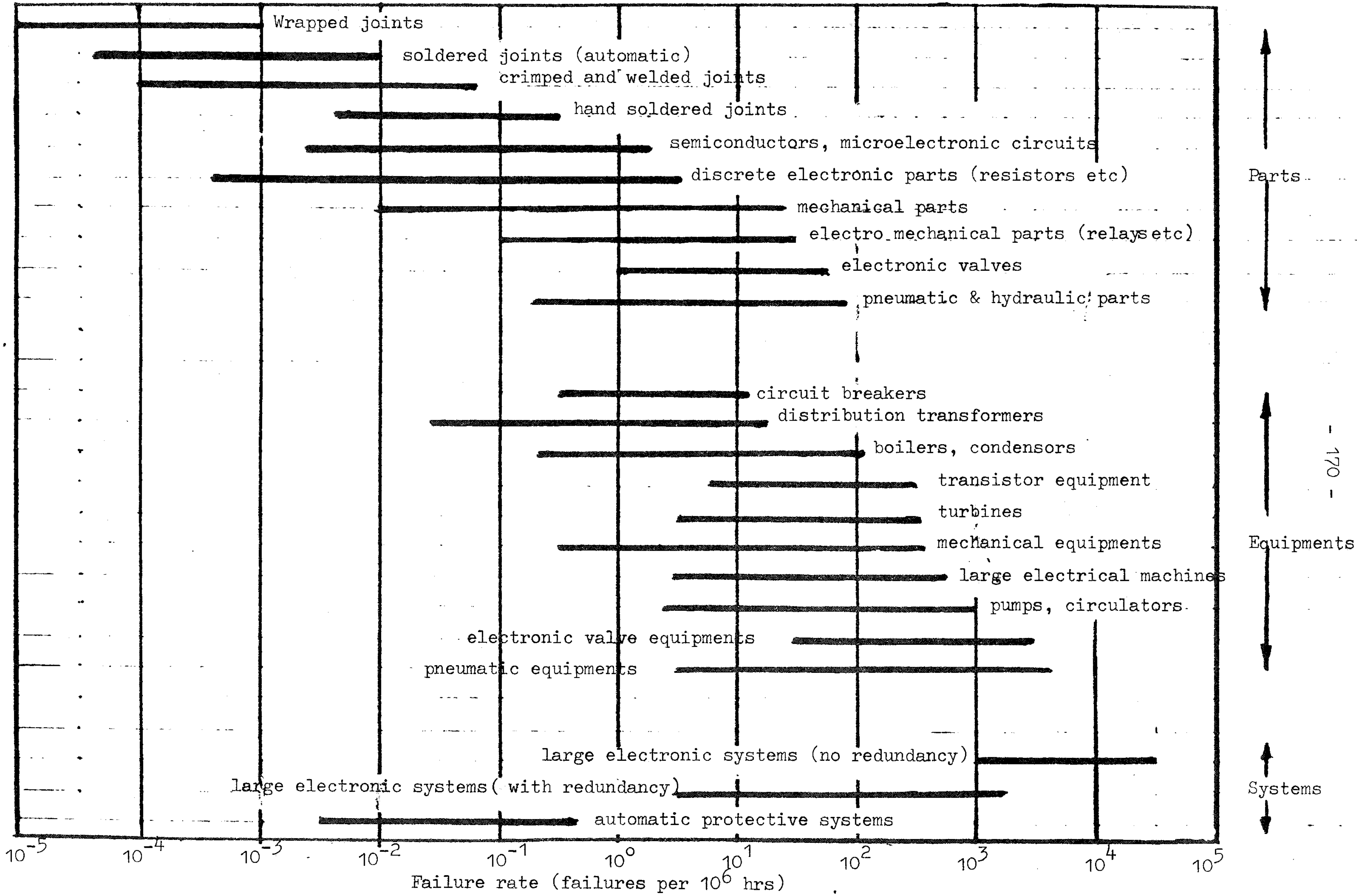
A chart "Typical ranges of failure rates for parts, equipments and systems" appears herein. This is an extract from "Reliability Technology" by Green and Bourne of the UKAEA to whom acknowledgements are due.

Bearing in mind the striving towards long life that is stressed throughout, the team feel that one can assume failure rates within the best third of the indications of the chart.

This means that for the applicable items, a failure rate of 10^{1} per 10^{6} hours of operating life or about 2 failures per 25 years for most of the components. The maintenance systems referred to under section 6 should be well able to cope with such a failure rate.

The Team is investigating examples of long, unattended operation in other fields and can so far point to the Domestic Refrigerator and Central Heating Pumps at one end of the scale - through such every day units as London Transport bogie bearings (quoting only 6 failures/year in 30,000 units after up to 35 years operation) to 3.5 MW gas compressors; some 1400 of which are in continuous use in the Eurodif enrichment plant at Tricastin, where 20 years unattended operation has been achieved (with acknowledgements to Nuclear Engineering). The Team's investigations continue.

Reproduced from "Reliability Technology by Green & Bourne of Ukaea with acknowledgements



Monitor

Enrichment plant compressors for 20-years' non-stop operation

By 1981 the French consortium Gercos will have supplied a total of 1400 motor compressors with a total rating of 3300 MW to the Eurodif enrichment plant at Tricastin. Precision mass production and standardisation are essential in meeting the quality objectives which are needed for a lifetime of continuous operation.

The Fr 15 000 million Eurodif gaseous diffusion enrichment plant at Tricastin, the largest project of its kind in Europe, will alone account for a third of the world's enrichment capacity by the time it is completed in 1982, with a full capacity of 10 800 t s.w.u. Each stage of the gaseous diffusion process comprises a diffusion barrier, which separates U-235 from U-238 in uranium hexafluoride gas, and a motor/compressor unit. The 1400-stage cascade of the separation plant includes three sizes of motor compressor: 280 units rated at 600kW (3000 rev/min handling 30kg/s), 400 stages rated at 1500kW (3000 rev/min, 80kg/s), and 720 stages rated at 3500kW (190kg/s, 1500 rev/min).

To carry out the design and manufacture of this large quantity of specialised motor/compressors, which alone will almost account for the full output of the neighbouring Tricastin nuclear power station (with four 925MWe PWR units, the first of which is due for commissioning towards the end of this year) a group was formed comprising two principal constructors, Hispano-Suiza, part of the Snecma group, with a two-thirds share in the consortium, and Alsthom Atlantique. The new group, know as Gercos, included both the major compressor manufacturers for the CEA's Pierrelatte enrichment plant. Here, a total of 2000 mostly smaller compressors were supplied. The machines installed at the Pierrelatte plant were originally designed for routine inspection but in service the design proved to be suitable for continuous operation, and those installed first in 1965 have now passed 110 000 hours continous operation.

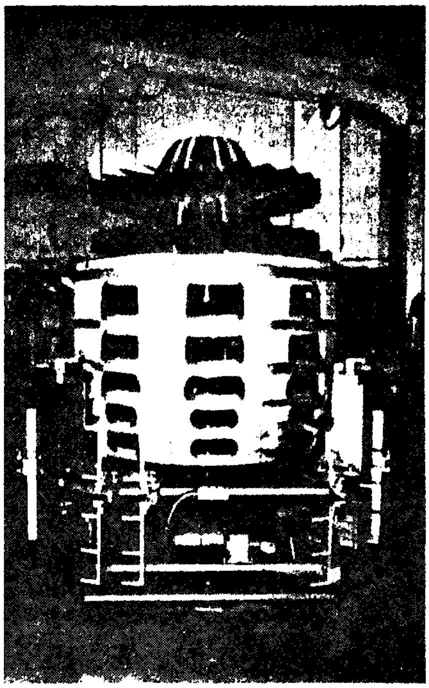
The contracts for the manfacture of the largest (1500kW) compressors was awarded jointly to Gercos and the Italian company Nuovo-Pignone working under a Gercos licence. All the machines of the smallest rating have now been delivered to site, and all but 30 of the medium sized compressors. Of the largest sizes, 80 Gercos and 25 Italian-made machines are now on site.

Clean room conditions

The compressors are made, and assembled under clean room conditions at the respective factories of the companies of the Gercos group, and the diffusion barrier is added by Eurodif at Tricastin. The main criteria of the design are the high reliability, for the prolonged non-stop operation (20 years without major overhaul), the absence of any foreign matter in the gas circuits, the resistance to the highly corrosive gases, the overall gas-tightness in view of the toxicity of UF_6, and overall efficiency of the machines.

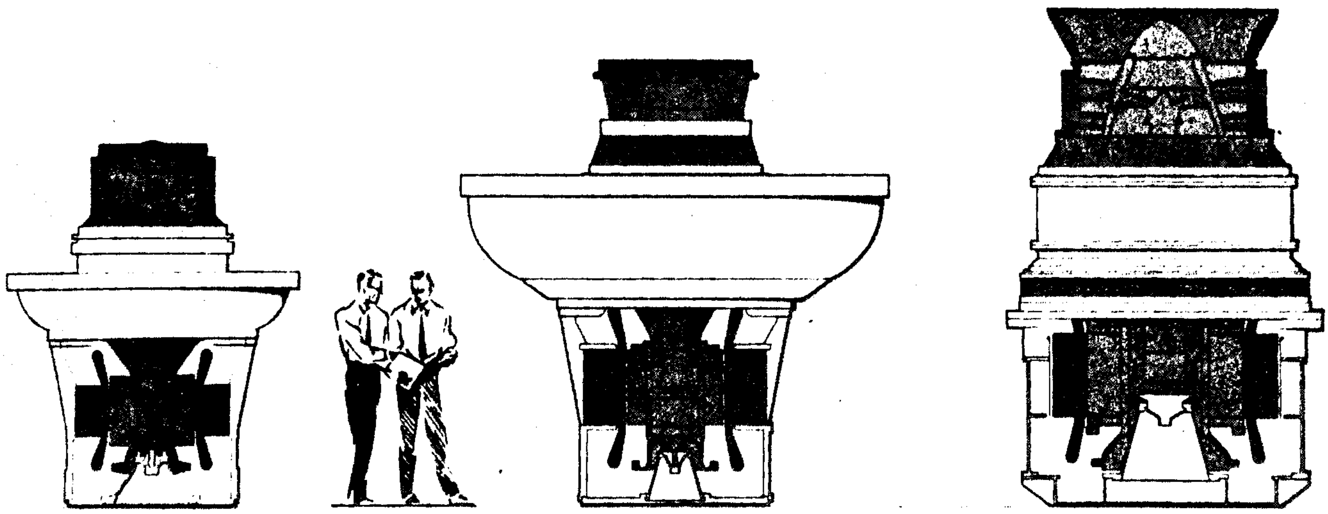
The efficiency of the plant is of enormous economic significance with such high total ratings: an increase of 1 per cent in the losses of the motor/compressors, for example, represents an additional annual running cost of Fr 20 million for the Tricastin plant.

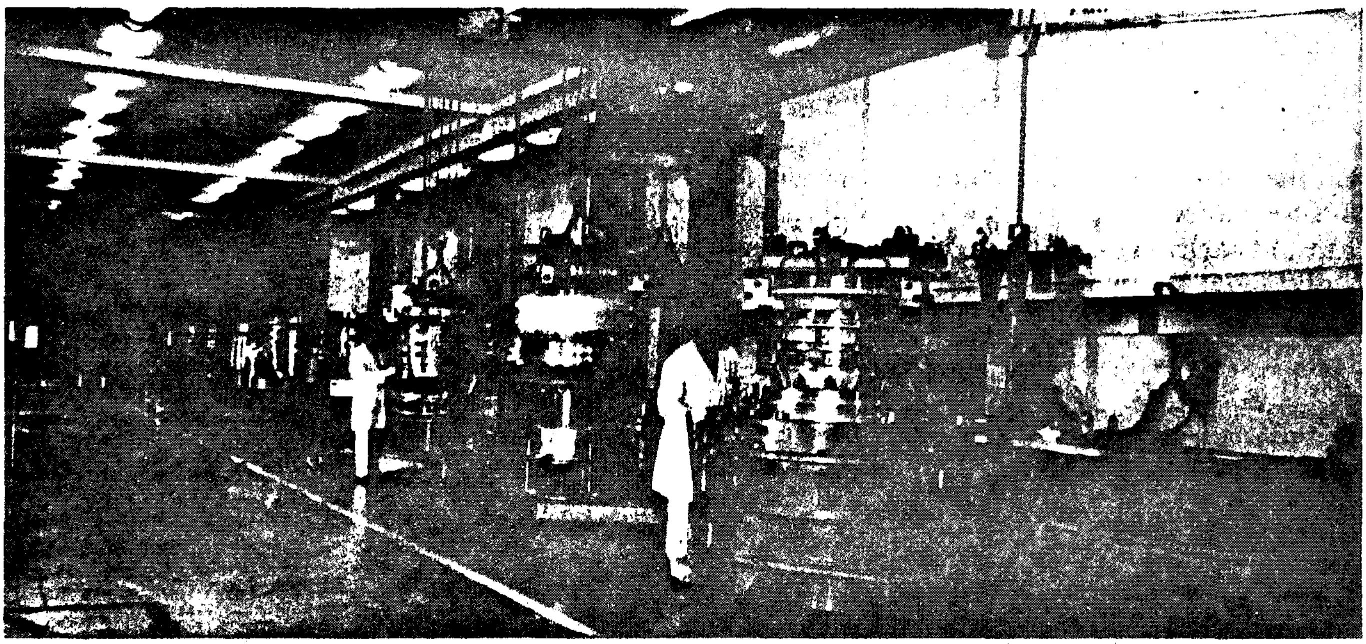
The multi-stage axial compressors are made at the Rateau factory of the Alsthom Atlantique at La Courneuve, and the Bois Colombés factory of Hispano-



The completed 3500kW compressor assembly ▷

Comparison of the three motor/compressor sizes: (left to right, 600kW, 1500kW and 3500kW) for the Tricastin gaseous diffusion plant ▽





The clean room assembly line at the Hispano-Suiza Bois-Colombes factory

Suiza, both to the north of Paris. To achieve the high quality design objectives demanded the establishment of a quality control procedure so extensive it accounts for 15 per cent of the total production cost. The quality assurance teams are responsible directly to the respective company presidents to ensure as independent a standpoint as possible within the organization.

The principal effect of the other major design criteria is that all larger components in the compressors, which are made from steel forgings, must be nickel plated to be resistant to corrosion by the uranium hexafluoride gas, and the assembly of the compressors must be carried out under strict clean-room conditions.

The compressors have two pressure flows: the low pressure intake at the main inlet to the compressor handles gas that has passed through the diffusion barrier of the previous stage. The recycled gas rejected at the previous barrier is at a higher pressure and is therefore brought in at the last, low-pressure mixing stage of the compressor. The two intakes have very similar mass flows, so that a simple adjustment of the recycled gas flow entering the last stage allows the mass flows to be accurately matched on each compressor.

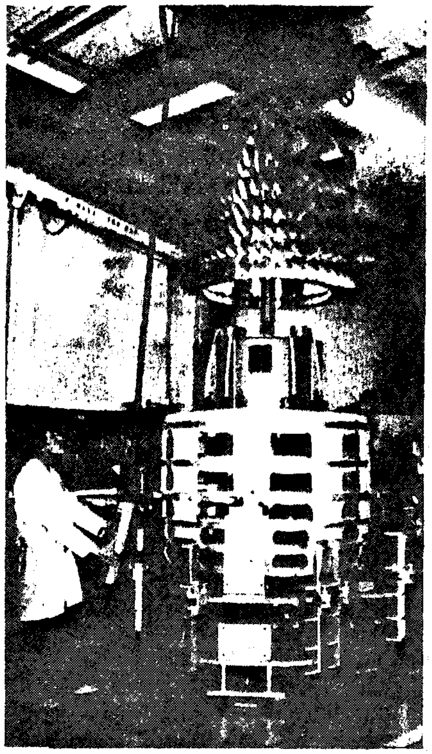
The compressor's rotor is mounted vertically, supported at a single suspension point, with a gas-tight seal specially developed by Hotchkiss-Brandt-Sogeme group. A remotely operated shutter is provided to isolate the compressor if necessary and allow the drive shaft to be disassembled without risk of gas leakage.

Main features of the motor/compressor for the Tricastin enrichment plant

		Small size	Medium size	Large size
Main stream flow	m^3/s	[illegible]	[illegible]	[illegible]
Speed (rev/min)		3,000	3,000	[illegible]
Electric motor power	kW	600	1,500	3,400
Max. rotor diameter	m	0.60	0.95	1.49
Overall height (compressor motor)	m	2.36	3.00	3.05
Total mass	kg	5,995	10,856	20,976

The production of these special compressors involved a number of major sub contractors, such as forgings by Creusot-Loire (France), Shafts by Terni (Italy) and Cockerill (Belgium), special heavy duty alloys by High Duty Alloys (Great Britain) and electric motors by Ansaldo/Marelli (Italy), CEM (France), and Brown Boveri (Italy and Spain). The total value of the motor/compressors contracts was about Fr 3 billion, which is above 20 per cent of the total estimated cost of the Eurodif enrichment plant at Tricastin.

Gercos now awaits a decision on the Coredif plant, which could be as large again as Tricastin. If this project is postponed beyond the completion of the Tricastin plant Gercos will remain, since there are no engineers or staff directly employed by the consortium. But since the two manufacturers in the group have a large investment in forging presses, numerically controlled machine tools and special assembly areas dedicated to this range of machines, together with the specialized team of constructors, and since the anticipated world enrichment capacity demands all of the new plant projected at present, Gercos argues strongly for the early continuation of the Coredif plant. Despite the high power demands of the gaseous diffusion process, there is a strong belief in the proven industrial-scale viability of the process, compared with centrifuge enrichment, by a wide section of French industry. ■



Mounting the rotor into the stator of the 3500kW compressor

S E C T I O N 6

I N S P E C T I O N & M A I N T E N A N C E

6. INSPECTION AND MAINTENANCE

6.1 Structures

The concrete structures of both the spine and the duck have been designed with long fatigue life and little or no maintenance in mind. Steps have been taken to eliminate the possibility of marine growth in vulnerable spots. Marine growth on the outer surfaces of the device is not expected to exceed 100 mm thickness of crustacea with some kelp attached. This is based on information from the Scottish Marine Biology Association. The design has allowed for such coverage.

6.2 Mechanical Equipment

A deliberate no-maintenance-at-sea policy has been adopted by the team as exemplified by the entire power conversion train being housed in a low pressure, non-oxidising, non-corroding environment. All components will be individually designed to embody this philosophy, such as the use of hydrostatic bearings throughout the system as far as possible. The method of installation (i.e. that of towing strings of ducks) will be repeated when maintenance is necessary. When a 5 day weather window appears likely (i.e. waves of less than 3 metres high) a reserve string will be towed from base and used to replace whichever length has the most significant faults. When the faulty length is returned to base those components external to the power canister will be refurbished or replaced. This string will inevitably contain some power canisters without faults. If the operating life has been less than 5 years, canisters will be re-used without doing any overhaul.

Any suspect canister will be stripped and inspected. Dependant upon condition, components will be replaced and the canister set up for a further period of duty. The life history of each and every component will be carefully monitored so that an efficient system of minimum cost can be operated. Power canisters will be salvaged in the event of a catastrophic failure, such as a ship collision, and returned to base for stripping, refurbishment and re-use.

6.3 Transmission and Electrical Equipment

A conscious effort has been made to ensure that the need for maintenance is avoided or at least minimised. Two examples of this are the higher cost for the generators and controlling their environment by nitrogen filling the generator compartment and the selection of 132 kV totally enclosed SF6 switchgear.

Table 2 (page 174) shows the assessment of the periods between inspection or maintenance for the main components of the arrangement. The maintenance system to be set up for mechanical components will allow for the electrical requirements in parallel.

6.3 TABLE 2

Item	Inspection or maintenance free period (years) Probable min	Probable max
Generators and associated equipment	10	15
Cables	25/circuit (faults)	100/circuit (faults)
Switchgear below 132 kV	10	20
132 kV SF6 switchgear	10	30
Transformers (and reactors)	20/transformer (faults)	40
Overhead lines	3/100 km/DC2 (faults/year)	0.3/100 km/DC2 (faults/year)
Monitoring*	10	25

* Based upon failure of all parallel or duplicated equipment

6.4 The removal of a Section of Spine for Maintenance or Repair

To remove a section of the spine the following procedure would have to be adopted.

(i) Prior to carrying out the above operations the section will have to be elctrically isolated and the watertight bulkheads sealed.

(ii) Recover the moorings.

Divers will be deployed from the Dive Support Vessel to attach lift lines to the clump weight.

(iii) The clump weight will be lifted using the crane on the D.S.V.

(iv) The assisting tug will attach a towing bridle to the section to be removed.

(v) The latch hydraulics will be connected up and activated to release the latches at each end of the section.

(vi) The spine will be "driven" apart. When the clearance between the sections of the spine is sufficient the tug will tow out the required section.

(vii) The main tug will attach a tow line to the section and tow it back to the shore.

6.5 Maintenance and Inspection of the String

Maintenance of the array will be determined by the findings of the Reliability Analysis being carried out on the device by various groups.

The information from the reliability studies will assist in building up a planned maintenance programme for the overall project.

It has been shown in the previous procedure that it is possible to remove a section of the spine and tow it back to shore for maintenance or repair.

The spine and the moorings will be surveyed annually by a Remotely Operated Vehicle. This survey will include a video inspection of the duck/spine joint, the flexible joint in the spine, and the moorings.

6.6 Maintenance Base and Fleet

This has, pro tem, been based on the Consultant's recommendation (see "Schedule of Standard Rates and Measurement for Standard or Special Rates"). The team feel, however, that bearing in mind the no-maintenance design philosophy and the on-board computing and monitoring system proposed, this requirement could be somewhat smaller than envisaged. This also applies to manning levels.

Further consideration will be given to both these aspects.

SECTION 7

PERFORMANCE & PRODUCTIVITY

Revised version 25th November 1981

7.0

7.1 Productivity Analysis of Salter Ducks

(1981 reference design)

Denis Mollison

(Heriot-Watt University)

Summary

The productivity of the 1981 reference design of the Salter duck is estimated at 18.3kW/m, and its unit cost at 2.75p/kWh.

This is for a 2GW scheme sited in 100m depth approximately 35km west of South Uist. The reference design is 14m diameter by 45m long, with a delivered power rating of 2.344MW (52.1kW/m). The reference scheme consists of 864 ducks, so that the precise overall rating is 2.025GW, and has estimated mean output 713MW. (These figures all refer to delivered power on Skye.)

The above figures do not allow for loss of output resulting from the breakdown of part or all of the scheme. A mean loss of 10% of the power units, and of the complete system for 5% of the time, leads to estimated net productivity of 16.9kW/m (656MW overall) at a unit cost of 2.99p/kWh.

Productivity has been calculated using the Energy Technology Support Unit's recommended set of 46 spectra, adjusted to a site of 100m depth using the Institute of Oceanographic Science's recommended comparison data.

Device performance is estimated from the device team's tank experiments with a model mounted on a suitably compliant rig in uni-directional seas. In estimating their performance in multi-directional seas a cosine law has been assumed. This is thought to be conservative: an assumption which is supported by initial wide tank experiments on a string of 6 ducks on a fixed axis (Oct. 1981).

The generation and transmission losses are calculated from figures available for the reference design at mid-October 1981. Estimated costs were provided by Project Assessment and Development, and Merz and McLellan; an allowance for maintenance was also included.

The interpretation of the above productivity and cost figures, and possibilities for improvements, are discussed.

1. Aims

1.1 To obtain data on: (a) front-end power extraction

(b) gyro and generation losses

(c) transmission losses

and hence to estimate output (to the electricity grid), from the 1981 reference design in the Department of Energy's chosen reference set of 46 seas (modified to model a site in 100 metres depth). Also to estimate output in I.O.S.'s comp reference set of 399 seas, and comment upon any significant differences.

1.2 To discuss with the device team (EWPP) whether further optimisation by altering ratings or operating strategy is desirable, and if so, to repeat the calculatio in Section 1.1 (above) with the modified design.

1.3 To prepare a brief report communicating relevant figures to RPT, together with the device team's comments on their interpretation, and possibilities for improvement.

2. Data

2.1 Front-end Power Extraction

Data on front-end power extraction were obtained from the most recent model tests carried out in the EWPP narrow tank, on 10cm diameter models mounted on a compliant rig. These tests were in mixed seas, with Pierson-Moskowitz shaped frequency spectrum, and covered a range of heights (Hrms) and periods (Te) matching the South Uist wave climate at scales from 1:100 to 1:160. That is, productivity in South Uist seas could be estimated almost entirely by interpolation between these test data (See Appendix 1). Directionality was allowed for by multiplying each spectral component by the cosine of the angle between it and the assumed facing direction of 260 degrees (i.e. 10 degre South of West).

2.2 Wave Data

The basic wave data set consisted of the 399 directional spectra for the 42m site at South Uist synthesised by John Crabb of IOS, corrected for a site in 100m depth using data from IOS's comparison of the two sites (Appendix 2).

While it was thought preferable to use all 399 sea states in optimising device parameters (see comments below, section 4.1) the basic productivity estimate uses the recommended subset of 46 sea states, according to the formula described by the Department of Energy's consultants, Rendall, Palmer and Tritton (Working Paper No. 42).

2.3 Hydraulic Losses

The figures provided by Robert Clerk (Appendix 3) indicated that these losses were well fitted by a loss function consisting of a constant (standing loss) and linear term. (The smoothing effect of the gyro at the first stage of power take-off greatly simplifies calculations, since it is thus adequate to consider only the mean input power level in each sea state.)

2.4 Electrical Losses

The figures provided by John Cure of Merz and McLellan (Appendix 4) indicated that these losses were well fitted by a loss function consisting of a constant (standing loss) plus quadratic term.

2.5 Unreliability Losses

The design philosophy of the Salter duck aims to minimise breakdowns and the need for maintenance. It is therefore thought conservative to estimate a mean loss of 10% of the power units (leading to a loss of 3% of mean output) plus a total loss for 5% of the time (from breakdown of transmission lines etc.): that is an overall loss of 8% of mean output is assumed

2.6 Costs

Cost estimates for the reference design were provided by Project Assessment and Development (structures and power module), Merz and McLellan (electrical) and Rendall, Palmer and Tritton (construction and maintenance facilities) (Appendix 5).

An amortisation* factor of 7.1% p.a. (based on the Department of Energy's 5% discount rate, over a 25 year life) was used to convert between capital and running costs.

3. Productivity of the Reference Design

3.1 The Reference Design

A preliminary optimisation of device diameter and power rating was carried out during August 1981, and taken into account in the choice of the reference design, which is 14m in diameter and 45m long. Merz and McLellan's electrical scheme includes 1872 generators with input rating 1.2MW in the 864 duck reference scheme - two per duck, plus one per six ducks for power take-off from the spine. Alternatively Project Assessment and Development propose 1800 x 1.25MW generators (2 per duck, plus 1 per 12 ducks in the spine). Either scheme leads to an overall generator rating of approximately 2248MW (57.8kW/m), with 2025MW (52.1kW/m) landed on Skye. The 'front-end' power rating is 2450MW (63.0kW/m).

3.2 Productivity

The estimated mean productivity for this reference design, using the recommend set of 46 sea states, is 22.1kW/m at the front end (see Appendix1), leading to 18.3kW/m delivered power at Uig, Skye. This represents a mean load factor of 35.2% giving a total mean productivity of 713MW from the 2GW scheme.

Given that the performance data are from uni-directional experiments, productivity can equally easily be estimated for the Institute of Oceanographi

* Amortize: "extinguish, usually by means of a sinking fund"! (concise Oxfd. Di

Sciences's full set of 399 sea states. This gives a more reliable estimate (see sec.4.1 below) of 18.8kW/m delivered power, i.e. nearly 3% higher.

3.3 Costs, Optimisation and Sensitivity Analysis

Using the cost figures referred to above (sec.2.5 and Appendix 5) the overall capitalised cost of the reference design, including maintenance, is estimated at £2.80m per duck, or £62,200 per metre. Amortising at 7.1% p.a. leads to a cost of 50.4p per hour per metre; finally dividing by the estimated mean net productivity (18.3kW/m), leads to a cost per unit of 2.75 pence/kWh. An 8% allowance for unreliability (sec.2.5) reduces the net productivity to 16.9kW/m, and increases the unit cost to 2.99p/kWh.

The effects of varying duck diameter and power rating are shown in Appendix 6. It turns out that the unit cost is insensitive to the precise choice of diameter and power rating. There is a broad plateau of near-optimal values; the reference design is not quite at the optimum, but towards the high productivity edge of this plateau. (See also section 4.3)

4. Interpretation and Possibilities for Improvement

4.1 Productivity

The recommended set of 46 spectra was chosen in a somewhat peculiar way which makes it difficult to predict whether it will overestimate or underestimate productivity of a particular device. On the one hand the recommended set excludes seas of extreme height (Hrms) and period (Te), and is also favourably biased in respect of directionality. On the other hand, the consultants' recommended method of estimating productivity (working paper no. 42) omits three categories of sea state (power < 10kW/m, Te < 7 secs, Te > 12.9 secs) which contribute 4.9% of the mean available power. This is the principal reason why the mean available power, allowing for directionality (cosine law), for the 100m depth site comes to 43.5kW/m, as opposed to the 46.9kW/m of the full Institute of Oceanographic Science's sample of 399 sea states. It is therefore not surprising that a device with broad frequency response and fairly low standing losses, such as the Salter duck, is estimated to produce nearly 3% more in the full Institute of Oceanographic Science's sample (18.8kW/m) than in the subsample of 46 (18.3kW/m).

It should also be borne in mind that the method of selection of the full Institute of Oceanographic Sciences' sample is such that it probably underestimates the long-term mean power available, perhaps by about 5% (Mollison, in "Power from Sea Waves" edited by B. Count, Academic Press 1980), and that the comparison figures used to correct for the 100m depth reference site, while they agree broadly with other evidence, are only based on data over a limited period. An informed guess (it can be no better) at a 95% confidence interval for the true mean productivity might be 19.5 ± 2.5kW/m.

Taking the full Institute of Oceanographic Sciences' sample as the most reliable present estimate, it is interesting to note that there is only limited scope for increasing productivity by improving device efficiency alone: a device with 100% efficiency and the same power limit would produce only 5kW/m more (approx. 24 instead of 18.8). If the front end power limit is raised to 100kW/m (delivered power limit 81.4kW/m), this scope for improvement increases (28.3kW/m from a 100% efficient device, 20.3 from the reference design). While improvements in efficiency are clearly worth seeking, their effect may therefore be small compared with possible cost reductions, and comparable with the effects of moving to a better site (such as the 'Cape Wrath Spur'?).

4.2 Reliability and Maintenance

As regards reliability, one advantage of the spine arrangement is that it allows power lost by failure of one or a number of power units to be taken up by units in neighbouring ducks. Thus the loss of 10% of power units is roughly equivalent to down-rating the scheme by 10%, which leads to a 3% loss of mean output, as already noted (sec. 2.5). While the collection and transmission arrangements are designed to limit the effects of failures as far as possible, more major breakdowns can simply be regarded as the loss of a proportion Pi of the system for a certain %age Qi of the time. As already noted it has been assumed that such losses are equivalent to the loss of the whole system for 5% of the time, i.e. that Σ PiQi = 5%.

For maintenance, £272,000 per duck has been allowed, following the Department of Energy's consultants' recommendation.

These figures for mean cost of output and maintenance costs may seem low, but the design philosophy is to aim for high reliability and low maintenance costs (for instance in using power units with a minimal number of moving parts in sealed canisters which can easily be detached and replaced if they prove faulty). Thus, higher figures would be taken as evidence of a design fault needing correction.

4.3 Costs

It should be noted that the gyro storage facility should raise the value attributed to a unit (kWh) of wave power. The potential of the gyros as quick-reaction spinning reserve is indeed the main reason why each duck's hydraulic system has been rated at 3.5MW; the electrical generators are rated at 2.6MW (input rating), but can run at the higher rate for 20 minutes, so that the reference scheme could provide approximately 2.7GW for such a period if required.

The final costs (Appendix 5) have risen compared with the estimates in the draft of this report largely because of requirements to strengthen the concrete structures and power module, so as to conform with standards which the device team believe to be inappropriate to wave power devices. It is expected that the design will eventually become more economical in these respects. For the same reason, the device team reject one of the possible implications of the optimisation shown in Appendix 6, namely that a smaller duck diameter might yield a lower unit cost.

Some comment seems necessary as regards maintenance and spares, where we understand that the consultants' allowances are worked out as a percentage of total costs. The device team object most strongly to this procedure, which makes a nonsense of one of their design principles, which is to be prepared to increase total costs where this will reduce maintenance and the need for spares. In any case they feel that the maintenance figure of £272,000 per duck is sufficiently generous to include allowance for refurbishment. Also, if a stock of spares is to be added to the costs, the unreliability losses (sec. 2.5), which assume that a proportion of units are out of action at any one time, should be reduced accordingly.

APPENDIX 1

Productivity Estimation

The data used were from tank experiments at scale 1:140 of a model on a suitable compliant rig, in uni-directional seas with Pierson-Moskowitz shaped frequency spectrum, as follows:

Table 1.1 Experimental Productivity data at scale 1:140

Te	Hrms	Efficiency (%)	Power Extracted (kW/m)
7.10	.52	75.8	11
	.77	75.5	25
9.47	.53	74.8	16
	.78	75.5	35
	1.02	71.0	55
11.83	.53	62.4	16
	.80	60.5	36
	1.05	57.0	59
	1.32	50.2	81
	1.55	43.2	97
	1.82	38.5	119
14.20	.55	47.1	16
	.81	42.6	31
	1.06	40.6	51
	1.33	36.5	72
	1.58	34.2	96
	1.85	30.2	115
	2.10	27.2	134
	2.37	24.7	155
15.38	.55	42.3	15
	.83	41.2	34
	1.08	35.2	50
	1.34	31.9	70
	1.60	26.3	81
	1.86	25.9	109
	2.11	23.5	127
	2.38	20.9	143

APPENDIX 1 (cont)

These data were used to estimate productivity for other values of Te and Hrms by a linear interpolation method, as follows. Efficiency values of Hrms within the experimental range were linearly interpolated. The power output in seas with different values of Te was then obtained by linearly interpolating between estimates of power extracted for experimental values of Te on either side. This method was chosen because the power extracted depends very little on Te (see figure 1.3 below), For values of Te below 7.1 (one in the set of 46, no.280), the efficiency was assumed to be the same as for a sea of the same Hrms with Te = 7.1. For values of Hrms too high for linear interpolation, the power extracted was assumed to rise linearly with Hrms from the highest interpolable value with the same Te (5 cases in the set of 46, but 3 of these lead to values which exceed the front end power limit of the reference design, and only one, no. 377, is appreciably above the interpolation range).

Full output predictions for the selected set of 46 seas are shown in table 1.2.

The process of interpolation, superimposing figures from tables 1.1 and 1.2 is illustrated in Figure 1.3.

APPENDIX 1 (cont.)

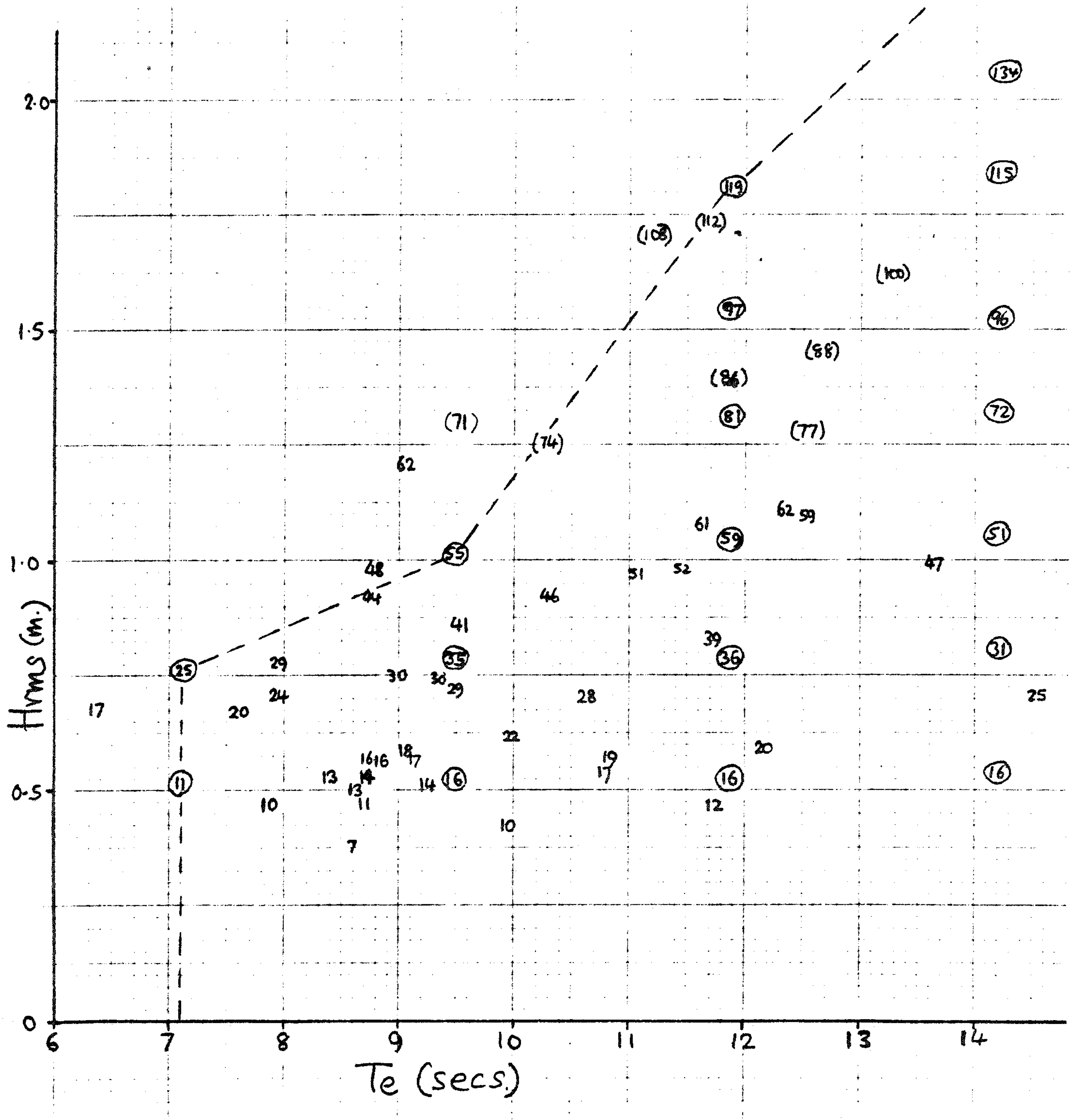
Table 1.2 Estimated productivity in the selected set of 46 seas

wt	no	Hrms	Te	p.sea	p.in	p.ltd	p.gen	p.transmitted
1.10	89	0.42	9.95	13.87	9.92	9.92	8.11	7.82
1.10	108	0.47	8.71	15.23	11.37	11.37	9.42	9.12
0.83	112	0.46	11.72	19.20	12.28	12.28	10.25	9.93
1.10	122	0.38	8.62	9.88	7.36	7.36	5.78	5.52
2.21	154	0.57	10.88	27.50	18.54	18.54	15.90	15.45
1.93	168	0.57	9.11	23.53	17.48	17.48	14.95	14.52
5.24	171	0.58	9.07	24.29	18.03	18.03	15.44	15.00
1.66	177	0.55	8.78	21.02	15.70	15.70	13.34	12.96
1.66	180	0.52	8.39	17.78	13.43	13.43	11.29	10.95
2.48	200	0.51	9.29	19.30	14.24	14.24	12.02	11.67
5.80	201	0.53	8.72	19.46	14.48	14.48	12.24	11.88
5.52	210	0.50	8.64	16.99	12.77	12.77	10.69	10.36
0.83	212	0.47	7.87	13.83	10.32	10.32	8.47	8.18
2.48	218	0.55	8.81	20.96	15.75	15.75	13.39	13.00
1.10	220	0.83	11.72	63.79	38.54	38.54	33.11	31.93
1.38	223	0.59	12.19	32.98	19.79	19.79	17.02	16.54
0.83	228	0.72	9.51	39.24	29.11	29.11	25.34	24.55
1.66	238	0.70	10.65	41.66	27.72	27.72	24.11	23.37
1.10	241	0.54	10.83	25.13	16.65	16.65	14.20	13.79
1.10	242	0.71	14.58	58.24	25.13	25.13	21.80	21.16
4.14	244	0.75	8.98	40.21	30.00	30.00	26.13	25.30
0.83	249	0.74	9.37	40.56	30.46	30.46	26.53	25.69
1.10	267	0.62	10.00	30.23	21.64	21.64	18.68	18.15
0.55	268	0.67	7.60	26.79	20.28	20.28	17.46	16.97
1.93	280	0.68	6.35	23.21	17.48	17.48	14.95	14.52
1.38	291	0.99	13.64	104.68	46.95	46.95	40.65	38.97
1.93	292	0.92	10.31	68.84	46.14	46.14	39.92	38.30
0.83	294	0.86	9.54	55.14	40.84	40.84	35.18	33.87
1.66	318	0.71	7.95	31.30	23.81	23.81	20.62	20.02
2.21	319	0.78	7.93	37.85	28.79	28.79	25.05	24.28
1.10	322	1.11	12.37	119.01	61.69	61.69	53.75	50.99
0.83	324	0.97	11.02	81.53	50.79	50.79	44.07	42.14
0.83	336	1.30	12.58	166.90	76.58	63.02	54.93	52.06
1.10	346	1.08	11.66	106.15	61.05	61.05	53.18	50.48
0.55	347	0.92	8.75	57.74	43.74	43.74	37.78	36.30
0.55	352	0.98	11.45	85.73	51.80	51.80	44.97	42.97
0.55	355	0.98	8.76	66.52	47.87	47.87	41.47	39.74
3.04	359	1.09	12.53	117.00	59.44	59.44	51.76	49.19
0.83	360	1.38	11.85	177.15	85.67			
0.83	366	1.26	10.30	129.09	74.27	63.02	54.93	52.06
1.38	371	1.45	12.67	208.33	88.00			
1.38	377	1.21	9.05	103.79	61.83	61.83	53.87	51.11
0.55	378	1.30	9.52	126.24	70.90			
0.55	381	1.62	13.26	273.70	100.31	63.02	54.93	52.06
1.10	391	1.70	11.22	254.02	107.87			
1.83	388	1.73	11.71	275.83	111.77			
		means:		43.5	24.5	22.1	19.0	18.3

APPENDIX 1 (cont.)

Figure 1.3

Scatter diagram of power extracted by the reference design (in kW/m), comparing the estimates for the 46 seas (bracketed where they exceed the front end power limit of 63.0kW/m) with values established experimentally at scale 1:140 (circled figures). The dashed line indicates the edge of the area where linear interpolation is possible.

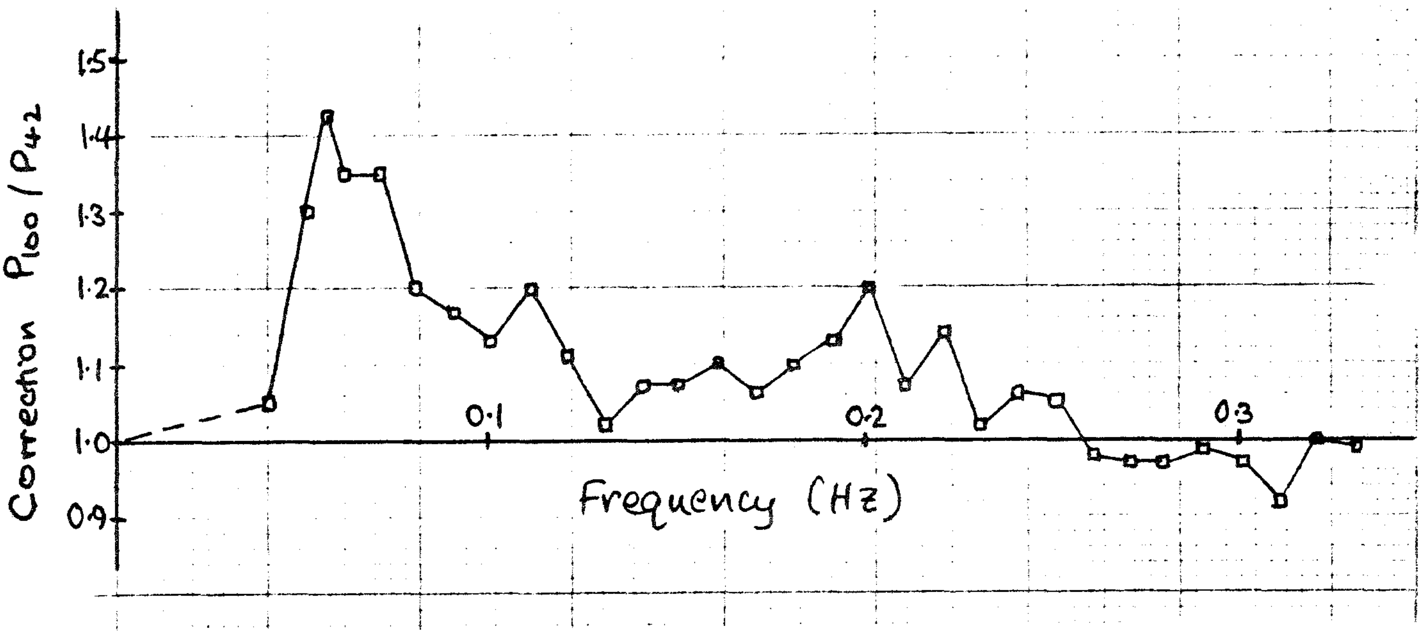


APPENDIX 2

Power Correction Factors for 100m Depth Site

(from RPT working paper No. 39)

This figure shows the ratio of power observed between IOS's waverider buoys off South Uist in 100m and 42m depths, expressed as correction factors - power at 100m buoy/power at 42m buoy - for each band of the frequency spectrum.



APPENDICES 3 & 4

APPENDIX 3

Power Losses (Hydraulic)
(Figures supplied by Robert Clerk)

Losses expressed as %age lost at full load

	Constant	Prop. to p	Prop. to p^2
Duck to backbone bearing	-	negligible	-
Ring cam pumps	-	0.35	-
Eductor pumps: on-gyro	-	0.74	-
Off-gyro	-	1.22	-
Slow speed bearings	-	0.22	-
High speed axial hydraulic pistons driving gyro discs	-	1.86	-
Gyro bearings	-	3.11	-
Axial piston driving electrical generator *	0.35(.175)	0.68	-

NOTES: Some power will also be taken out through spine joints. This involves an additional standing loss of about 0.3%, but the other hydraulic losses are lower (about 5% total) because there are no gyro bearing losses involved. It thus seems reasonably conservative to ignore this operating mode when calculating losses.

* see note to Appendix 4 below

APPENDIX 4

Power Losses (Electrical)

(Figures supplied by Merz and McLellan)

	Constant	Prop. to p	Prop. to p
Electrical generators*	3.0(1.5)	-	2.0(4.0)
Transformers)) Cables ashore)	0.42	-	4.75

* NOTE: At less than half of full load, one of the two generators in each duck can be shut down, leading to the loss figures shown in brackets.

APPENDIX 5

Reference Design Costs

Item	£000/duck	Source of information	Dependence on diameter D or power rating P
CONCRETE STRUCTURES			
1.1 Duck bodies			
Ballast trim			
2.1 Spine cylinder	1137	P.A.D.	$\frac{1}{4}C+\frac{1}{2}D+\frac{1}{4}D^2$
2.2 Spine joint			
3 Spine-duck bearing			
- Assemble and test	35	P.A.D.	
POWER MODULE			
1.2 Pressure vessel			
1.3 Gyro flywheel			
Hydraulic motors	732	P.A.D.	$\frac{1}{8}C+\frac{3}{8}P+\frac{1}{2}D$
Control etc.			
Assembly			
4 Mooring	45	P.A.D.	$\frac{1}{2}C+\frac{1}{2}D$
5 Electrical generation and transmission to Skye	366	M & M	$\frac{1}{4}C+\frac{3}{4}P$
6 Construction facility	116	R.P.T.	P
7 Float-out and installation	97	R.P.T.	$\frac{3}{4}C+\frac{1}{4}D$
8 Maintenance	272	R.P.T.	C
TOTAL	£2.80m/duck	-	21/30/38/11% ($C/P/D/D^2$) (see sec. 3.3)
Mean output	825kW/duck (759)	(see secs. 3.2, 4.2)	-
Unit cost	2.75p/kWh (2.99)	(Amortising @ 7.1% p.a.)	

APPENDIX 6

Optimisation of Unit Costs

This shows mean output per unit cost (reference design = 100 (marked *)) a

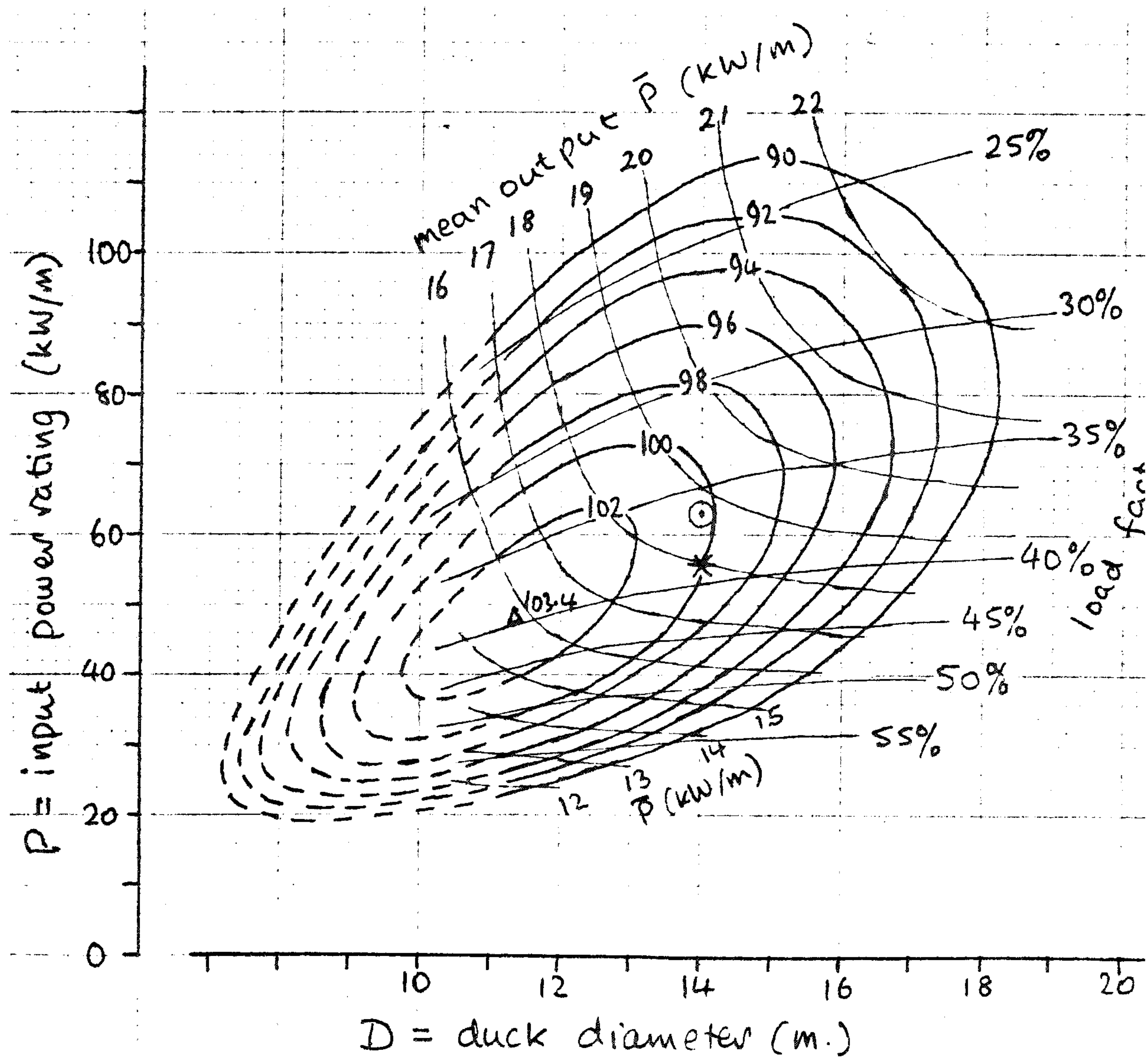
a function of duck diameter D and input ('front-end') power rating P, assu

a cost breakdown of 28/28/28/16 for the reference design, that is 28% cons

28% each proportional to P and D, and 16% to D^2.

NOTE: this refers to the draft (Oct. 1981) reference design. The revised (Nov 1981) design is at the point circled.

It should be noted that this cost formula is bound to become unrealistic f large departures from the reference design, particularly for variation in reduced below about 10m it sinks.



Duck Productivity Supplement 2nd December, 1981.

To be read in conjunction with "Productivity Analysis of Salter Ducks (1981 reference design)" (revised version, 25th November, 1981).

Denis Mollison - (Heriot-Watt University)

The unit cost of the 1981 reference scheme has been recalculated, allowing for some additional costs recommended by Rendel, Palmer and Tritton, and using a discounted cash flow. The revised estimate of the unit cost is 3.0p/kWh, or 3.3 allowing for losses through unreliability. The sensitivity to various assumptions is examined, especially as regards costs, reliability, lifetime and the discount rate.

Details

The revised costs, with the years in which they become payable are as set out in Table 1. The net present cost, discounting at 5%, is £2,027m for the scheme. The discounted productivity is 67.3×10^9 kWh, or 61.9 allowing for losses through unreliability, leading to the unit costs given above. As will be seen from Table 1, it has been assumed that the main capital costs are paid in year 0 (construction facilities), year 2 (maintenance facility and half of the basic transmission scheme) and year 6 (remainder of the basic transmission scheme). One string of 108 ducks is built in each of years 2 to 9 (total 864 ducks), together with its connections to the transmission scheme. Each string produces power for 25 years, starting with the year after it is built. Maintenance costs of £8.9m p.a. are paid in each of years 3 to 34, although the full scheme is only in operation in years 10 to 27; this may be thought of as allowing for extra maintenance to cope with any teething troubles in years 3 to 9.

TABLE 1 DISCOUNTED CASH FLOW FOR DUCKS

I COSTS (£million)	basic cost	discounting factor	net present cost
(a) Capital costs			
Year 0: Construction facility	116	1.0	146
Mech. build and test facility	30		
Year 2: Maintenance facility	45	0.906 (i)	108
½ basic transmission scheme	74		
Year 6: Other ½ ditto	74	0.744 (i)	55
TOTAL	339		309
(b) Annual Manufacturing Costs			
Years 2 - 9 (108 ducks p.a.):			
Civil structures	139.0 p.a.	6.14 (ii)	854
Civil construction facility	3.6 "		22
Mechanical components	68.6 "		421
Mechanical facility	1.1 "		7
Plus 1% spares (1)	0.7 "		4
Float-out	9.2 "		57
Installation and mooring	6.2 "		38
Connection to transmission scheme	21.0 "		129
Years 14 - 21: renew moorings (2)	-		-
TOTAL	249.4 p.a.		1532
(c) Maintenance			
Years 3 - 34: Maintenance (3)	8.9 p.a.	14.25 (ii)	127
Years 3 - 27/10 - 34: Refurbishment (1% of mech. components p.a.)	5.5 "	10.78 (iii)	59
TOTAL	14.4 p.a.		186
TOTAL NET PRESENT COST OF SCHEME			£2027m

II PRODUCTIVITY	mean output	discounting factor	
Years 3 - 27/10 - 34:	713MW	10.78	67.3×10^9 k
ditto, allowing for losses through unreliability	656MW	"	61.9 "

III UNIT COST

Divide £2027m by 69.3×10^9 kWh, giving 3.01p/kWh;

or, allowing for unreliability, divide by 61.9, giving 3.27p/kWh.

N O T E S

Comments

(1) The allowance for spares may seem small, at 1% of the mechanical components' cost, but it should be noted that our unreliability losses already allow for reduced output when components are out of action; to have a large allowance for spares as well would be double accounting. In any case, a substantial allowance has been made for refurbishment.

(2) Because ducks have low stress compliant moorings, it is thought unnecessary to renew them during the scheme's 25 year life.

(3) Including a full maintenance cost from year 3 makes implicit allowance for heavier maintenance per duck during the installation period (years 3 - 9) due to any teething troubles.

Discount Factors

If the annual discount factor is x, here = 1/1.05 = .952, the discounting factors are given by:

(i) for capital cost in year n, d.f. = x^n

(ii) for cost equally spread over m years, starting in year n,

$$\text{d.f.} = x^n g(m), \qquad \text{where } g(m) = (1-x^m) / (1-x)$$

(iii) for a sum divided into k equal parts, each spread over m years (as in (2) above), with starts spread over the k years beginning in year n,

$$\text{d.f.} = x^n g(m) g(k)/k$$

Sensitivity Analysis

The unit cost is sensitive to the following main factors: manufacturing costs, maintenance costs and unreliability, delays in construction, lifetime of scheme and above all the assumed discount rate.

1. Manufacturing costs: On the civil side, the costs are dominated by the steel post-tensioning and reinforcements; the design team believe that these amounts can be reduced by at least half. On the mechanical side, there should be scope for reductions in costing if full account is taken of the advantages of mass production of items which have previously only been produced in small numbers.

Overall it seems quite possible that the costs of manufacture, and hence the unit cost, could be reduced by about one-third.

2. Maintenance costs and unreliability: Losses from unreliability are magnified by the additional costs of maintenance that they impose (see Figure 1). While a loss of 20% would be regarded as quite satisfactory in the first year of operation a similar loss rate over the 25 year lifetime would be very poor. The device team believe that we can and should achieve at worst the 8% mean loss allowed for in our calculations of unit cost.

3. Delays in construction: Like all wave energy designs, the scheme is modular. The major costs of manufacture of each part of the scheme begin to be recouped in the following year; the basic capital costs represent only 15% of the net present cost of the scheme. Thus if part or all of the scheme is delayed by a year its share of the costs is increased by only about one-sixth of the assumed discount rate, i.e. less than 1%.

4. Lifetime of scheme: The sensitivity of the unit cost to variations in the assumed lifetime of 25 years is shown in Figure 2. The unit cost decreases by about 2% for each extra year of life between 20 and 30 years; the limiting value for an assumed infinite life is about 2.4 p/kWh. (This result depends heavily on the assumed discount rate of 5% - see below.)

One interesting case is where the concrete structures are assumed to have a very long (100 year) life, while the mechanical components are replaced every 25 years. This leads to a unit cost of 2.6p/kWh, 20% below the reference case. (With a lower discount rate the saving would of course be more marked.)

5. Discount rate: Finally, the sensitivity to variations in the assumed discount rate is shown in Figure 3. With real energy prices likely to continue to rise, and government stock with a real return of 2 to 3% selling well, it seems difficult to justify such a high discount rate as 5% for energy projects.

If we both increase the lifetime and lower the discount rate, we of course produce a combined decrease in unit cost, as shown in Figure 4.

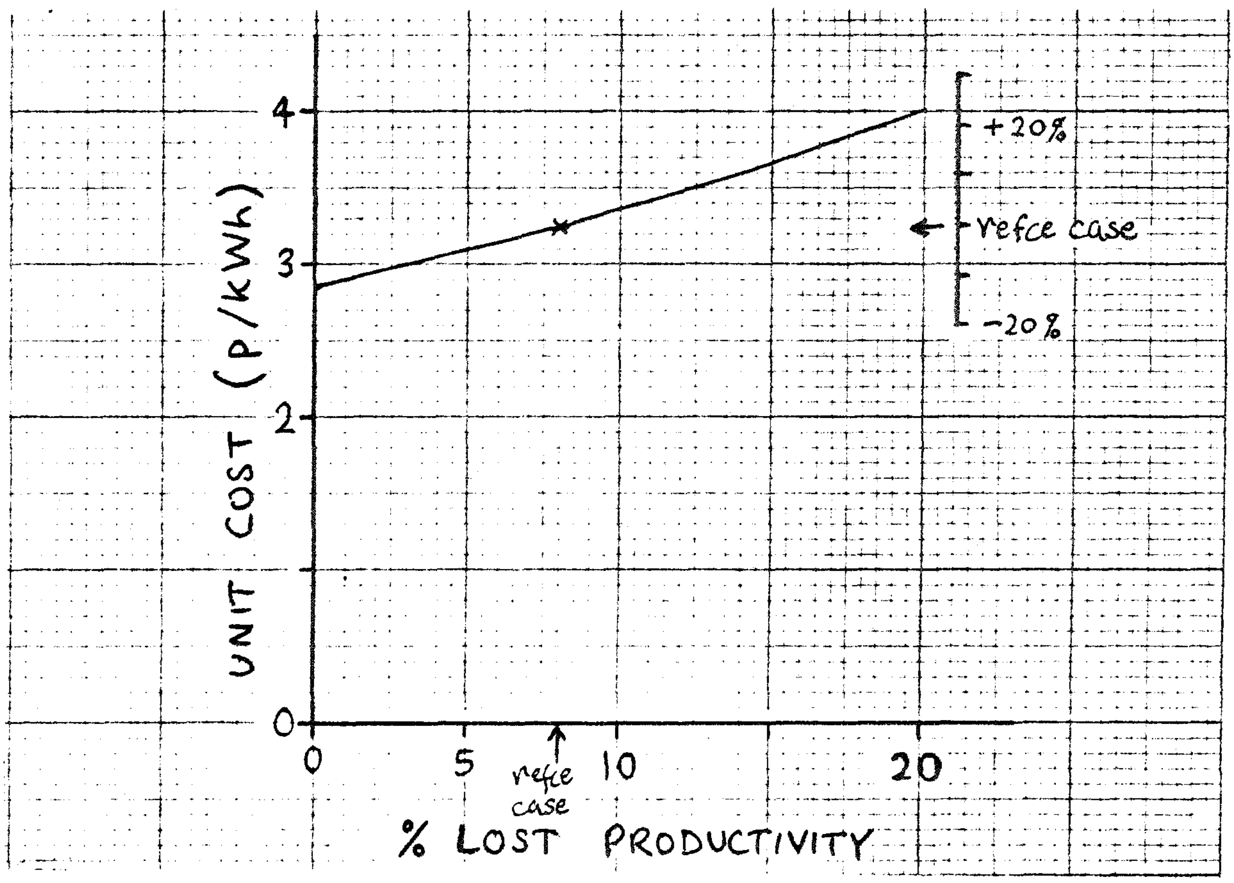


Figure 1: Sensitivity to lost productivity (The costs of maintenance and refurbishment have been assumed proportional to 0.5+ 0.5(u/8) where u is the % age of lost productivity.)

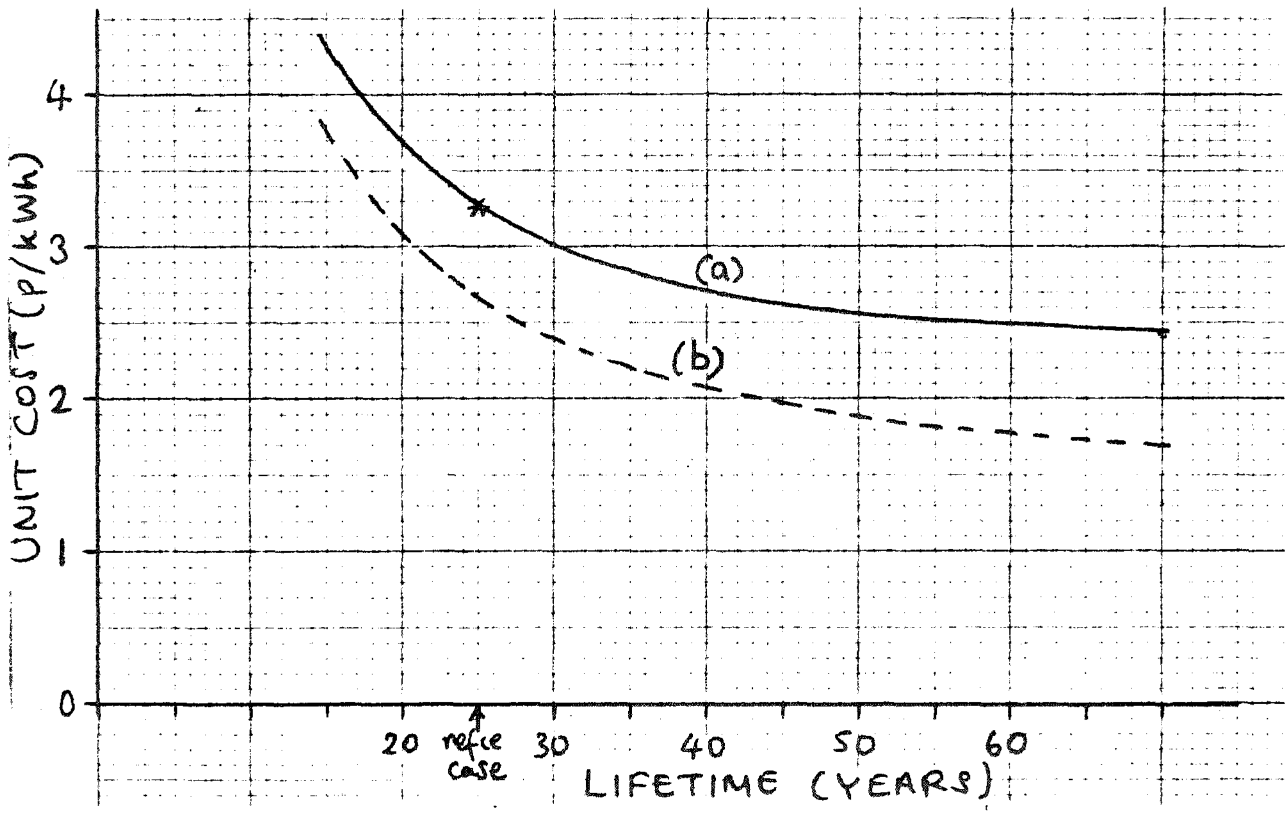


Figure 2: Dependence of unit cost on lifetime of Salter ducks:

(a) reference case, assuming 5% discount rate (solid line),

(b) at 3% discount rate (dashed line).

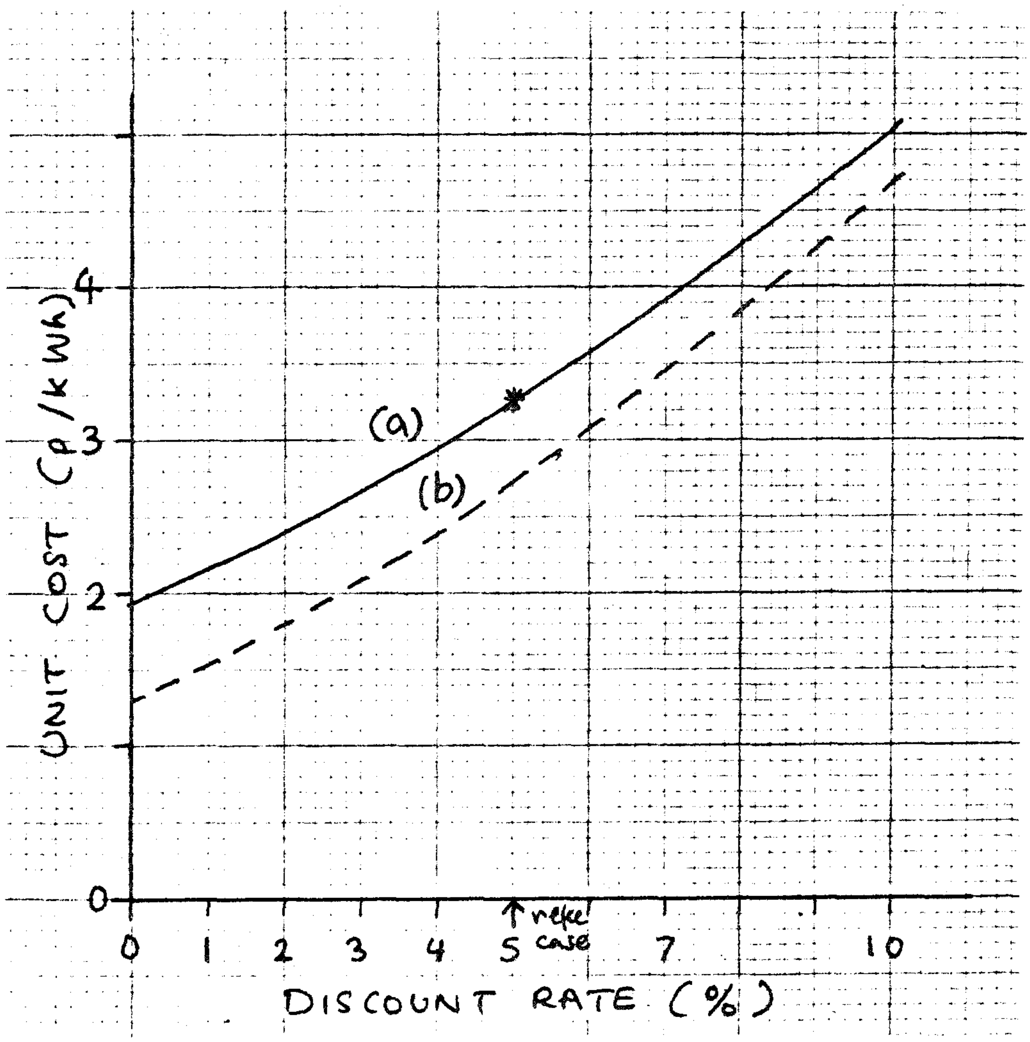


Figure 3: Dependence of unit cost on discount rate:

(a) reference case, assuming 25 year lifetime (solid line),

(b) assuming 40 year lifetime (dashed line).

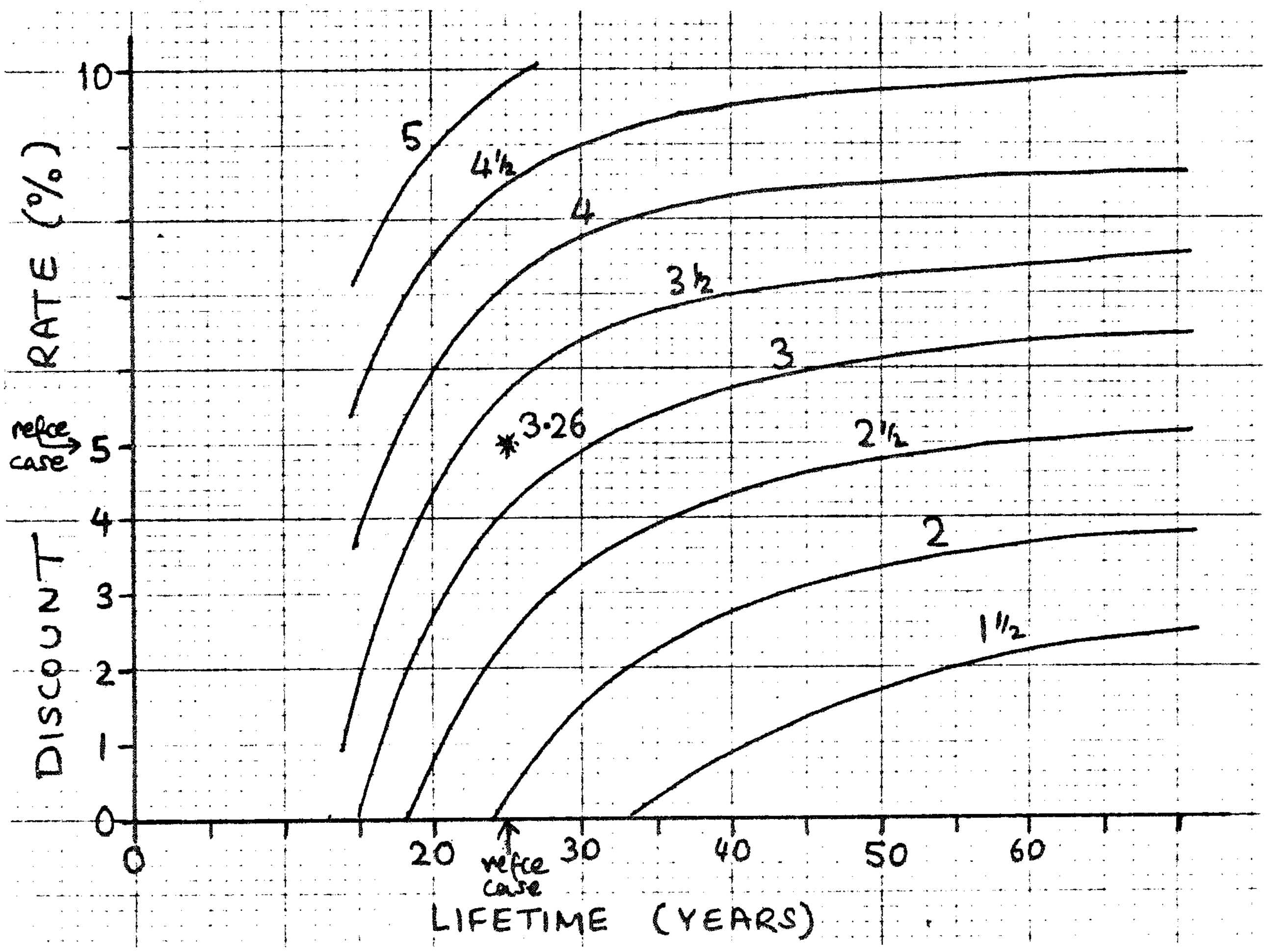


Figure 4: The dependence of unit cost (in p/kWh) on lifetime and discount rate.

7.2

5% DISCOUNTED CASH FLOW

(9 YEAR BUILD PERIOD - 25 YEAR LIFE)

864 DUCKS - 2 GW STATION

YEAR		£
0	CAPITAL COST	$= -339 \times 10^6$
1	$[-96 \times 2.261 \times 10^6] \times 0.952$	$= -206.64 \times 10^6$
2	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + 96\ W] \times 0.952^2$	$= -208.59 \times 10^6 + 87.0\ W$
3	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (2 \times 96W)] \times 0.952^3$	$= -198.85 \times 10^6 + 165.7W$
4	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (3 \times 96W)] \times 0.952^4$	$= -189.42 \times 10^6 + 236.6W$
5	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (4 \times 96W)] \times 0.952^5$	$= -180.44 \times 10^6 + 300.3W$
6	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (5 \times 96W)] \times 0.952^6$	$= -171.70 \times 10^6 + 357.3W$
7	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (6 \times 96W)] \times 0.952^7$	$= -163.41 \times 10^6 + 408.2W$
8	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (7 \times 96W)] \times 0.952^8$	$= -155.82 \times 10^6 + 453.4W$
9	$[(-96 \times 2.261 \times 10^6) - (13.1 \times 10^6) + (8 \times 96W)] \times 0.952^9$	$= -148.45 \times 10^6 + 493.3W$
10-29.5	$[-8.9 \times 10^6] + (9 \times 96W) \times K = 8.08$	$= -71.91 \times 10^6 + 6981.1W$
		$= (-2034.23 \times 10^6) + 9482.9W$

W = Output of one Duck for one year

Assume Mean Output of 761 KW/Duck (16.9 KW/m) then:

$$£\ \frac{2034.23 \times 10^6}{9482.9 \times 0.761} = £0.28/\text{MWYr}$$

$$= 3.22\text{p/Kwhr}$$

7.3

TABLE 1

EDINBURGH UNIVERSITY WAVE ENERGY SCHEME

APPROXIMATE GENERATION AND TRANSMISSION LOSSES

	Normal load (%)	Fixed losses (MW)	Variable losses (proportional to power^2) (MW)
2248 MW Electrical Generation (3%* fixed +2% variable) *Individual 1.2 MW generators	100	67.5	45
128 km 3.3 kV cables @ 100 watts/metre	100	-	12.8
144 - 3.3/33 kV 20 MVA transformers	100	3.5	20.2
33 kV cables 22 km @ 40 watts/metre) 86 km @ 25 watts/metre)	100	-	3.0
8 - 33/132 kV 300 MVA transformers	100	1.3	6.0
270 km 132 kV 1600 mm^2 Cable @ 130 watts/metre	80	-	35.1
72 km 132 kV of O/H line @ 100 watts/metre	80	-	7.2
3 x 1100 MVA 132/400 kV transformers	67	3.2	5.3
3 x 35 km 400 kV 1300 mm^2 XLPE cables @ 110 watts/metre	67	1	12.0
Total individual losses		76.5	146.6
Total losses		223.1 MW	

SECTION 8

COSTING OF THE 2 GW SCHEME

8. COSTING OF THE 2GW SCHEME

8.1 Device Team's Costing

8.1.1 Summary of Costs

8.1.2 Electrical Costs

8.1.3 Component Cost Analysis Sheets

8.1.4 Bills of Quantities

8.1.5 Base Fleet Costs

8.1.6 Installation

8.1.1

SUMMARY OF COSTS - 864 DUCKS

(9 YEAR BUILD PERIOD - 25 YEAR LIFE)

2 GW STATION

	1	2 (1 x 96)	3 (1 x 864)	4	5	6
DESCRIPTION	DUCK SPECIFIC PER DUCK (£ x 10^3)	DUCK SPECIFIC ANNUAL COST (£ x 10^6)	DUCK SPECIFIC TOTAL COST (£ x 10^6)	SCHEME CAPITAL COST (£ x 10^6)	SCHEME ANNUAL COST (£ x 10^6)	SCHEME TOTAL (£ x 10^6)
CIVIL CONSTRUCTION FACILITY	-	-	-	116.0	3.2 (9)	144.8
MECHANICAL BUILD AND TEST FACILITY	-	-	-	30.0	1.0 (9)	39.0
CIVIL STRUCTURES	1287.0	123.0	1112.0	-	-	1112.0
MECHANICAL COMPONENTS	635.6	61.0	549.2	-	-	549.2
FLOAT OUT	85.6	8.2	74.0	-	-	74.0
INSTALLATION AND MOORINGS	57.8	5.5	48.5	-	-	48.5
ELECTRICAL GENERATION AND TRANSMISSION	194.8	18.7	168.3	148.0	-	316.3
MAINTENANCE	-	-	-	45.0	8.9 (25)	268.0
TOTALS	2260.8	216.2	1952.0	339.0	13.1 (9) +8.9 (16)	2551.8

8.1.2 Electrical Costs

We have made use of equipment costs collected from the many contracts being engineered by the firm and additionally we have obtained budgetary quotations for items. This information has enabled us to draw up mean price levels and to plot relationships between rating and cost.

In general, these cost trend curves have been used to examine alternative arrangements and as a basis of the cost estimates.

For the cabling within and parallel to the spine, the point-to-point distances have been multiplied by a factor of 1.1 to determine the lengths of cable. For the other submarine cables and the overhead line routes a multiplying factor of 1.2 has been used.

The cost of installation for the main submarine cables has been derived from discussion with Wharton Williams who have experience of the laying of oil pipes. The laying equipment and procedures required for cables is not dissimilar to those for oil pipes, and experience of high voltage cable laying is irrelevant because of the difference in scale of this project. Other cables, such as those within the spine have been allocated 30 per cent of the supply cost to cover installation.

All the transformers and switchgear have been allocated 40 per cent of supply cost for installation. This is higher than for normal application where 25 per cent increase is commonly used.

The overhead line costs are based upon normal practice and the generator installation costs also; in the latter case 25% has been used.

The item costs are detailed in the following table. The contingency charges have been agreed with the Consultants.

8.1.2 COST ESTIMATES

	No.	Rate £	Total £	Total £
ITEM 1 280MW SPINE & DUCK EQUIPMENT				
1.2MW Gen + Switchgear 3.3kV	234	46k	10,764,000	
3.3kV Flexible cables 20m of 50mm^2	216	22/m	95,040	
3.3kV Cable 95mm^2. 18x890m		12.3/m	197,046	
3.3kV Barrier bushings	216	84	18,144	
3.3kV Fuses + busbars	234	50	11,700	
3.3/33kV 20MVA transformer	18	83k	1,494,000	
33kV Cable 185mm^2 2673m		41/m	109,593	
33kV Cable 400mm^2 10692m		52/m	555,984	
33kV Barrier bushings	156	140	21,840	
33kV 5 Panel Sw.Boards	18	130k	2,340,000	
33/132kV 300MVA transformer	1	840k	840,000	
132kV 4 Sw.Panel. 6 per 8 transformer	0.75	560k	420,000	
280MW Spine & Duck Equipment			16,867,347	16,867,000
ITEM 2 TRANSMISSION TO S.UIST				
132kV 1600mm^2 Cable 45km	6	255/m	68,850,000	
132kV 1600mm^2 Intercon.Cable 6446m	5	255/m	8,218,650	
132kV 1000mm^2 Intercon.Cable 6446m	2	225/m	2,900,700	
Control & Monitoring cable 181km		30/m	5,430,000	
2000MW Sea-Shore Transmission			85,399,350	85,400,000
ITEM 3 S.UIST AND SKYE				
132kV 1½ CB. Bays	8	370k	2,960,000	
132kV 50 MVAR Shunt reactors	2	210k	420,000	
132kV O/H Lines Double CCT Twin ACSR 250mm^2	72km	80k	5,760,000	
132kV 1½ CB Bays	7	370k	2,590,000	
132/400kV 1100MVA Transformer	3	1.7m	5,100,000	
33kV, 100MVAR Shunt reactors	3	240k	720,000	
33kV Swgr + Conns	3	70k	210,000	
Monitoring & control	1		4,000,000	
400kV 1½ CB Bays	3	1.2m	3,600,000	
400kV Cable 1300mm^2 1056km		345/m	36,230,000	
			61,590,000	61,590,000

ITEM 4 TOTAL COSTS - 2246MW

		Total 1981 £	Total 1979* £
2246 MW Generation		86,112,000	95,680,000
Spine Collection		(48,827,000)	
	126,125,000	( )	129,000,000
Spine-Shore Transmission		(85,400,000)	
S.Uist Transmission		25,360,000	26,400,000
Cables to Skye		36,230,000	35,000,000
		281,929,000	286,080,000

Contingencies

Submarine items 15%			
Others 10%			
Submarine costs	(121,629,000)	18,244,000	
Other costs	(160,299,000)	16,030,000	
Engineering & Management costs not included			
Total contingencies		34,274,000	44,342,000‡

SCHEME COST	£316,203,000	330,422,000

* 1979 Costs increased by 20% for inflation and for the increased generator and cable route lengths

‡ Comprises 5% engineering and 10% contingencies as in the 1979 report.

C O M P O N E N T   C O S T   A N A L Y S I S

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £$x10^6$
1.	DUCK UNITS									
1.1	BODY									
	CONSTRUCTION:		)						)	
	CEMENT		)						)	
	AGGREGATE		)						)	
	FORMWORK		)						)	
	REINFORCEMENT		)						) 445.3	384.7
	PRESTRESSING		)	SEE BILL OF QUANTITIES APPENDED					)	
	STRAND		)						)	
	ANCHORAGES		)						)	
	ATTACHMENT		)						)	
							TOTAL THIS PAGE		445.3	384.7
							RUNNING TOTAL			

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
1.	DUCK UNITS (Contd.)									
1.2	POWER CANISTER									
	CANISTER SHELLS	2	STEEL		150	975	RPT		146.25	126.4
	MISCELLANEOUS STIFFENING	2 SETS	STEEL		5	370	RPT		1.85	1.6
	ASSEMBLY OF MACHINERY	2 SETS	-	-	350	400	CURRENT CONTS.		140.00	121.0
	TESTING				350	50			17.5	15.1
	FLOTATION TO SITE		)							
	ASSEMBLY ONTO DUCK		) INCLUDED IN BILL OF QUANTITIES APPENDED							
	INSTALLATION OF MOORINGS		)							
							TOTAL THIS PAGE		305.6	264.1
							RUNNING TOTAL		750.9	648.8

C O M P O N E N T   C O S T   A N A L Y S I S

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
1.	DUCK UNITS (Contd.)									
1.3	POWER T/O SYSTEM (DUCK)									
	GYRO DISCS	4	STEEL	1	123	400	QUOT.		49.2	42.5
	GYRO SHAFT ASSEMBLY	4			35	1000			35.0	30.2
	GYRO SHAFT BEARINGS	8			11	1200			13.2	11.4
	GYRO SPACE FRAME	4			80	500			40.0	34.6
	FRAME BEARINGS	16			22	1000			22.0	19.0
	DRIVE HYD. MOTORS	8			5	6000ea	QUOT		48.0	41.5
	RING CAM PUMPS	256			35	500			17.5	15.1
	RING CAMS	8			12	500			6.0	5.2
	R.C.P. MOUNTING	64			2	370	RPT		0.7	0.6
	SUPPORT STRUCTURE	2 SETS			10	370	RPT		3.7	3.2
	HYDRAULIC PIPEWORK	2 SETS			15	600	QUOT.		9.0	7.8
							TOTAL THIS PAGE		244.3	211.1
							RUNNING TOTAL		995.2	859.9

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
1.	DUCK UNITS (Contd.)									
1.3	POWER T/O SYSTEM (Contd.)									
	GENERATOR DRIVE HYD. MOTOR	2			1.3	6000 ea	QUOT.		12.0	10.4
	EDUCTOR PUMP	10			0.7	750			0.5	0.4
	BLACK START EQUIPMENT	2			0.5	750	QUOT.		0.4	0.35
	VACUUM PUMP	2			0.5	750	QUOT.		0.4	0.35
	HEAT EXCHANGER	2			1.0	750	QUOT.		0.8	0.7
							TOTAL THIS PAGE		14.1	12.2
							RUNNING TOTAL		1009.3	872.1

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE									
2.1	SPINE CYLINDER									
	CONSTRUCTION:									
	CEMENT									
	AGGREGATE									
	FORMWORK									
	REINFORCEMENT			SEE BILL OF QUANTITIES APPENDED					482.0	416.5
	PRESTRESSING									
	STRAND									
	ANCHORAGES									
	ATTACHMENT									
	SUPPLY & INSTALL EPOXY JOINT								15.0	13.0
								TOTAL THIS PAGE	497.0	429.5
								RUNNING TOTAL	1506.3	1301.6

N.B. See Note on page 9.

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE (Contd.)									
2.1	SPINE CYLINDER (Contd.)									
	ASSEMBLE SPINES & DUCKS (Including supply of bearings, Hooke's Joint shroud and seal)	)	INCLUDED IN BILL OF QUANTITIES							
	FIX BALLAST TANKS IN SPINES	)	INCLUDED IN BILL OF QUANTITIES							
	WALKWAYS, AIRLOCKS & MISC. STEELWORK	)	INCLUDED IN BILL OF QUANTITIES							
2.2	SPINE JOINT									
	HOOKE'S JOINT STRUCTURE									
	CEMENT	)	Included in Spine							
	AGGREGATE	)	Included in Spine							
	REINFORCEMENT	)	Included in Spine							
	FORMWORK	)	Included in Spine							
							TOTAL THIS PAGE		-	-
							RUNNING TOTAL		1506.3	1301.6

N.B. See Note on page 9.

C O M P O N E N T   C O S T   A N A L Y S I S

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE (Contd.)									
2.1	SPINE JOINT (Contd.)									
	HOOKE'S JOINT SPIDER	½			75	370			27.8	24.0
	HOOKE'S JOINT BEARINGS	2			12	1000	)			
							) Included in B.O.Q.'s			
	RUBBER HOOKE'S JOINT BEARINGS				2	500	)			
	RAMS	4			25	15000ea	QUOT.		60.0	51.8
	RAM SUPPORT	4			30	370	Included in Civil		11.1	9.6
	RAM ENDS (MECHANICAL)	8			64	1000			64.0	55.3
	RAM ENDS (RUBBER)	8			0.3	500			0.2	0.2
							TOTAL THIS PAGE		163.1	140.9
							RUNNING TOTAL		1669.4	1442.5

N.B. See note on page 9.

C O M P O N E N T   C O S T   A N A L Y S I S

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE (Contd.)									
2.2	SPINE JOINT (Contd.)									
	CLOSING MEMBERS:									
	PART SPHERES	1	STEEL		22	370			8.1	7.0
	CEMENT		N/A							
	AGGREGATE		N/A							
	FORMWORK		N/A							
	ROLLING SEALS	0.5	RUBBER		2	12000 ea	QUOT.		6.0	5.2
	END MEMBRANE	)								
		) Included in Spine								
	BULKHEAD CLOSURES	)								
	RAM ENCLOSURES		STEEL		5	370			1.9	1.6
	PRESSURISATION EQUIPMENT	0.5				5000ea			2.5	2.1
							TOTAL THIS PAGE		18.5	15.9
							RUNNING TOTAL		1687.9	1458.4

N.B. See Note on page 9.

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE (Contd.)									
2.3	POWER TAKE OFF SYSTEM (MECHANICAL)									
	HYD. MOTOR (PRIMARY-SURGE)				0.1				1.5	1.3
	HYD. MOTOR (PRIMARY-HEAVE)				0.1					
	HYD. MOTOR (SECONDARY-SURGE)				0.05					
	HYD. MOTOR (SECONDARY-HEAVE)				0.05				1.13	1.0
	HYD. MOTOR (GEN. DRIVE)	0.17			0.05	FROM PRIMARY SYSTEM				
	HYD. MOTOR (FLYWHEEL)				0.05					
	FLYWHEEL LAMINATIONS				2					
	FLYWHEEL SHAFT ASSY.				0.55				10.25	8.9
	FLYWHEEL BEARINGS				0.2					
	FLYWHEEL MOUNTING									
	HYDRAULIC PIPEWORK				0.5	£450/100m (QUOT. - BSC) (£1.6m total)				
							TOTAL THIS PAGE		12.9	11.2
							RUNNING TOTAL		1700.8	1469.6

N.B. For all spine equipment, where an item applies to a number of ducks, weights are divided by that number to give weights per duck.

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
2.	SPINE (Contd.)									
2.3	POWER TAKE OFF (Contd.)									
	(Electrical)	SEE SECTION 5								

	COST PER DUCK (£000's)	TOTAL COST £x10^6
TOTAL THIS PAGE	-	-
RUNNING TOTAL	1700.8	1469.6

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
3.	DUCK/SPINE BEARING									
	FERRITE MAGNETS				50	£1200	Mullard	60,000	)	
	MOUNTING RUBBER				850m	£ 50/m	) QUOT.	(42,500)	)	
	OUTER SHROUD				425m	£ 45/m	) DUNLOP	(19,000)	)	
	BELLOWS				1700m^2	£ 10/m^2		(17,000)	) 175	151.2
	INSTALLATION								)	
	RETAINING MEMBER				425m	£ 45/m		(19,000)	)	
	DOSING EQUIPMENT					8900		8,900	)	
	AXIAL LOCATION					1000ea		8,600	)	
							TOTAL THIS PAGE		175.0	151.2
							RUNNING TOTAL		1875.8	1620.8

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
3.	DUCK/SPINE BEARING (Contd.)									
	ALTERNATIVE:									
	MECHANICAL BEARING									
	ROLLERS				2260	20	45200		)	
	CHAIN				100 ft	60/ft	6000		) 60	51.8
	ATTACHMENTS						3800		)	
	INSTALLATION						5000		)	
								TOTAL THIS PAGE	60.0	51.8
								RUNNING TOTAL	1935.8	1672.6

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST • PER DUCK (£000's)	TOTAL COST £x10^6
4.	<u>MOORING</u>									
4.1	<u>RODES</u>									
	ROPES		PARAFIL							
	TERMINATIONS		S. STEEL							
4.2	<u>BUOYS</u>			PRICED AT 330 x 10^3 PER MOORING (RPT) x 864					30.0	25.9
	SINKERS									
	RISERS									
4.3	<u>ANCHORS</u>									
	MASS CONCRETE									
4.4	<u>INSTALLATION</u>			INCLUDED IN 1.2						
							TOTAL THIS PAGE		30.0	25.9
							RUNNING TOTAL		1965.8	1698.5

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
5.	ELECTRICAL SYSTEM									
	2246MW GENERATION	)							99.7	86.1
	SPINE COLLECTION	)							56.5	48.8
	SPINE - SHORE TRANSMISSION	)							99.0	85.5
	S. UIST TRANSMISSION	)		FOR DETAILS SEE MERZ & McLELLAN SHEETS APPENDED					29.4	25.4
	CABLES TO SKYE	)							41.9	36.2
	SUBMARINE ITEMS	)							21.0	18.2
	CONTROL SYSTEM	)							18.5	16.0
							TOTAL THIS PAGE		366.0	316.2
							RUNNING TOTAL		2331.8	2014.7

C O M P O N E N T   C O S T   A N A L Y S I S

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
6.	CONSTRUCTION									
6.1	CIVIL FACILITY									
	SHIP LIFT	2				29x10^6	RPT		67.1	58.0
	ASSEMBLY YARD			7.9		)				
	WORKSHOPS					)				
						)113.8	RPT		131.7	113.8
	YARD SERVICES			4.70		)x 10^6				
	PERSONNEL ACCOMMODATION					)				
6.2	MECHANICAL FACILITY									
	WORKSHOPS	1				)				
						) 30m	Est's.		34.7	30.0
	WELDING EQUIPMENT					)				
							TOTAL THIS PAGE		233.5	201.8
							RUNNING TOTAL		2565.3	2216.5

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £x10^6
7.	TOW-OUT COSTS									
7.1	POWER CANISTER/DUCK ASSEMBLY								45.0	38.8
	DUCK/SPINE ASSEMBLY								)	
									) 57.8	49.9
	SPINE JOINT ASSSEMBLY								)	
	TOW-OUT STRING								85.6	74.0
								TOTAL THIS PAGE	188.4	162.7
								RUNNING TOTAL	2753.7	2379.2

COMPONENT COST ANALYSIS

REF.	ITEM	QTY. PER DUCK	MATERIAL	SIZE DEPEND. C,D,D^2 E ETC.	WEIGHT (TONNES)	COST RATE (£/T)	SOURCE (COST)	% COST	COST PER DUCK (£000's)	TOTAL COST £$x10^6$
8.	<u>MAINTENANCE FACILITIES</u>									
8.1	<u>BUILDINGS</u>									
	WORKSHOP					$35x10^6$ea	RPT		40.5	35.0
	LIFTING DEVICES					$10x10^6$ea	RPT		11.6	10.0
	OPERATING COSTS					£$8x10^6$/yr	NOT CAPITAL COST			
	OPERATING COSTS - LIFTING DEVICES					£0.9 $x10^6$/yr	NOT CAPITAL COST			
8.2	<u>SPARES</u> - 5% of Total								147.7	127.6
							TOTAL THIS PAGE		199.8	172.6
							RUNNING TOTAL		2953.5	2551.8

* Per Annum

8.1.4

BILL OF QUANTITIES

BILL OF QUANTITIES - PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	£ MILLION RATE	£
	COST CENTRE 1.1 PROVISION OF CONSTRUCTION FACILITY				
1.1.1.	Provision of General Items (including 500m of causeway)			Sum	26.7(
1.1.2.	Provision of Pre Cast Concrete Factory (area to be established by Development Team.)	—	m^2	—	
1.1.3.	Provision of other Buildings			Sum	3.7(
1.1.4	Provision of Services			Sum	1.3!
1.1.5.	Provision of large span roofed area (if required).	10000	m^2	.0005	5.0(
1.1.6	Provision of road and rail connection	10	km	.850	8.5(
1.1.7	Provision of particular items to complete the type of facility selected by Development Teams Shiplift	2	No	29.00	58.0(
	CONSTRUCTION ITEMS			Sum	.5!
	Sub Total				103.8(
1.1.8	Operation and maintenance of General Items, Buildings and Services (Duration to be established by Development Teams).	10	Year	1.00	10.0(
	TOTAL for ONE FACILITY				113.8(
	Estimated number of Devices constructed in ONE facility 864				
	HENCE proportion of facility cost per Device Unit	.116%	=	£131718	

c/f to SUMMARY

BILL OF QUANTITIES – PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	RATE	£ 000
	COST CENTRE 1.2. CONCRETE STRUCTURES				
Part 1	CONSTRUCTION ON LAND				
1.2.	Concrete				
1.2.1.	In situ (excluding formwork measured in Section 2.O.	2250	m^3	58.00	130.5
	A. In situ Lytag Concrete	1555	m^3	85.00	132.1
1.2.2.	Pre cast (including reinforcement formwork and assembly, excluding pre-stressing measured in Section 4.0)	216	m^3	180.00	38.8
	Formwork				
1.2.3.	Permanent void former	-	m^3		
1.2.4.	Difficult (e.g. in confined locations/ low re-use).	1300	m^2	17.90	23.2
1.2.5.	Curved to one radius in one place (e.g. cylinder)	8700	m^2	9.40	81.7
1.2.6.	Other curved (e.g. dome)	-	m^2		
1.2.7.	Propped (e.g. soffit)	-	m^2		
1.2.8.	General (all other situations)	-	m^2		
1.2.9.	Not used.				
	Reinforcement				
1.2.10.	For in situ concrete, bar diameter not exceeding 12mm	-	t		
1.2.11.	Ditto bar diameter exceeding 12mm	448	t	422.00	189.0
	Prestressing				
1.2.12.	Pre-tensioning for pre-cast concrete units	-	t		
1.2.13.	Post-tensioning wire system, tendon length less than 10m	-	t		
1.2.14.	Ditto, tendon length from 10 to 50m	146	t	1650.00	240.9
1.2.15.	Ditto, tendon length greater than 50m		t		
1.2.16.	Post tensioning bar system	-	t	1750.00	17.5
				854.07	

c/f to Collection Page 6

BILL OF QUANTITIES - PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	RATE	£
	COST CENTRE 1.2. CONCRETE STRUCTURES, contd.				
Part 3	**DEVICE SPECIFIC ITEMS**				
3.1.	**Other permanent work items**				
	(To be completed by Development Team)				
1.2.32					
	A. Supply and Install Epoxy joints between Spines			SUM	15000
	B. Assemble spines and ducks and supply of bearings, Hookes Joint, Shroud and Seal.			SUM	300702
	C. Fix ballast Tanks in spines (Materials in Concrete Quantities)			SUM	3664
	D. Walkways Airlocks and Miscellaneous Steelwork	13	t	1125.00	14625
					333991

c/f to Collection Page 6

BILL OF QUANTITIES - PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	£ MILLION RATE	
	COST CENTRE 1.2. CONCRETE STRUCTURES				
Part 3	**DEVICE SPECIFIC ITEMS contd.**				
3.2.	**Special temporary works items (excluding cost of facility in Cost Centre 1.1.)**				
	(To be completed by Development Team)				
	A. Establish Casting Area				7.9
	B. Establish Gantries and lifting gear				4.7
	C. Operation and Maintenance of Casting Area and Gantries	9	YR	2.20	19.8
	c/f to Collection Page 6 : 864 = £37500 PER DUCK				32.4

BILL OF QUANTITIES - PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	RATE	
	COST CENTRE 1.2. CONCRETE STRUCTURES COLLECTION			£000	
Part 1	General Items Construction on Land			854	07
Part 2	General Items Construction over Water				
Part 3	Device Specific Items				
3.1.	Other Permanent Work			333	99
3.2.	Special Temporary Work			37	50
	Sub Total			1225	56
Part 4	ADD 5% of Sub Total for Miscellaneous Fittings.			61	28
				1286	84
	c/f to SUMMARY				

BILL OF QUANTITIES - PART 1
CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	£ MILLION RATE	
	COST CENTRE 1.5 FLOAT OUT.				
	NOTE: Costs here are for float out and tow to deep water storage NEARBY. Costs for towing to a holding area prior to final positioning are included in INSTALLATION SECTION				
1.5.1.	Float out Devices (i.e. costs for operating the facility type selected by the Development Team Shiplift	18	Year	2.80	50.
1.5.2.	Allowance for towing Device Units to temporary deep water storage area nearby (provision of tugs)	1296	No	.008	10.
	other attendant vessels		Sum	.16	0.
	Sub Total Annual Charges				60.
1.5.3.	Provision of moorings at temporary storage area nearby	434	No	.03	13.
	TOTAL at ONE FACILITY				73.
	Estimated number of Devices constructed at ONE facility 864				
	Hence proportion of Float Out cost per Device Unit	.116%	=	£85596 PER DUCK	
	c/f to SUMMARY				

BILL OF QUANTITIES - PART 1

CIVIL CONSTRUCTION OF DEVICES

ITEM	DESCRIPTION	QUANTITY	UNIT	RATE	£
COST CENTRE	SUMMARY WITH STANDARD RATES PER DEVICE UNIT				£000
1.1.	Provision of Construction Facility				131.718
1.2.	Concrete Structures				1286.840
1.3.	Steel Structures				-
1.4.	Install Machinery (EXCLUDES connection and commissioning)				
1.5.	Float Out to temporary storage area (EXCLUDES tow to holding area)				85.596
	TOTAL FOR ONE UNIT				1504.154
	Total number of Devices Required 864				
					1299589.06
	Hence TOTAL CIVIL CONSTRUCTION COST				
	MAINTENANCE FACILITY				
A	Construction of basic facility				35000.00
B	Operation of basic facility				200000.00
C	Additional costs for lifting devices out of the water : Construction				10000.00
D	Ditto : Operation				23000.00
	Sub Total Maintenance Facility				268000.000
	TOTAL : 864 = 310185 PER DUCK				

MAKE UP OF ALTERNATIVE RATES INSTALLATION

Item	Method & Resources	Quantity	Unit	Unit Rates			Totals			Prime Cost	Additions	Total Cost	Alternative Rate
				Lab.	Mat.	Plant	Labour	Material	Plant		35 %		
A	B	C	D	E	F	G	H = CxE	J = CxF	K = CxG	L = H+J+K	M = L	N = L+M	P = N ÷ C
	TOW OUT 40 lengths of 1 KM												
	4 No. TUGS @ £5000 = 20000/ DAY												
	ESCORT VESSEL 3000												
	SUPPLY VESSEL 3000												
	DIVE SUPPORT VESSEL 1800												
	27800/DAY												
	X5 DAYS	40								5560000	1946000	7506000	
	MAKE JOINTS AT SEA								÷ 40	= £187650	PER TOW		
	DIVERS + SUPPORT VESSEL 18000/DAY												
	TUG 5000												
	SUPPLY BOAT 3000												
	26000 PER DAY												
	ALLOW 10 DAY X 26000	32								8320000	2912000	11232000	
	WINCHES 3 @ 24000									72000	25200	97200	
	SPARES & MAINTENANCE WINCHES									18000	6300	24300	
	FLANGE CLAMPS (2 NO./JT)												
	£60000 X 2	32								3840000	1344000	5184000	
												16537500	
									÷32 JOINTS		£516796 PER JT		

MAKE UP OF ALTERNATIVE RATES

MAINTENANCE AND OPERATION
OF CASTING AREA

Item	Method & Resources	Quantity	Unit	Unit Rates Lab.	Mat.	Plant	Totals Labour	Material	Plant	Prime Cost	Additions 35%	Total Cost	
A	B	C	D	E	F	G	H = CxE	J = CxF	K = CxG	L = H+J+K	M = L	N = L+M	
	POWER - GENERATOR £1300/MTH x 12 MTH	6	No						93600	93600	32760		126360
	SPARES 3% x 8000 x 93600 / 1000								22464	22464	7862		30326
	FUEL 6 x 47WK x 168HR x 18.15LT x 60% x .15								77389	77389	27086		104475
	LUB OIL SAY								20000	20000	7000		27000
	GANTRY SPARES SAY 10% x 1405000								140500	140500	49175		189675
	TRANSFER UNIT SPARES SAY								125000	125000	43750		168750
	LIFT GEAR SPARES SAY								100000	100000	35000		135000
	LABOUR 6 MACHINES x 4 MEN x 24HR x 7 DAY x 47WK £5.50						1042272			1042272	364795		1407067
													2188653
										X 9 YEARS			19697877
										SAY £19.7 MILLION			

MAKE UP OF ALTERNATIVE RATES CASTING AREA (DUCKS & SPINES)

Item	Method & Resources	Quantity	Unit	Unit Rates Lab.	Unit Rates Mat.	Unit Rates Plant	Totals Labour	Totals Material	Totals Plant	Prime Cost	Additions 35 %	Total Cost	Total Cost £000	
A	B	C	D	E	F	G	H = CxE	J = CxF	K = CxG	L = H+J+K	M = L	N = L+M		
	CONCRETE BEAMS AND HARDSTANDINGS	71000	m^3	STANDARD RATE £58.0/m^3 Incl. Additions									4118)	
	REINFORCEMENT	8280	t	STANDARD RATE £422/t									3494)	7.9
	FORMWORK	36000	m^2	STANDARD RATE £8.5/m^2									306)	
	RAILS ON BEAMS	525	t			£285	+ £370			655	230	885	465)	
	PLATES ON BEAMS FOR SKATES	550	t			£220	+ £370			590	207	797	438)	
	GANTRIES 4 NO. @ 250t	1000	t	STANDARD RATE £1405									1405)	
	LIFTING GEAR	4	No							200000	70000	270000	1080)	4.71
	ROLLOVER UNIT	2	No					SAY		150000	52500	202500	405)	
	DUCK TRANSFER PLATFORM							SAY		150000	52500	202500	203)	
	SPINE TRANSFER PLATFORM							SAY		400000	140000	540000	540)	
	SKATES	500	No							250	88	338	169)	
													12623	
											SAY	£12.62m		

MAKE UP OF ALTERNATIVE RATES

ASSEMBLY OF UNITS IN STORAGE AREA
CONNECT SPINES TO DUCKS

Item	Method & Resources	Quantity	Unit	Unit Rates Lab.	Unit Rates Mat.	Unit Rates Plant	Totals Labour	Totals Material	Totals Plant	Prime Cost	Additions 35%	Total Cost	Alternative Rate
A	B	C	D	E	F	G	H = CxE	J = CxF	K = CxG	L = H+J+K	M = L	N = L+M	P = N ÷ C
	SUPPLY BEARING BETWEEN SPINE AND DUCK	864	No		10000			8640000		8640000	3024000	11664000	
	SUPPLY SHROUD (432 x 22t)	9504	t		1405			STANDARD RATE		13353120	-	13353120	
	SUPPLY SHROUD SEAL	432	No		28500			12312000		12312000	4309200	16621200	
	SUPPLY WHEELS (432 x 30T)	12960	t		1405			STANDARD RATE		18208800	-	18208800	
	SUPPLY HOOKES JOINT (432 x 220t)	95040	t		1405			STANDARD RATE		133531200	-	133531200	
	SUPPLY BEARINGS FOR HOOKES JT	432	No		16000			6912000		6912000	2419200	9331200	
	BARGE & EQUIPMENT ALLOW									1500000	525000	2025000	
	OPERATION COST SAY £5000 PER WK x 47WK x 9YR									2115000	740250	2855250	
	LABOUR												
	2 TEAM OF 30 MEN x 168 HR x 47 WK x 9 YR x £5.50 PER HR									23451120	8207892	31659012	
	DIVERS												
	4 No x 3 SHIFT x 47 WK x 9 YR x £3000/WK									15228000	5329800	20557800	
												259806582	
								÷ 864 UNITS	=	£300702 PER DUCK			

MAKE UP OF ALTERNATIVE RATES PLACE BALLAST TANKS IN SPINES

Item	Method & Resources	Quantity	Unit	Unit Rates Lab.	Unit Rates Mat.	Unit Rates Plant	Totals Labour	Totals Material	Totals Plant	Prime Cost	Additions 35 %	Total Cost	Alternative Rate
A	B	C	D	E	F	G	H = CxE	J = CxF	K = CxG	L = H+J+K	M = L	N = L+M	P = N ÷ C
	6 MEN x 168 HRS x 47 WKS x 9YR £5.50									2345112	820789	3165901	
								÷864 = £3664 PER DUCK					

8.1.5 Base Fleet Costs

1. Vessels Daily Operating Costs

Ocean Going Tug 8000 IHP 96T Bolland Pull (including Navigation Package)	£5,000/day
Ocean Going Tug 6000 IHP 60T Bolland Pull	£4,000/day
Dive Support Vessel (including Diving Team Crew, Fuel Victuals, etc.)	£1,800/day
Escort/Navigation Vessel	£3,000/day
Supply Boat	£3,000/day

2. Tow Out to Installation Site

The Device will be towed from the construction site (Hunterston) to the installation site approx. 40 Km off the west coast of the Outer Hebrides.

The Towing Configuration will be:-

2 x Ocean Going Tugs 96T Bollard Pull on main tow line

1 x Ocean Going Tug 60T Bollard Pull on stern pennant line

1 x Escort Navigation Vessel

Assume 5 days for towing

2 x Ocean Going Tug	£50,000
1 x Ocean Tug	£20,000
1 x Escort/Navigation Vessel	£15,000
Towing Ancillaries - (Buoyancy Towing Hawser, etc.)	£20,000

N.B. Maximum Towing Speed 2.1/2.3 Knotts

3. Installation

The device will be installed using a pull in technique whereby one device is installed on its moorings and the next device is pulled in to mate up with hydraulic latches around the circumference of the mating flange.

8.1.5 Base Fleet Costs(Continued)

It is estimated that each installation will take approximately 5 days inclusive of installing the moorings.

1 D.S.V. x 5 days	£90,000
1 Tug x 5 days	£25,000
Pull-in Winches	£30,000 each
(Pull-in winches will be mounted on the devices. The winches will be removed and re-used.)	
Hydraulic latches (3 per flange)	£20,000 each

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8.1.6 Installation

			£000	£000
Tow from storage area to location at sea in 1 Km length	40	No	187.65	7506.00
Join into 5 Km lenths	32	JTS	516.80	16537.60
Moorings	864	No	30.00	25920.00
				49963.60

÷ 864 Units = £57830 Per Duck

SUMMARY (PER DUCK)

Construction Facility (Operation and capital cost allocation)	131718.00
Concrete Structures (including Spine Jointing)	1,286,840.00
Supply and install machinery (in test facility and spares)	715333.00
Float Out (including Ship Lifts and Temp. moorings)	85596.00
Maintenance Facility (establishment and operation)	310185.00
Installation at Sea (including permanent moorings)	57830.00
	2587502.00
Electrical Generation Transmission and Installation	366000.00
	£2953502.00

X 864 total for scheme $£2551.8 \times 10^6$

SECTION 9

APPENDICES

APPENDIX A

LIST OF COMPANIES PARTICIPATING

We gratefully acknowledge the interest and support of the following organisations, among others:-

Ampohm Ltd.	Redditch
Andre Rubber Company Limited	London
Avon Rubber Company Limited	Melksham
B.P. Oil Limited	London, S.W.1
Bradford Cylinders Limited	Bradford
British Oxygen (Cryoplants) Limited	London
British Ropes Limited	Doncaster
British Steel Corporation	Glasgow
Bryant Symons Ltd.	Tottenham
CEGB Research Limited	Leatherhead
Dunlop Limited	Birmingham
Dunlop Limited	Liverpool
Easams Ltd.	Frimley
Firth Brown Limited	Sheffield
Glacier Metal Company Limited	Alperton
W.H. Gore Limited	Dunfermline
Harland & Wolff Limited	Belfast
Heriot Watt University	Edinburgh
Inco Europe Limited	London
F.H. Lloyd Limited	Wednesbury
McTaggart Scott Limited	Edinburgh
Markham & Company Limited	Chesterfield
Markon Engineering Limited	Oakham
Monro & Miller Limited	Edinburgh
Mullard Limited	London
Pirelli Cables Limited	Eastleigh
Plastic Coatings Limited	Guildford
Poclain Limited	Marlow
Redman Broughton Limited	Birmingham

Appendix A Continued

Redpath Engineering Limited	Glengarnock
Reinforced Plastic Structures Limited	Lancing
Rexroth Limited	St. Neots
Smiths Industries Limited	Rugby
Sunderland Polytechnic	Sunderland
Sunter/ITM Limited	London
Towler Hydraulics Limited	Rodley
Uniroyal Limited	Glasgow
Von Roll	Zurich
Whessoe Limited	Darlington
Y-ARD Limited	Glasgow

AMPOHM LIMITED

UNIT 8, ENFIELD INDUSTRIAL ESTATE,
REDDITCH, WORCESTERSHIRE, B97 6BG, ENGLAND.

TEL: REDDITCH (0527) 65896
CABLES: AMPOHM REDDITCH

Your Ref
Our Ref EJO/SN/2

16 October 1981

P B Williams Esq
John Laing Design Associates Ltd
PO Box 31
Page Street
Millhill
LONDON
NW7 2ER

Dear Mr Williams

I confirm our telephone conversation of 15 October referring to the magnet elements in the University of Edinburgh Wave Power Project.

As promised, I enclose copies of the correspondence to date and will arrange to provide you with copies of further correspondence on this item. We will also provide you with copies of letters received from the University.

We assure you of our keen interest in this project.

Yours sincerely
for AMPOHM LIMITED

E J Ogle

E J OGLE
Managing Director

ANDRÉ RUBBER COMPANY LIMITED
KINGSTON-BY-PASS SURBITON SURREY KT6 7LY ENGLAND TELEPHONE 01-397 5272 TELEX 25636 ANSIL G TELEGRAMS ANDRÉ SURBITON

John Laing Design Associates Limited,
Box 31,
Page Street,
Millhill,
LONDON NW7 2ER.

For the attention of
Mr. P.B. Williams.

10 November 1981

EGJ/jpn

Dear Sir,

Salter/Laing Wave Energy 'Nodding Ducks'

We confirm having completed our feasibility study and budget estimates, which are as follows:-

Description	Quantity	Price Each
Gimble bearing	2000	£ 4000 each
Rod end half bush	16000	£ 1000 each
Rolling seal AR 28266 - Tyrecord	500	£12000 each
- Kevlar		£16500 each
Rolling seal AR 28267 - Tyrecord	4000	£ 1500 each
- Kevlar		£ 4500 each

The above prices are budget only but do take account of the full amortization of the tooling required.

The volumes required are such that our present manufacturing capacity would almost certainly not be able to produce at a satisfactory rate and it is likely that two further presses would have to be considered. In my opinion the purchase of these presses, including ancillary equipment, installation and commissioning, would cost approximately £500,000.

This cost has in no way been incorporated into our budget prices, and to obtain a true figure in respect of volume rate and unit cost one should amortize this cost across the gimble and rod end half bush.

One further item of cost to be considered is that it would be necessary for us

(continued)

to produce models, at least of the rod end half bush and gimble, in order to fully evaluate its performance and carry out simulated life cycle tests.

The mould tools and test equipment costs of this work would be the subject of a development contract which we would budget at approximately £150,000.

In summation, we would confirm that the items listed are within our current manufacturing technology although the sizes and production volumes are outside our present experience - but we in no way see that as a major problem.

Assuring you of our closest co-operation at all times,

We remain,

Yours faithfully,
ANDRE RUBBER COMPANY LIMITED

pp. E.G. James JPM.

E.G. James,
Director & Business Manager - Engineering.

BP Oil Limited

BP House Victoria Street London SW1E 5NJ

Your Ref

Our ref LUBS/TS

Laing Project Assessment &
Development Limited
P O Box 31
Page Street
Mill Hill
London NW7 2ER

Tel 01-821 2804

Date 26 October 1981

For the attention of Mr P B Williams
Alternative Energy Group

Dear Mr Williams

With reference to our recent telephone conversation concerning the hydraulic fluids for the Salter "ducks", I have reviewed my notes and the information provided by our Research Centre colleagues following our meeting in November 1979.

On the technical aspects, there are essentially no changes to our original recommendations; thus we still feel that the lubrication requirements of the sealed pods would best be met by a special narrow cut low viscosity mineral oil, and those of the linkages by a conventional hydraulic oil, possibly with enhanced corrosion inhibition.

As for the commercial considerations, whilst I am in no position to quote actual prices, I suggest that if you apply a factor of 1.5 to the original November 1979 prices, this should provide sufficiently accurate figures for the purposes of your calculations. I should emphasise that this does not constitute a quotation and I would also reiterate our earlier comments that the original prices were themselves best estimates.

May I take this opportunity to wish you success in the re-submission of your project. Please contact me again should you require further information or assistance.

Yours sincerely

Ray Cecil

DR R CECIL

BRADFORD CYLINDERS LIMITED

QUOTATION

Soho Works,
Allerton Road,
Bradford, Yorkshire
England. BD8. OBA.

Telephone. 0274. 495611.
Telex. 517526.
Cables:-
Metallic Bradford.

John Laing Design,
P.O.Box 31,
Page Street,
Mill Hill,
London.
NW7 2ER.

Your ref.

Our ref AVA/NR/EB0514

Date. 6th November 1981

For the attention of Mr.P.B.Williams.

Dear Sirs, Confirmation of our telex 23.10.81

Your Enquiry Ref:& Telephone conversation Dated: 6.11.81

We thank you for the above enquiry and have pleasure in quoting as follows:-

WAVE ENERGY GROUP - SALTER DUCKS

8 - Off 610mm bore x 250mm dia rod x 2200mm stroke Bradford Hydraulic cylinders, (Nodding Ducks) of the double acting class. Rear flange mounting. Piston rod terminating in a male screwed rod end. Cylinder suitable for a maximum working pressure of 6000 p.s.i. The hydraulic cylinders would have a service life expectancy of 25 years when operating in a sealed chamber pressurised to 1.5 atmospheres with inert gas.

PRICE:-£22,000.00 each.

continued............

This Quotation Does Not Include V.A.T.

PLEASE NOTE: To cover the cost of documentation a surcharge of £10.00 will be added to orders of less than £75.00 in value.

DELIVERY: weeks from receipt of your order subject to confirmation at that time.

TERMS: 30 days nett from date of despatch (packing and carriage charged extra).

We trust that this offer is acceptable and look forward to receiving your order instructions in the near future.

Yours faithfully
for and on behalf of
BRADFORD CYLINDERS LTD.

~~Sales Office Manager~~

CONDITIONS OF SALE OVERLEAF

Under full production circumstances where large quantities were required, say 10,000 units in total at a theretical production rate of 10 cylinders per week, a significant reduction in manufacturing costs could be acheived and under such circumstances our budget price would be £15,000.00 per cylinder.

Terms and conditions to be agreed.

Validity 120 days.

Yours faithfully,
BRADFORD CYLINDERS LTD.

A.V.Attwood

A.V.Attwood.
Commercial Manager.

British Steel Corporation
Tubes Division

Corby Works, Corby, Northamptonshire
Telephone Corby (053 66) 2121 Telex 34418 Telegrams Tubemakers Corby Northants Telex
VAT Reg. No. 238 7122 60

PLEASE REPLY TO CORBY MR. D. T. TOWNSEND, TEL. 05366-64174

JOHN LAING DESIGN ASSOCIATES
ATTN OF MR P E WILLIAMS
BOX 31
PAGE STREET
MILL HILL
LONDON NW17 2ER

REF HQ 133559/DTT/81

2 NOV 81

QUOTATION

DEAR SIRS

YOUR ENQUIRY REF OF 28 OCT 81

WE THANK YOU FOR YOUR ENQUIRY AND HAVE PLEASURE IN QUOTING-

HOT FINISHED SEAMLESS STEEL LINEPIPE TO API 5L GRADE B 31ST EDITION 1980. ENDS PLAIN AND CUT SQUARE.

ITEM 1

360000 METRE(S)
60.3MM.OD 7.48KG/M 5.54MM.TK.
RANDOM LENGTHS 5M/8M

AT £460.21 PER 100 METRES
SUM £1,656,756

ALTERNATIVELY

ELECTRIC RESISTANCE WELDED STEEL LINEPIPE TO API 5L GRADE B 31ST EDITION 1980. ENDS PLAIN AND CUT SQUARE.

ITEM 2

360000 METRE(S)
60.3MM.OD 7.48KG/M 5.54MM.TK.
RANDOM LENGTHS 5M/8M

AT £438.3 PER 100 METRES
SUM £1,577,880

ALL ORDERS EX WAREHOUSE ARE DELIVERED CARRIAGE PAID ON THE MAINLAND OF GREAT BRITAIN.

ORDERS EX WORKS ARE DELIVERED CARRIAGE PAID, ON THE MAINLAND OF GREAT BRITAIN AND NORTHERN IRELAND SUBJECT TO A MINIMUM QUANTITY OF 1 TONNE OF TUBE OR £200 NET VALUE OF OTHER GOODS.

PRICES WILL BE THOSE RULING AT DATE OF DESPATCH.

CONTINUED ON SHEET 2

British Steel Corporation
Tubes Division

Corby Works, Corby, Northamptonshire
Telephone Corby (053 66) 2121 Telex 34418 Telegrams Tubemakers Corby Northants Telex
VAT Reg. No. 238 7122 60

PLEASE REPLY TO CORBY MR. D. T. TOWNSEND, TEL. 05366-64174

JOHN LAING DESIGN ASSOCIATES
ATTN OF MR P B WILLIAMS REF HQ 133559/DTT/81

2 NOV 81

SHEET 2

AS TUBES ARE BEING SUPPLIED IN RANDOM LENGTHS THE TOTAL VALUES STATED ARE APPROXIMATE.

TERMS-TO BE ARRANGED

NOTWITHSTANDING ANYTHING CONTAINED IN YOUR GENERAL CONDITIONS OF CONTRACT, OUR CONDITIONS OF CONTRACT, AS PRINTED ON THE REVERSE SIDE OF THIS LETTER, SHALL APPLY AND TAKE PRECEDENCE OVER ALL OTHER CONDITIONS.

IF YOU HAVE ANY QUERIES CONCERNING THIS QUOTATION PLEASE CONTACT MR. D. T. TOWNSEND, TEL. 05366-64174 .

YOURS FAITHFULLY
BRITISH STEEL CORPORATION, TUBES DIVISION.
QUALITY TUBE SALES INTERNATIONAL, CORBY

Cryoplants Limited

Angel Road London N18 3BW England
Telephone 01-803 1300 Telex 263247
Telegrams Cryoplants London N18

13 August 1981

Mr. P.B. Williams
Project Assessment & Development
Page Street
LONDON NW7 2ER

Dear Mr. Williams,

<u>Slater Duck Wave Energy Device</u>

I write in response to your letter of 20th July regarding the feasibility of producing welds in vessels requiring a high degree of leak tightness.

We see no reason why individual welds satisfying a leak rate giving a pressure rise of less than 15 torr in 23 years cannot be produced in 6 mm material. It would in our view be essential that the procedure for such welds be developed using appropriate pressure vessel code standards and that the welds be performed by operators qualified to similar standards. Some thought should also be given to the levels of other forms of NDT which should be applied to complement the leak detection procedures.

I should mention that the leak rate you are looking for is greater than those we currently specify for our vacuum insulated vessels. On the other hand your volume is much greater and the two factors therefore tend to cancel each other out. What is clear is that to ensure the integrity of the vessels will require the application of modern mass spectrometry leak detection equipment such as are in use here at Cryoplants. I may not have mentioned previously that one of the BOC Group Companies, Edwards High Vacuum, are manufacturers of this type of equipment and also the vacuum pumps that would be required.

Yours sincerely,

N. van Tromp

N. VAN TROMP
Manager - Distribution Equipment

NVT/jr

Report on telephone conversation with Cryoplant Limited
(A British Oxygen Company)

Contact: Allen Hale - Production Engineer

The discussion centred on vacuum jacket vessels (cryo flasks) and the problems they have in maintaining a low leakage rate.

In production they find that a leak of 1×10^{-2} Torr litre/sec is detectable with soapy water.

A leak of 1×10^{-3} Tl/sec is detectable by Halogen test (using fluon gas and a gas detector).

A leak of 1×10^{-8} or 10^{-9} Tl/sec is detectable by mass spectrometer methods which they have been using for some 5 years.

Their standard is a leak rate of 1×10^{-6} Tl/sec. This, of course, is with air in contact with the outside; not sea water as in our case.

They seldom have trouble at ordinary seam welds - most of their faults appear at pipe junctions with jackets or with through branches or nozzles.

Their production units are guaranteed for 5 years but often last 10 or 15.

They see little difficulty with our requirements and are anxious to quote for vessels for the 'ducks'.

As a comparison - consider the duck pressure at 5 Torr rising to 10 Torr over 25 years. It could have a leak rate of:

$$\frac{490{,}000 \text{ litres} \times 5 \text{ Torr}}{365 \text{ days} \times 24 \text{ hrs} \times 3600 \text{ secs} \times 25 \text{ years}}$$

$$= \frac{2.45 \times 10^6}{788.4 \times 10^6}$$

$$= \underline{\underline{3.10 \times 10^{-3}}} \text{ Torr litre/sec.}$$

or 3000 times greater than the leak rate that BOC are currently setting as their production standard!

P.B. Williams
15.8.80
John Laing

Circulation:	Mr. F.S. Nundy	-	J.L. Ltd.
	Mr. S.H. Salter	-	Edinburgh University
	Mr. M.S. Cloke	-	ETSU Harwell

DUNLOP LIMITED

DUNLOP BELTING DIVISION
PO Box No 7 Liverpool L24 1UY
Phone 051-486 4551 Telex 627080
Cables Dunlobelt Liverpool

Our ref: BJE/PB

5th November, 1981

John Laing Design Consultants,
P.O. Box 31,
Page Street,
Mill Hill,
NW7 2ER

For the attention of: Mr. P.B. Williams

Dear Sir,

Reference: Wave Energy Device - Nodding Ducks

Thank you for the courtesy extended on the occasion of my visit to your offices on Wednesday October 14th, 1981.

I apologise for the delay in submitting our recommendations for 'Magnetic Belts' for use with the Duck Wave Energy Device but the report submitted following my visit is being studied by the production and R & D sections of our Division and I hope to have their replies to hand shortly.

Three points that would require early investigation are a) the effect of prolonged immersion in sea water on the rubber compounds comprising the belt b) the alignment of the magnetic lines to prevent the polar lines erasing and creating an attractive force instead of an opposing force c) the problems that could be met when trying to assemble the magnetic belts onto the ducks and overcoming the opposing forces.

If you have any further ideas on use of the proposed magnetic belts, please keep me informed, meanwhile, I remain,

yours faithfully,
for DUNLOP LIMITED
Belting Division

B.J. Elliott

B.J. ELLIOTT
ORIGINAL EQUIPMENT REPRESENTATIVE

F. H. LLOYD & CO. LIMITED
Registered Number 32352 England

Registered Office :
P.O. Box 5,
James Bridge Steel Works,
Wednesbury,
West Midlands WS10 9SD
England
Telephone 021 526 3121
Telex 337538

Date 9th November 1981. Your Ref Our Ref JSS/GM

Mr. P.B. Williams,
John Laing Design Associates,
Alternative Energy Group,
P.O. Box 31,
Page Street,
Millhill,
London NW7 2ER.

Dear Mr. Williams,

Further to our recent conversations regarding the "Edinburgh Duck Project" I have pleasure in confirming that our interest remains keen.

We are able to assure you that we have the facility to manufacture all the items that have so far been discussed, and in all probability other items which have not yet been considered.

With regard to price levels, as you are aware a great variance can arise in the same casting depending on the degree of non-destructive testing called for, but taking a good commercial quality as a standard and a carbon steel as the material, the price range would be of the order of £900 per tonne for unmachined castings weighing 2 tonnes or so (providing there is no requirement for complex core making and setting) up to £1400 per tonne for machined castings, with approximately £1200 per tonne for machined castinngs where the machining requirement is simple boring and facing operations only.

We would point out that the foregoing remarks concerning price are all of necessity, very general at this stage and that each item would need to be considered on its merit. We would stress, however, our willingness to discuss any or all aspects of the project at any time and would confirm that the high volumes envisaged may well enable us to negotiate special prices.

It may be of interest to you that we are currently developing new technologies for the production of node castings for application in Off-Shore Oil Rigs. These improvements are aimed at enhancing the cast properties particularly so far as notch sensitivity is concerned. Results to date indicate that the cast materials have better impact values than wrought and do not suffer the directional variances associated with wrought materials. As a result one can expect better fatigue resistance in all directions.

We would add that our machining capabilities are considerable and of course not confied to machining castings of our own manufacture but available for any components or structures from other sources.

Yours sincerely,

J.S. Smith.

Firth Brown Limited
Atlas Works . Savile Street . Sheffield S4 7US

Telephone 0742 20081
Cables Firth Sheffield
Telex 54279

Registered Office:
Atlas Works,
Sheffield S4 7US

Special Steels . Forgings . Forged Steel Rolls
Nickel Base Alloys . Rolled Bars, Billets & Rings

A member of the Steel Division of Johnson & Firth Brown Limited

Our Ref: PJD/KAH/79/81

17 April 1981

Mr P B Williams
Wave Energy Group
Project Development &
Assessment Ltd
Page Street
LONDON NW7 2BR

Dear Mr Williams

Prestressing Rings, Quotation References V7990 and V7991

Following our meeting on the 17th March I have had estimates prepared for G110 maraging steel pre-stressing rings both for the full scale device and for the ⅓rd model. If G125 were to be considered the prices would need increasing by 4% :-

⅓ Scale Model - 2 alternatively 4 rings solution treated and rough machined

to 20¼" OD x 12¼" ID x 4" wide

to finish 20" OD x 12½" ID x 3¾" wide

Price £1650 each. Delivery 18/20 working weeks from order.

Full Scale Device - This has proved more difficult to estimate. Our proposals involve the manufacture of a 26" diameter ingot which would be upset forged and punched prior to expanding to the proposed forged size of 57⅜" OD x 37⅛" ID x 12½" thick. This is fairly common practice with conventional steels and our experience with maraging steel in smaller lumps is sufficient for certain assumptions to be made in arriving at a cost.

For the first order our price would be around £23,500 each and delivery 20/25 working weeks. With increasing experience, I would expect to be able to get our price down to around £20,000 each.

We could leave it for further discussion on the quantity to be ordered in the first place. I would need a minimum of two rings for the necessary development work to be carried out, but in the event you may wish to proceed at a higher rate.

.../2

The rings supplied to you would be in the solution treated condition and rough machined to 3 mm a side above finished size i.e.,

1406 OD x 994 ID x 266 thick

A test ring would be taken from the bore at one end and we have included for immersion ultrasonic test as well.

I trust these notes will be of interest and look forward to your news with considerable anticipation.

Yours sincerely

Peter Dickenson

P.J.Dickenson

Mullard

Mullard Limited Mullard House Torrington Place London WC1E 7HD

Ref: ID/RAD/SRB

Mr. P. B. Williams, Box 31,
John Laing Design Associates Limited,
Page Street,
Mill Hill,
London,
NW7 2ER

23rd October 1981

Dear Mr. Williams,

WAVE ENERGY PROJECT

Thank you for your hospitality when I visited you last month.

First, we would like to assure you and your organisation that Mullard Limited is the electronic components company operating in the United Kingdom, but within the multi-national N.V. Philips Gloeilampenfabrieken concern. We have very considerable resources to produce magnetic materials of both the permanent and non-permanent types.

In the case of permanent magnets, we are able to offer products in a choice of ceramic, both cast and certain sintered metal and also plastic bonded (matrix) materials. Currently we manufacture products for in-house and third party customers for use in the Consumer, Domestic Appliance, Professional and Automobile industries.

Last year there was contact between Mr. S.H. Salter of University Of Edinburgh and Mr. C.P. Southworth from an application laboratory in a Mullard magnetic materials factory. As a result of some preliminary work, a letter dated 6th October 1980 was written to Mr. Salter by Mr. Southworth.

From this work, it appears that forces approaching the desired bearing force can be achieved using ceramic magnets. Improvement in performance of the bearing, toward the value of force indicated theoretically, may be possible by adjusting the dimension of the magnet specified. It is envisaged that only small improvements will be made to the properties of the basic ceramic material over the next few years.

Significantly higher values of Brem can be obtained by using metal magnets. However, for this application, with a hostile salt water environment, ceramic is probably the better choice. The ceramic is normally inert in almost all environments and is usually ground using diamond grinding wheels. When operated over a long time in sea water, I would envisage that some wear will occur on the magnet, due to the continual washing by abrasive (sand) bearing salt water.

..../continued over

Telephone 01-580 6633 Telex 264341 Cables Mullectron London WC1E 7HD

The initial feasability work was based on a block of ceramic material 3 in long, 1.75 in wide and in multiples of 0.375 in thick. The 1.75 dimension was envisaged parallel to the bearing axis whilst the thickness, and magnetic axis, of 0.375 was radial to the bearing. One half of the total bearing surface was assumed to be covered by magnetic material.

The guide price for finished blocks of this general type would be in the order of £1200 per ton for a barium ferrite and £1500 per ton for a strontium ferrite material. This statement assumes a fairly high volume of production and 1981 prices.

We trust that this letter contains the broad outline you requested concerning our capability and the feasability of the magnetic bearing. We look forward to hearing from you in due course, but, in the meantime, please do not hesitate to contact the writer if you have any queries.

Yours sincerely,

R.A. Davey

R. A. Davey
INDUSTRIAL DIVISION

Poclain Hydraulics Limited
Fieldhouse Lane,
Marlow, Bucks. SL7 1LW.
Telephone: (06284) 74616
Telex: 848019 POCHYD

Poclain HYDRAULICS

Mr. P. B. Williams
Project Assessment & Development Ltd.
(John Laing Co.)
Page Street
Mill Hill
London NW7 2ER

ASE/GMG/Q0970

23rd July, 1981

Dear Mr. Williams,

Re: Wave Energy Project

Further to our letter dated 29th June 1981, we have pleasure in enclosing a drawing and quotation for the cylinders generally as originally discussed. You will see that the sizes vary slightly to suit preferred bore/rod dimensions.

Cylinders to 800 mm bore can be manufactured and as a price guide you should add 20% for each 100 mm bore increase.

It would appear that our first limitation on maximum size is in fact our existing handling rather than machining capacity. To provide a precise costing is therefore rather difficult at this stage.

We trust you will find the enclosed adequate for your present feasibility study.

Yours sincerely,

A. S. Eldridge
U.K. Sales Manager

Poclain Hydraulics Limited
Fieldhouse Lane,
Marlow, Bucks. SL7 1LW.
Telephone: (06284) 74616
Telex: 848019 POCHYD

Poclain HYDRAULICS

QUOTATION REF: ASE/GMG/Q0970 23rd July, 1981

To: PROJECT ASSESSMENT & DEVELOPMENT LTD

ITEM	DESCRIPTION	UNIT PRICE
1	Prototype 529/250 x 2200 mm Double Acting Cylinder to our drawing/offer CR5868	£8,470.00
	5 units/week as above	£7,114.00
	10 units/week as above	£7,004.00

DELIVERY: Prototypes 20 weeks from order

PRICES: Ex Works London. Onward carriage and V.A.T. extra.

QUOTATION VALID FOR 90 DAYS

Poclain HYDRAULICS

FROM:- Mr. G. Vicentini
TO:- Mr. A.S. Eldridge
SUBJECT:- PHUK225 (Salters Ducks)
OFFER DRAWING NO. 5868.

ASE/AT

27th August 1981

FOR THE SEALING ARRANGEMENT

The aim of the study has been to avoid wear where possible. Consequently:-

- No rubber seal with high pressure.
- On bearing side, slide fit between bronze ring and the road.
- To avoid external leakage, a multi-lip seal. This seal, under low pressure, will have a very long life. It is also easily replaced.
- On piston side, where small clearances are more difficult to maintain; a case iron piston ring with skew cuts is used.

PHILOSOPHY

On large diameters (piston) the risk of siezure of bronze on steel is high. Consequently, we have added teflon nickel bands on metallic support, with a rate of dynamic compression of over 20Kg/sq. millimetre.

For the bearing we have chosen a bronze ring with a very good surface finish.

In order to avoid assemply problems, a plastic ring has been added behind the rod seal.

We have opted for a design as simple as possible to minimise machining times on the main components (body and rod) and prototype tooling costs. This solution enables us to offer short lead times by utilising the available machine tools.

MATERIAL CHOICE

To decrease the weight and ensure supplies, we have chosen to use treated tube. The rod has been sized to resist bending.

- 0 -

N.B. This design study can be modified eventually if necessary.

REDMAN BROUGHTON

a Redman Heenan Company

Redman Broughton Limited
372 Farm Street
Hockley, Birmingham B19 2UD
England
Telephone: 021-554 9373
Telex: 338242 Grams: Quipfeed B'ham.

Quotation

Messrs. John Laing Ltd.,
Page Street,
Mill Hill,
London.
NW7 2ER

Your Ref:

Our Ref: DTW/EPB/E3476/81

Date: 9th October, 1981

For the attention of Mr. P.B. Williams

Dear Sirs,

Ref.: E3476/81 - SALTER DUCK WAVE ENERGY PROJECT

With reference to the above enquiry we are now pleased to submit our budget quotation for your requirements.

PROTOTYPE HYDRAULIC CYLINDERS FOR SPIN JOINT

Size Options per Spine Joint	Budget Prices
8 off - 610 bore x 250 dia.rod x 2200 stroke	£23,750.00 each.
12 off - 520 bore x 250 dia.rod x 2200 stroke	£22,000.00 each.
16 off - 470 bore x 250 dia.rod x 2200 stroke	£20,000.00 each.

GENERAL SPECIFICATION - R.B. Drawing No.S5460

Heavy duty double acting Hydraulic Cylinders.

Rear Circular Flange mounted.

Flanged tube construction.

End caps bolted to cylinder tube assembly.

Bolt on gland housing for easy maintenance.

Chevron sealing arrangements for gland and piston.

Cylinder tube of high yield centrifugally spun steel.

Piston rod of high tensile steel with a heavy deposit of chrome plating.

Heavy duty bronze gland and piston bearings.

Port connection of the 4 - bolt flange S.A.E. type.

.../2

Annulus port piped to rear end of cylinder.

Rod to be protected by a gaiter.

Maximum working pressure: 410 bar(6,000 p.s.i.)

Maximum test pressure: 517 bar(7,500 p.s.i.)

For use on Mineral Oil.

Special marine paint finish on cylinder exterior.

Delivery of prototypes to be negotiated.

Prices for production quantities will be advised as soon as they are available.

The quantities required for a prototype machine would cause no embarrassment within our current facilities and resources, however in the event of our company being considered for all or part of the main contract we would give full consideration to the setting up of a new plant of sufficient size and equipped with the necessary facilities to undertake this project. We would add that our Company is part of a public quoted group of companies, and this matter has been discussed with our Chairman who has confirmed that Redman Broughton is a company nominated for growth within the main group, and the fullest committment would be given to such a project.

We have undertaken a basic design study concerning cylinder construction, coupled with the working environment, and enclosed will be found a copy of our drawing No. S5460 which gives details of Spine Joint Cylinders. These designs have been prepared around known available materials, but we feel that in the event of our company being considered for this contract further meetings should be convened to discuss our offer in greater detail.

Yours faithfully,
for REDMAN BROUGHTON LIMITED.

DT Whatmore

D.T. Whatmore,
Sales Engineer,
Power Energy Projects.

REDMAN BROUGHTON

a Redman Heenan Company

Redman Broughton Limited
372 Farm Street
Hockley, Birmingham B19 2UD
England
Telephone: 021-554 9373
Telex: 338242 Grams: Quipfeed B'ham.

Quotation

Your Ref:

Our Ref: MAJN/EPB /E3476/81/R1

Date: 24th November, 1981

Messrs. John Laing Limited,
Page Street,
Mill Hill,
London.
NW7 2ER

For the attention of Mr. P.B. Williams.

Dear Sirs,

Ref: SALTER DUCK WAVE ENERGY PROJECT

We would refer to the recent meeting between your Mr.P.B.Williams and our Mr. D.T.Whatmore together with Engineers from the Swiss company Von Roll, and now have pleasure in submitting herewith our budget prices for the required axial piston pumps and motors for the above project.

As you are aware, Von Roll is one of the few companies in the world able to develop the axial piston pumps and motors for this particular project, and have the necessary knowhow for new developments of this kind. One of the main aspects of their particular design is the detail given to components where the life time is one of the major points, and where the installations are in service for 24 hours duty cycles.

As you are aware, the required axial piston pumps and motors for this project have never been built before and to reach the required life time of the units development of the project would be essential, and Von Roll would be pleased to sign an exclusive contract for this purpose. Below is given a breakdown of the project costs involved, but obviously these would be subject to time factors and the development programme as is felt necessary. We are convinced that this development offers an important improvement of the axial technical data, and for this reason we cannot overlook the importance of the development programme for this project and unfortunately we are not in a position to indicate precise costs for each pump or motor.

Breakdown of costs is as follows:-

.../2

1 complete, separate test bank for 1'000 h.p. made for a 24 hours non-stop service.

4 specialists involved 1 year for this project.

Construction of 2 prototypes including material costs and individual manufacturing.

Tests of 8 months, including all the costs for electricity power, cooling system, maintenance, etc.

Costs of development	£503,000.00	BUDGET PRICE.
Price per unit, for a series of 10 units	£ 29,240.00	BUDGET PRICE.
Special toolings and auxiliaries for manufacturing	£73,100.00	BUDGET PRICE.

<u>Terms:</u>

Development	18 months	
Erection of the prototype	10 months	
Total	28 months	(could be shorter)

We trust the foregoing will in this instance be sufficient for your requirements, but should there be any doubtful points or any further information which you may require, please do not hesitate to contact us.

Yours faithfully,
for REDMAN BROUGHTON LIMITED.

M. A. J. Northwood,
<u>U.K. Sales Manager.</u>

REXROTH

G. L. REXROTH LIMITED
CROMWELL ROAD, ST. NEOTS, HUNTINGDON,
CAMBS., PE19 2ES
Telephone: Huntingdon (0480) 76041 Telex 32161
Telegrams: Rexroth St. Neots
Registered Number : 768471 England
Registered Office : As above
V.A.T. Reg. Number : 196 708 126

representing:
LOHMANN + STOLTERFOHT GmbH
Witten, West Germany

John Lange Design Associates Ltd.,
Page Street,
Mill Hill,
London. N.W.7

For the attention of Mr. P. B. Williams
Alternative Energy Group

YOUR REF:

OUR REF: 0108/99/81/DHP/JAH

DATE: 18th September 1981

Dear Sirs,

Thank you for your enquiry in respect of the Wave Energy Project. Attached is our tender for the hydraulic cylinders required.

There will be, of course, further technical discussions required and I would suggest that these take place at our Holland factory at a time convenient to yourselves.

In respect of the hydro-static bearings proposed by yourselves, the experience of Hydradyne would indicate this may not be the best choice in view of the 25 year life requirement. However, this is certainly one point for further discussion which would take place during your visit.

If there are any further questions please do not hesitate to contact the undersigned.

Yours faithfully,
G. L. REXROTH LIMITED

PP R P Murray

D. H. Piper
Manager - Cylinder Department

QUOTATION

REXROTH

G.L. REXROTH LIMITED
CROMWELL ROAD, ST. NEOTS, HUNTS. PE19 2ES
Telephone: 0480 76041 Telex: 32161
Telegrams: Rexroth St. Neots

V.A.T. Reg. No. 196 708 126

John Lang Design Associates Ltd.,
Page Street,
Mill Hill,
London N.W.7.

For the attention of Mr. P. B. Williams

YOUR REF.

OUR REF. 0108/99/81/DHP/JAH

DATE 18th September 1981

SUBJECT: YOUR ENQUIRY DATED

Dear Sir,
With reference to your enquiry we have pleasure in submitting our quotation as follows:

DELIVERY ADDRESS: To be advised

DELIVERY:

TERMS: To be negociated.

ITEM	QTY.	PART No. DESCRIPTION	PRICE EACH
	40	(8 per joint) Hydraulic cylinders, our code:- C5H650/320-2200 KZ5A Piston diameter 650 mm Rod diameter 320 mm Stroke 2200 mm Maximum working pressure 410 bar Test pressure 450 bar Construction: Bolted head and bottom. Mounting: Single blade pivot mounting with maintenance-less bearings. Seals - rubber/fabric type - extra seal on piston rod for drain connection. Piston rod material - Stainless steel type A1S1431, hard chromium plated to 50 microns. Cylinder body - DIN 1629, St 52, outside diameter 820 mm Weight per cylinder approximately 13000 kilos. With test certificate according to DIN 50049 (3.1B) EX WORKS BOXTEL, HOLLAND.	£39,300 budget each.

QUOTATION continuation

ITEM	QTY.	PART No. DESCRIPTION	PRICE

Payment Terms

All prices ex works and valid for 60 days
Carriage and packing extra
All prices shown are exclusive of VAT
All goods invoiced are subject to 15% VAT

Yours faithfully,
G. L. REXROTH LIMITED

pp R R Murray

D. H. Piper
Manager, Cylinder Department

RR74A

REXROTH

G. L. REXROTH LIMITED
LINGUAPHONE HOUSE, BEAVOR LANE,
HAMMERSMITH, LONDON W6 9AR.
Telephone: 01-741 4356/7 Telex: 28871

Registered Number: 768471 England
Registered Office:
Head Office and Works:
CROMWELL ROAD, ST. NEOTS, CAMBS. PE19 2ES
V.A.T. Reg. Number: 196 708 126

YOUR REF.

OUR REF. L/TCR/MJT

DATE 12th November 1981

Laing Design Associates Ltd.
Page Street
Mill Hill
London, NW7 2ER

Attn: Mr. P.B. Williams

Dear Mr. Williams

Further to our recent telephone conversation, I confirm that Rexroth is interested in developing with yourselves, a hydraulic gyroscope drive system. We feel that our existing range can be adapted subject to further discussions with our piston pump/motor designers.

Due to the complexity and size of the system under discussion, we trust this letter is sufficient to confirm our intentions and look forward to our meeting on November 19th.

Yours sincerely

pp T.C. RUSHTON
Area Sales Manager

REXROTH

G. L. REXROTH LIMITED

LINGUAPHONE HOUSE, BEAVOR LANE,
HAMMERSMITH, LONDON W6 9AR.
Telephone: 01-741 4356/7 Telex: 28871

Registered Number: 768471 England
Registered Office:
Head Office and Works:
CROMWELL ROAD, ST. NEOTS, CAMBS. PE19 2ES
V.A.T. Reg. Number: 196 708 126

Laing Design Associates
Alternative Energy Dept.
Page Street
Mill Hill
London, NW7 2ER

Attn: Mr. P.B. Williams

YOUR REF.

OUR REF. L/TCR/MJT

DATE 10th September 1981

Dear Mr. Williams

Edinburgh Scopa Laing Wave Energy Device

Further to the visit of our Mr. D. Piper and the Writer, of 30th June and the subsequent visit of Mr. Piper on 8th September, we would confirm our intent of working with yourselves during initial design and subsequent prototype supply, for the above project.

The hydraulic cylinder design is already at an advanced stage, which we trust is satisfactory and a full quotation, with drawings, will be forwarded in the near future.

Further to the most recent meeting and the information given regarding hydraulic motor specification, we have calculated that each motor displacement will be 3230cm^3/rev., which would be a problem, as the largest one we manufacture at present is size 2000cm^3/rev. Would it be possible to use two motors of half the displacement coupled through a gearbox? Space available would then have to be considered more closely.

The other point raised was response time of the motor from zero to full flow and would ask whether in fact, fixed displacement or variable displacement motors are envisaged, as response time tends to be associated with the latter.

Following talks with our pump/motor manufacturing works to develop the 3230cm^3/rev. motor ready for production would take no less than two years. We would welcome further talks on this matter, either at your offices, or our Head Office and await your comments.

Our offer of a visit to the Hydraudyne factory in Holland is re-stated and the offer would be extended to our piston pump/motor factories in Germany - both of which we feel sure would be very useful and informative.

We look forward to hearing your comments and in the meantime, remain,

Yours sincerely

Greadaway

pp T.C. RUSHTON
Senior Sales Engineer
Encs.

WHESSOE

Whessoe Limited
40 Broadway
London SW1H 0BR
Telephone : 01–930–3201 Telex : 23821
Cables : Whessoe London Telex
Head Office : Darlington Co. Durham DL3 6DS England. Telephone : 03–25–60188

Heavy Engineering Division

Registration : London 166242. Registered Office : Brinkburn Road, Darlington, Co. Durham DL3 6DS.

LONDON 24th.June, 1981. your ref our ref JT/AED.

Project Assessment & Development
Limited,
Page Street,
London, NW7 2ER

London Office
New Telephone Number
01-222 9311
0325 381 818
Darlington

For the attention of Mr.Peter Williams.

Dear Sirs,

Salter Ducks.

Following our recent meeting when we discussed your current requirements for a steel lining to the concrete pressure vessel we now comment as follows.

Prototype.

If we were to manufacture a single protoype lining, this would be carried out in our Darlington Works and a rough assessment of the order of cost for supplying, fabricating, welding and testing a steel lining 5 metres diameter x 25 metres long x 5 mm thick complete with flat ends and suitable attachments for tying back into the concrete vessel would produce a figure in the order of £850/tonne.

For this purpose we have assumed that special permission would be granted for transportation by road to Teesside.

2000MW Project.

Considering an overall project puts a different light on the subject and we would consider it to be totally illogical to utilise inland fabricating facilities, as actual handling costs, transportation costs and transportation problems would put the overall price out of context.

Based on your programme requirements of three vessels per week for a period of six years immediately points to the utilisation of either an area of the intended 'Duck' construction facility or if you were to use Graythorp Yard, our offshore facility at Dock Point could well suit your needs. Whichever of the circumstances, further consideration must be given to whether or not automated production will be required and further comments on this are made later.

/..........

Regarding the steel lining, your two main areas of difficulties are areas in which Whessoe have inavluable experience and in depth construction expertise, these are:-

1) Concrete/Steel interface.

Our experience in steel lining concrete vessels for the AGR Nuclear Power Stations is second to none and over the years we have developed a suitable method of attachment.

2) Producing large diameter lining to suit your Programme Requirements.

As mentioned above without an in depth study into the most economical and practical method of producing these linings, it is difficult to assess the feasibility of setting up an automated production run. On the surface it would appear to be a sensible suggestion, however substantial tooling costs and trial and error would be experienced.

From a quick investigation of your programme requirements it should be possible to achieve your needs with a squad of aprroximately 50 men working to our normal practice and using some light rolling equipment purposely installed on the construction facility.

In fact the overall dimensions although much thinner are very similar to work we have carried out for the CEGB at Dinorwic (Penstocks and Tunnel linings) and also a current similar scheme in Sri Lanka, where we are rolling the plate at site.

We hope this information although provided for information purposes only will be of use to you and we would welcome any further discussion you may require.

Yours faithfully,
WHESSOE HEAVY ENGINEERING LTD.,
PROCESS PLANT DIVISION.

John Tate

John Tate.
Sales Engineer.

SHS phoned 18th Oct 1988. They have not been asked any questions from ETSU about the difference between this and ETSU parametrics. Michael Fallon is M.P. OK to ask PQ.

S E C T I O N 10

D R A W I N G S

10. DRAWINGS

The following drawings are included herewith:-

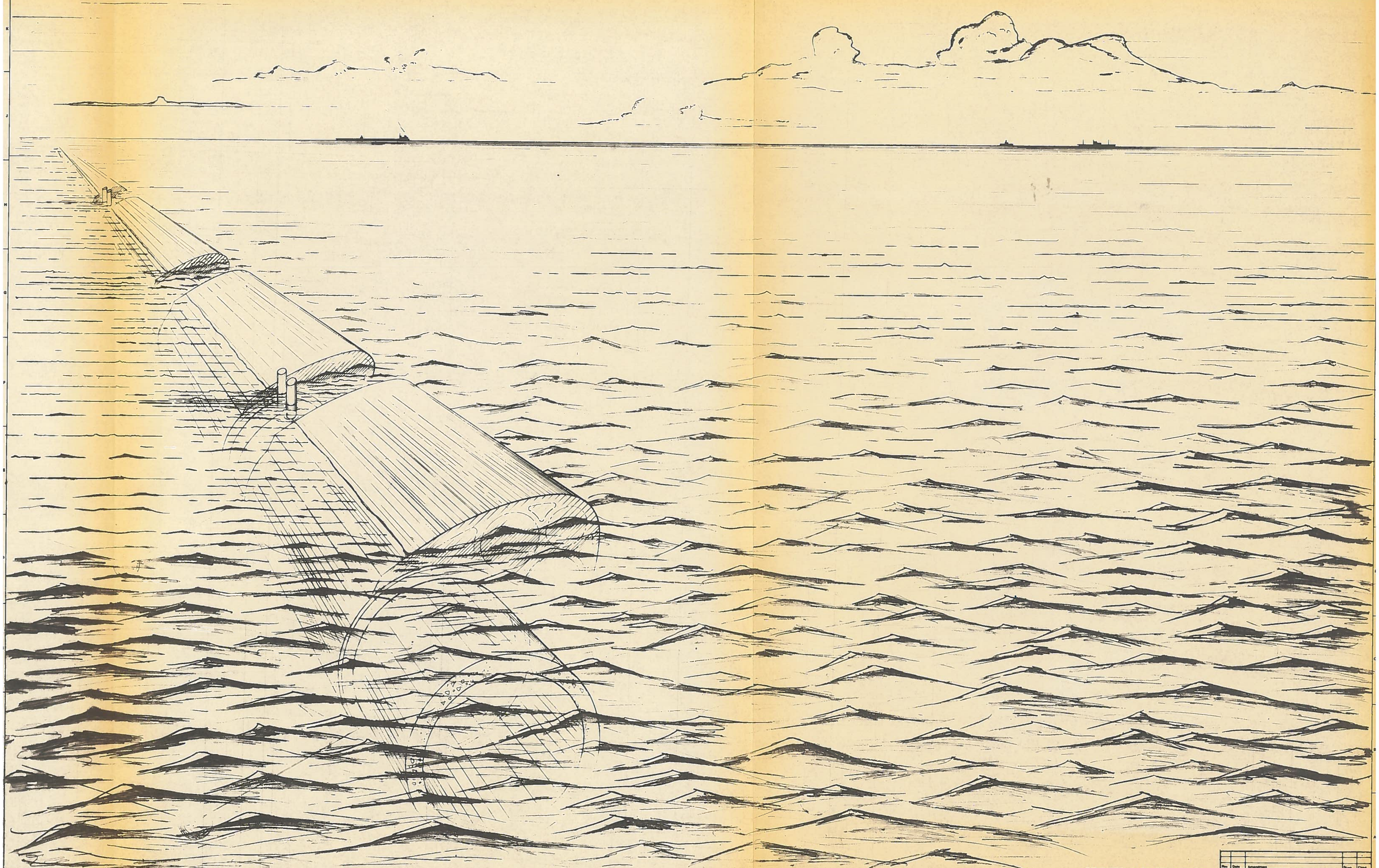
No.	Title
10020	Artist's Impression
10021	Geographical Location of Device
10030	G.A. of Duck Power Canister
10031	G.A. of Gyro Flywheel
10032	Gyro Main Bearing - Locating
10033	Gyro Cage Polar Bearing - Locating
10034	Gyro Cage Polar Bearing - Non Locating
10035	Gyro Outer Gimbal Bearings
10036	G.A. of Centrifugal Filter
10040	Schematic Arrangement of Spine Joint
10041	G.A. of Spine Joint
10042	Spine Joint Hydraulic Ram
10043	Arrangement of Spine Power Canister
10050	G.A. of Spine/Duck Bearing
10051	Spine/Duck Radial Bearing - Alter. Dets.
10052	Spine/Duck Axial Bearing - Alter. Dets.
10060	Ring Cam Pump Assembly - Elev.
10061	Ring Cam Pump Assembly - Plan
10070	Schematic Arrangement of P.C. Take-Up
10071	Schematic Arrangement of Power Trans.
10072	Schematic Diagram of Generator Circuits
10073	Arrangement of Spine Group Electrical Equip.
10080	Duck Power Canister Hydraulic Circuit

10. Drawings (Continued)

No.	Title
10081	Spine Joint Hydraulic Circuit
10082	Arrangement of Duck Power Canister Hyd. Mains
10083	Variable Axial Pump/Motor
10084	Start-up auxiliaries
10085	Smart Impedance & Eductor Pumps
10090	Schematic Arrangement of Mooring System
10101	G.A. of Spine/Duck
10102	Details of Spine
10201	Manufacturing Facility for Duck Bodies
10202	Manufacturing Facility for Spine Bodies
10203	Mechanical Construction Site

CHARTS

10103	Characteristic BM vs Stress Curves for Concrete
10104	Characteristic BM vs Stress Curves for Rebar
10105	Characteristic BM vs Stress Curves for Tendons
10106	S - N Curves for Plain Concrete
10107	Wave Climate for 1 Year Period
10108	Characteristic BM Histogram Build-up
10109	Mean BM vs Stress Curves for Concrete
10110	Mean BM Histogram Build-up
10111	Duck Buoyancy Calculations



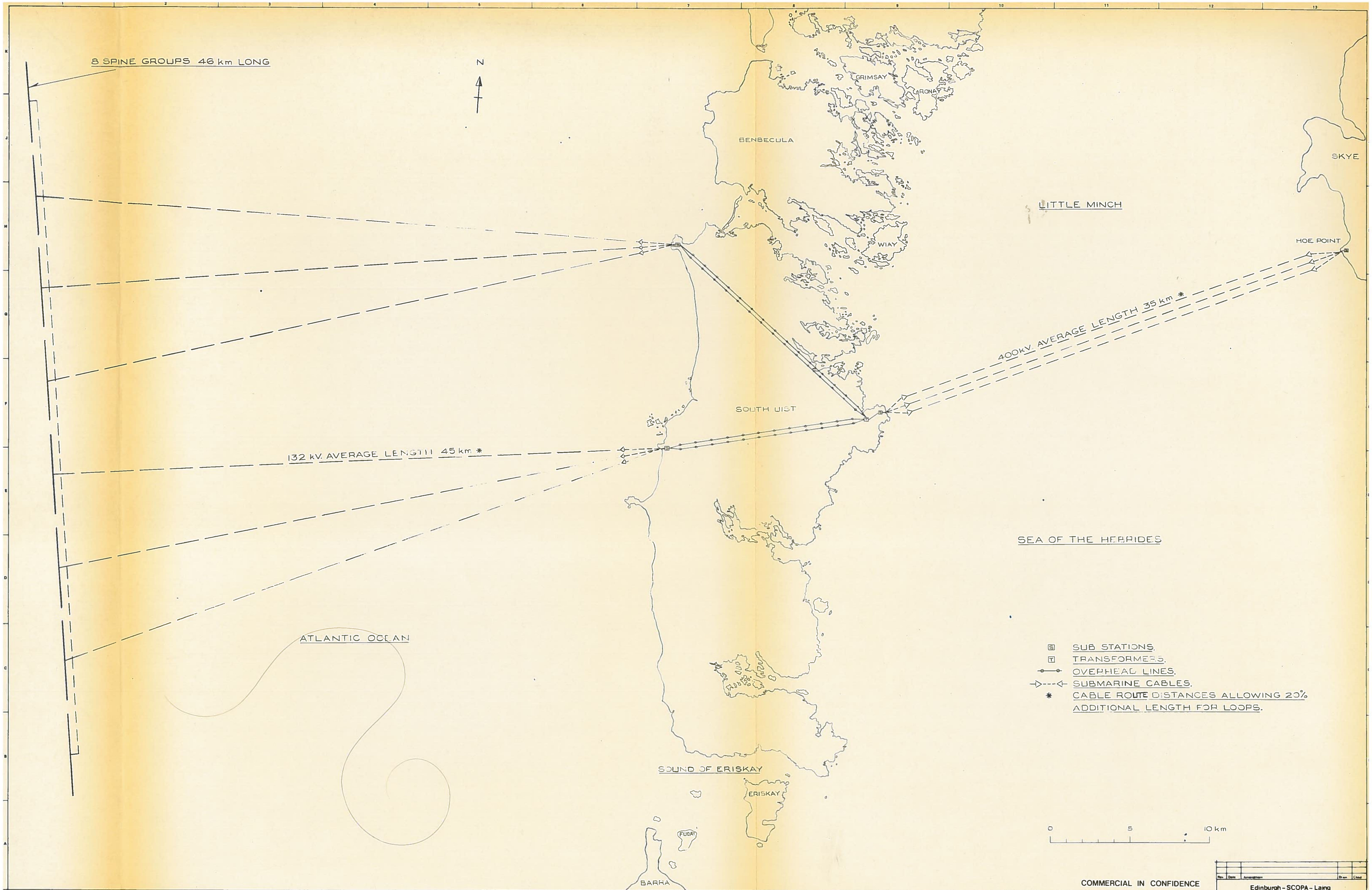
COMMERCIAL IN CONFIDENCE

Rev	Date	Amendment	Drwn	Chkd

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title
ARTIST'S IMPRESSION

Project No	PAD 100	L	9187-10020 - 0
Scale	NTS	Drawn MAR 18.11.81	Date	Chkd



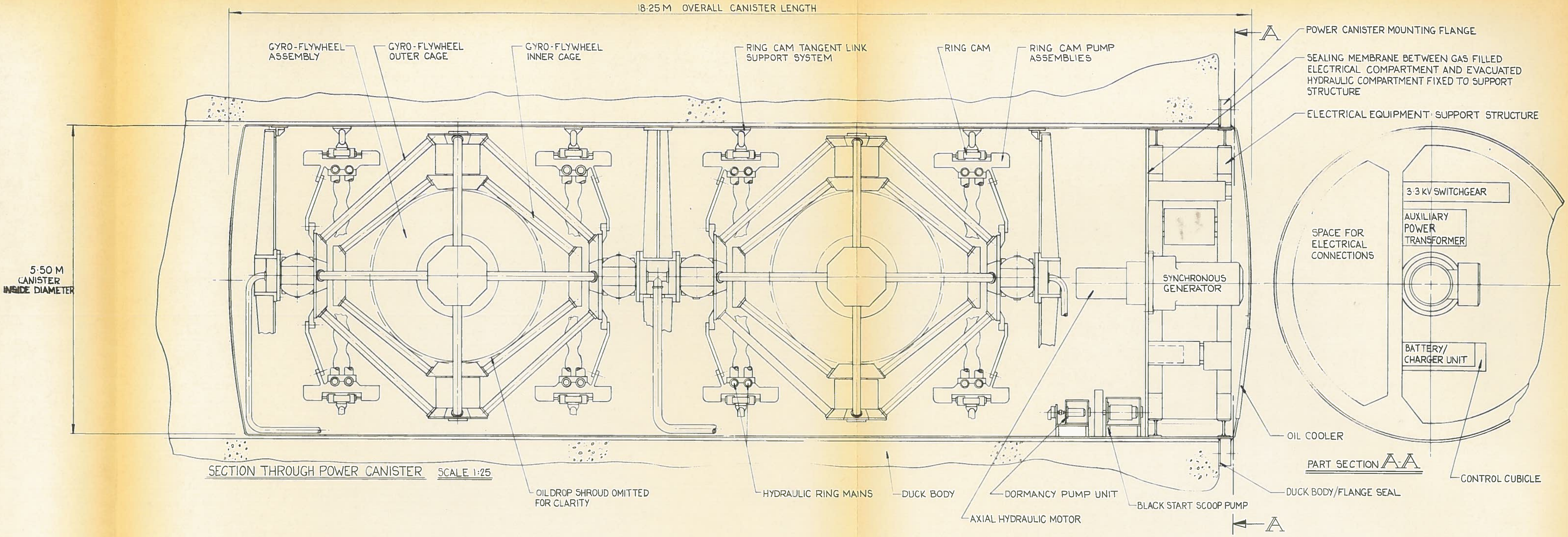
COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

GEOGRAPHICAL LOCATION OF DEVICES

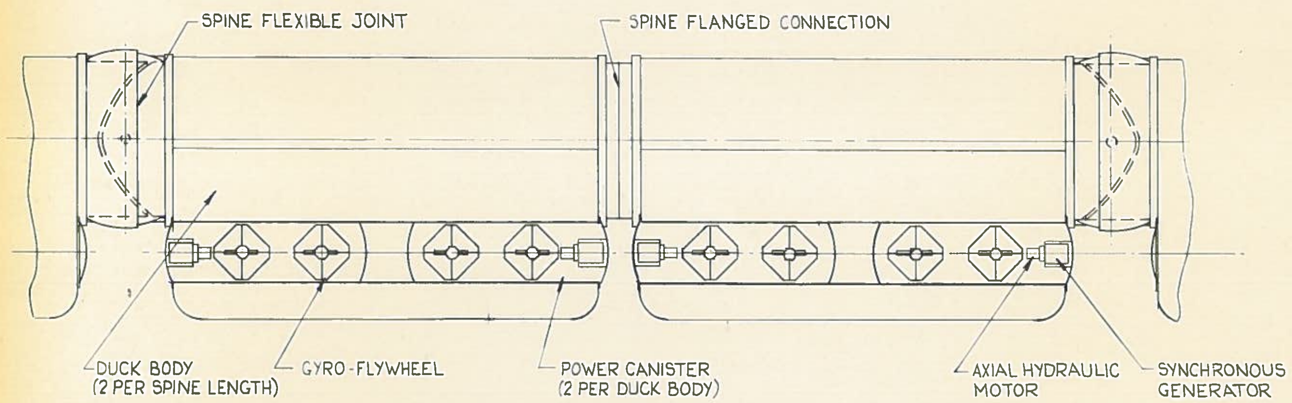
PAD 100 | L 9187-10021 | 0

AJH 13.11.81

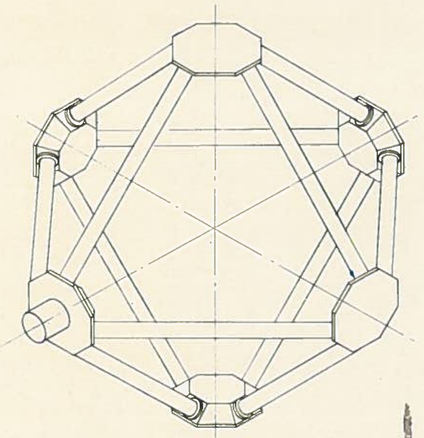


SECTION THROUGH POWER CANISTER SCALE 1:25

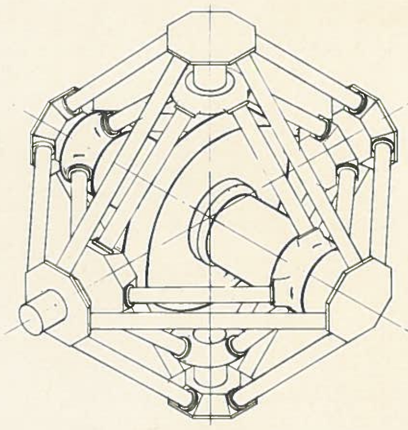
PART SECTION A-A



KEY ARRANGEMENT
SCALE 1:250



ISOMETRIC OF GEODETIC GYRO-FLYWHEEL SUPPORT CAGE (OUTER CAGE SHOWN)



ISOMETRIC OF GYRO-FLYWHEEL WITHIN SUPPORT CAGES

NOT TO SCALE

COMMERCIAL IN CONFIDENCE

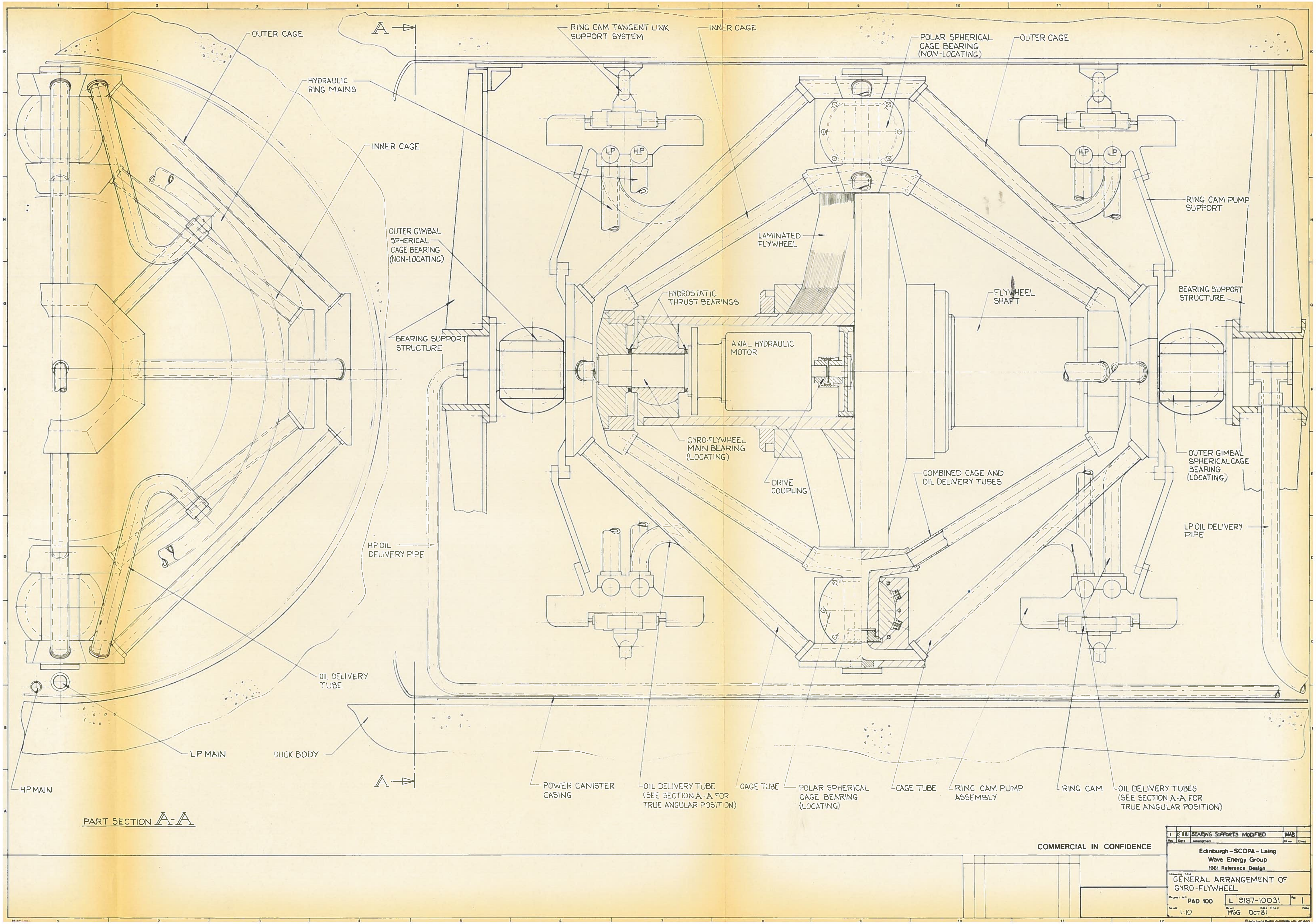
Rev	Date	Amendment	Drwn	Chkd
1	Nov 81	KEY ARRANGEMENT MODIFIED & ELECT. EQUIPMENT ADDED	MSG	

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

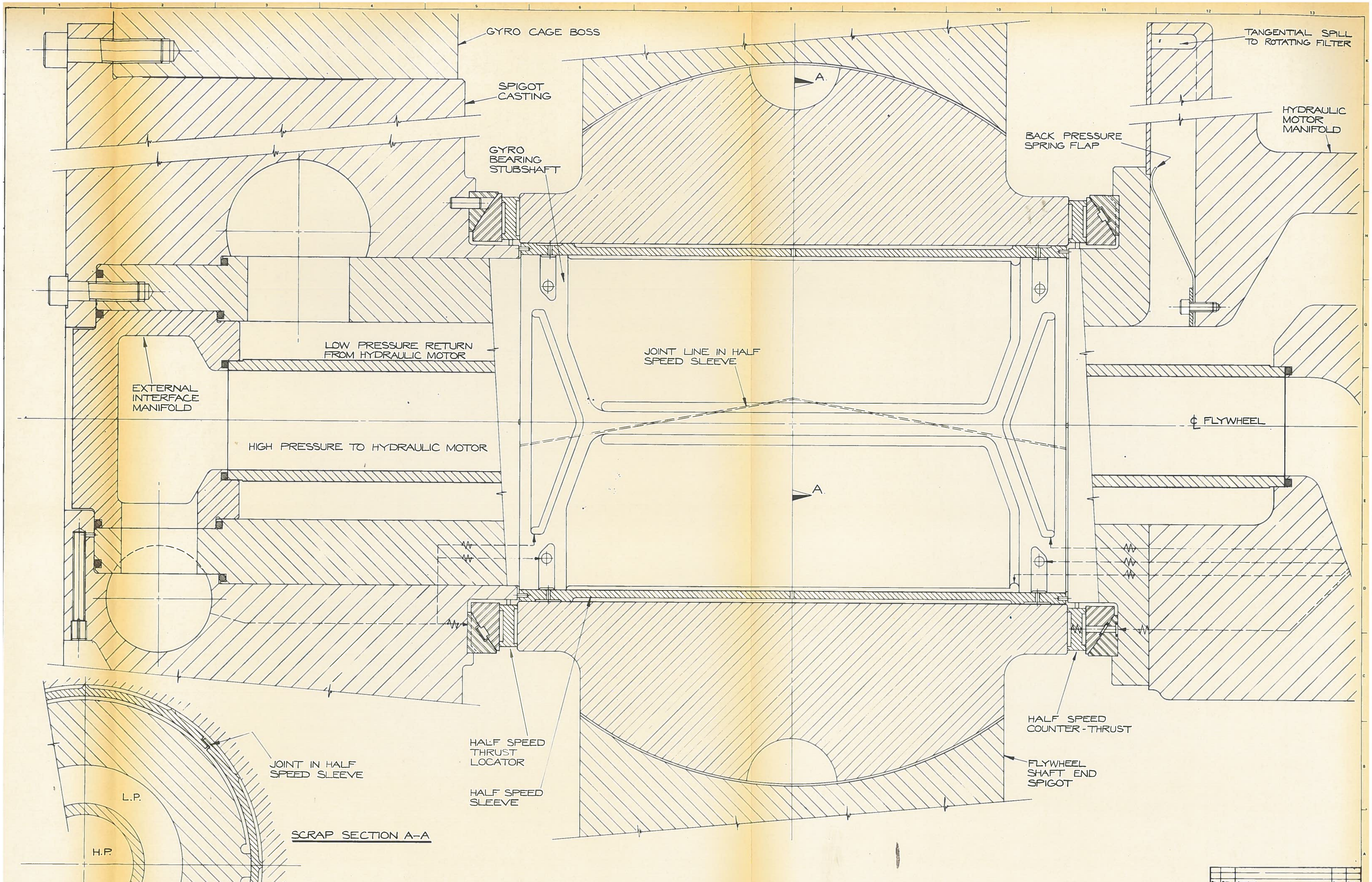
Drawing Title: GENERAL ARRANGEMENT OF DUCK POWER CANISTER

Project No: PAD 100 | L 9187-10030 | Rev 1

Scale: SEE DRAWING | Drwn: MSG | Date: OCT 81 | Chkd:

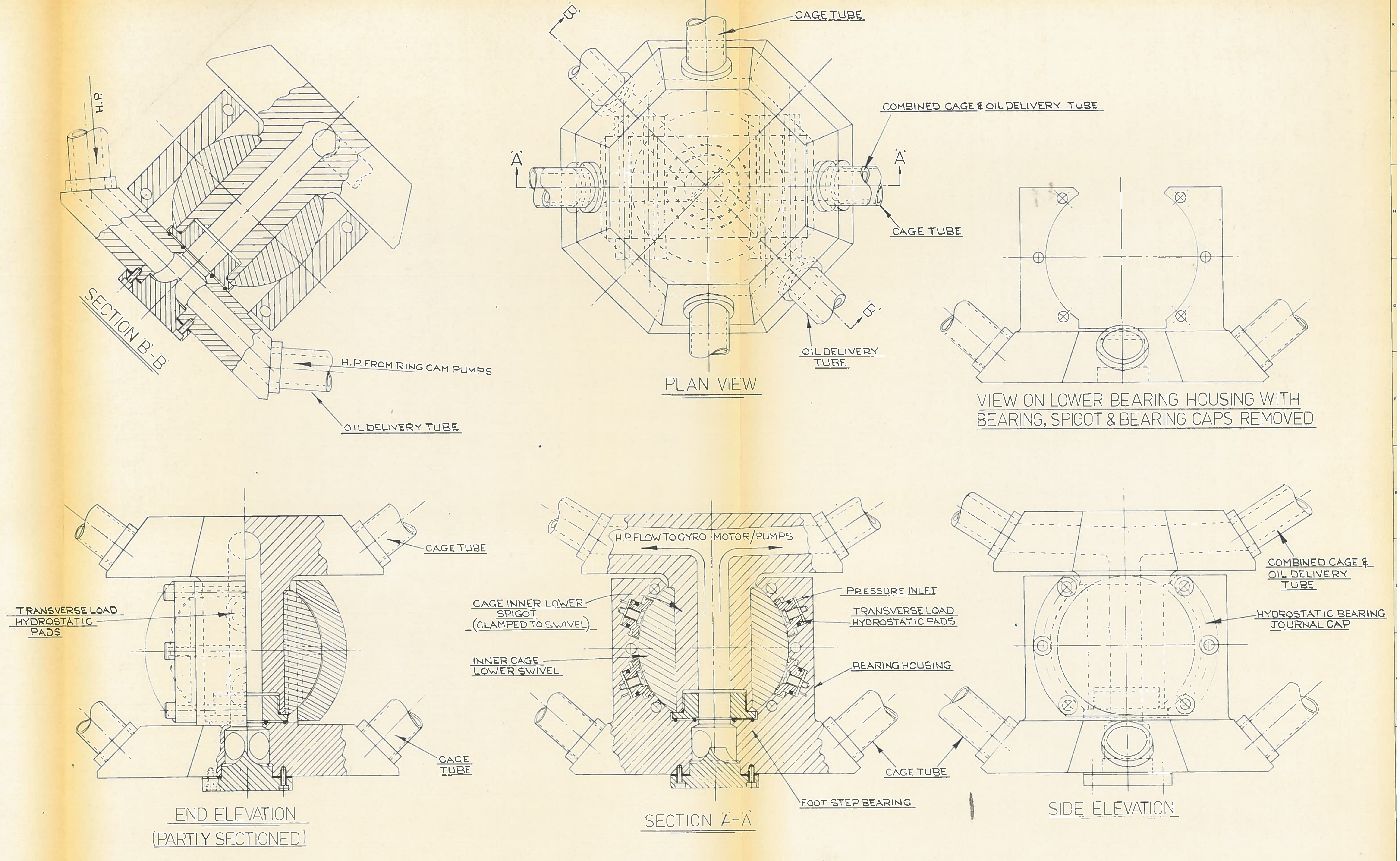


OUTER CAGE
HYDRAULIC RING MAINS
INNER CAGE
OUTER GIMBAL SPHERICAL CAGE BEARING (NON-LOCATING)
BEARING SUPPORT STRUCTURE
HP OIL DELIVERY PIPE
OIL DELIVERY TUBE
LP MAIN
DUCK BODY
HP MAIN
PART SECTION A-A
A
RING CAM TANGENT LINK SUPPORT SYSTEM
INNER CAGE
POLAR SPHERICAL CAGE BEARING (NON-LOCATING)
OUTER CAGE
L.P
H.P
RING CAM PUMP SUPPORT
LAMINATED FLYWHEEL
HYDROSTATIC THRUST BEARINGS
FLYWHEEL SHAFT
BEARING SUPPORT STRUCTURE
AXIAL HYDRAULIC MOTOR
GYRO-FLYWHEEL MAIN BEARING (LOCATING)
DRIVE COUPLING
COMBINED CAGE AND OIL DELIVERY TUBES
OUTER GIMBAL SPHERICAL CAGE BEARING (LOCATING)
LP OIL DELIVERY PIPE
POWER CANISTER CASING
OIL DELIVERY TUBE (SEE SECTION A-A FOR TRUE ANGULAR POSITION)
CAGE TUBE
POLAR SPHERICAL CAGE BEARING (LOCATING)
CAGE TUBE
RING CAM PUMP ASSEMBLY
RING CAM
OIL DELIVERY TUBES (SEE SECTION A-A FOR TRUE ANGULAR POSITION)
COMMERCIAL IN CONFIDENCE
1 12.11.81 BEARING SUPPORTS MODIFIED MAB
Edinburgh-SCOPA-Laing Wave Energy Group
1981 Reference Design
GENERAL ARRANGEMENT OF GYRO-FLYWHEEL
PAD 100
L 9187-10031
1
1:10
MSG Oct 81



COMMERCIAL IN CONFIDENCE

Rev	Date	Amendment
Edinburgh - SCOPA - Laing Wave Energy Group 1981 Reference Design		
Drawing Title: GYRO MAIN BEARING LOCATING END		
Project No: PAD 100	L 9187-10032	Rev: 0
Scale: FULL SIZE	Drawn: MAB Date: 27.2.80	Chkd



THIRD ANGLE PROJECTION

COMMERCIAL IN CONFIDENCE

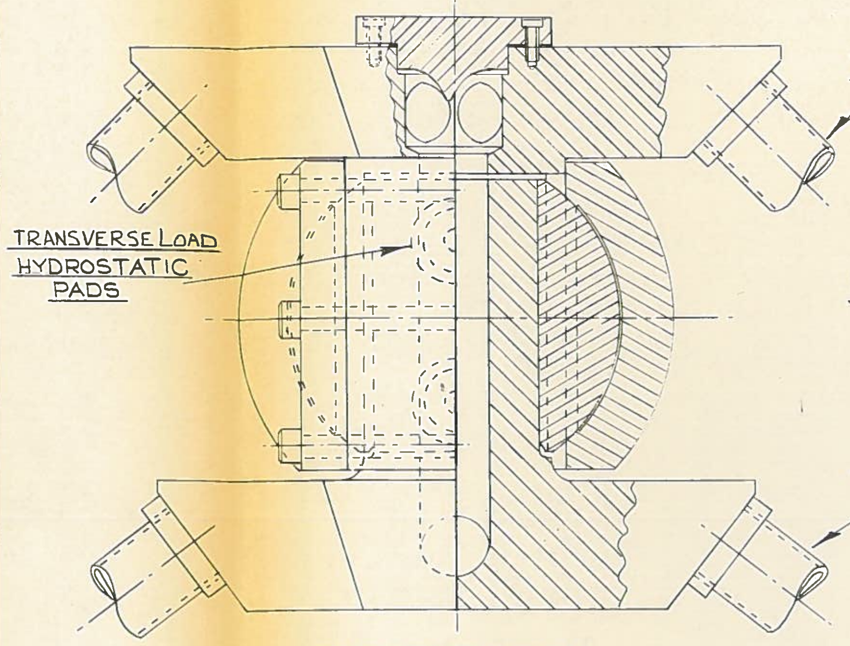
Rev	Date	Amendment	Drawn	Chkd
1		HYDRO-STATIC PADS MODIFIED	N.E.M.	

Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

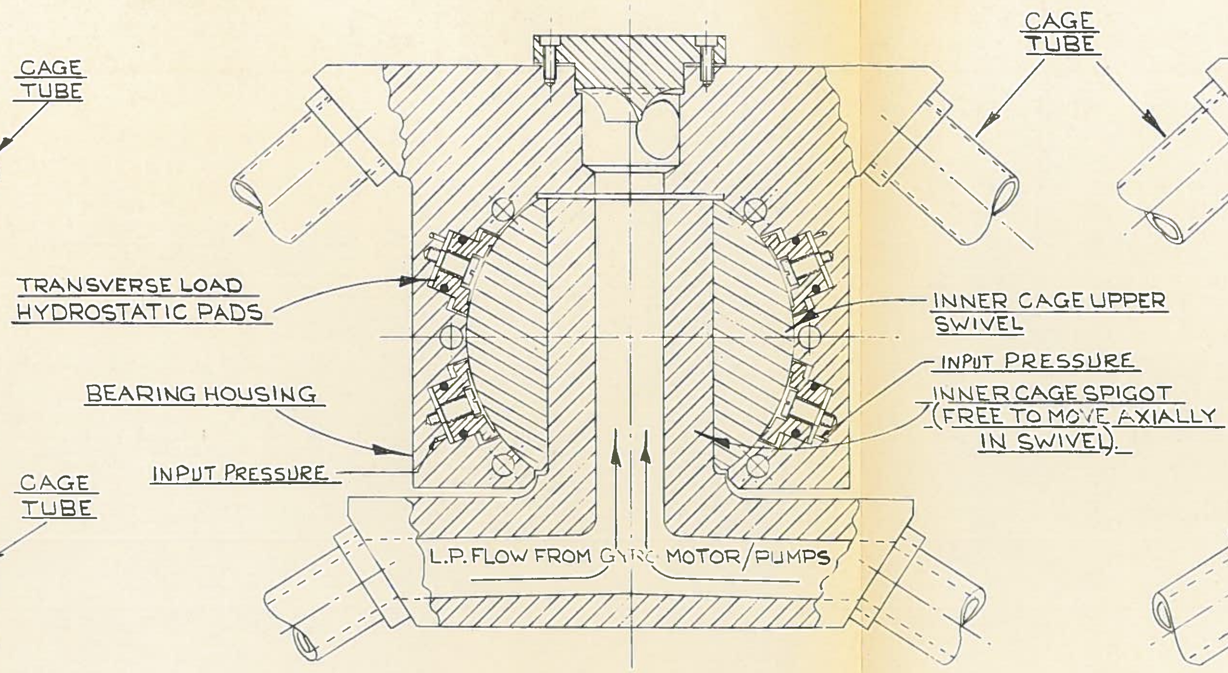
Drawing Title: GYRO CAGE POLAR BEARING - LOCATING (SOUTH).

Project No: PAD 100 — L 9187-10033 — Rev 1

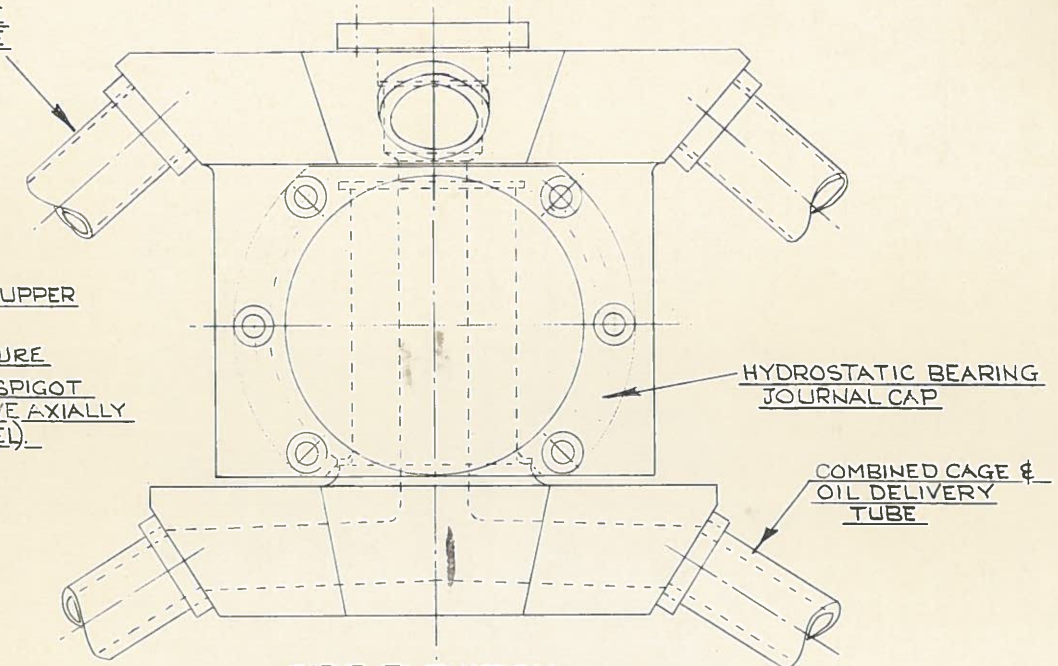
Scale: 1:5 — Drawn: N.E.M. 30-10-81



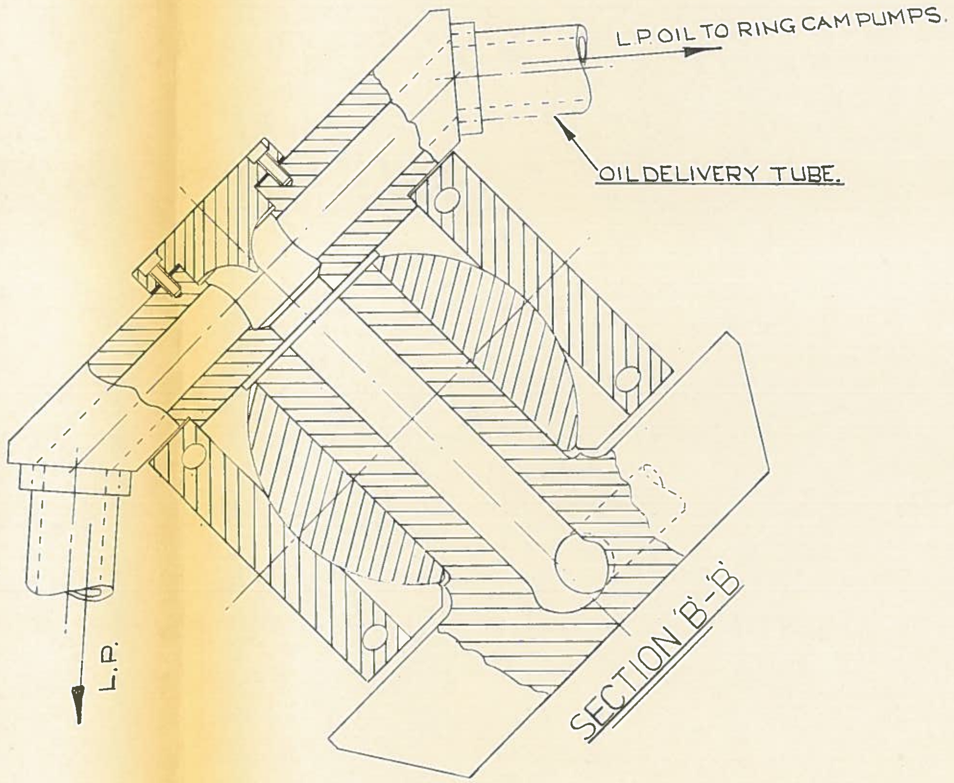
END ELEVATION



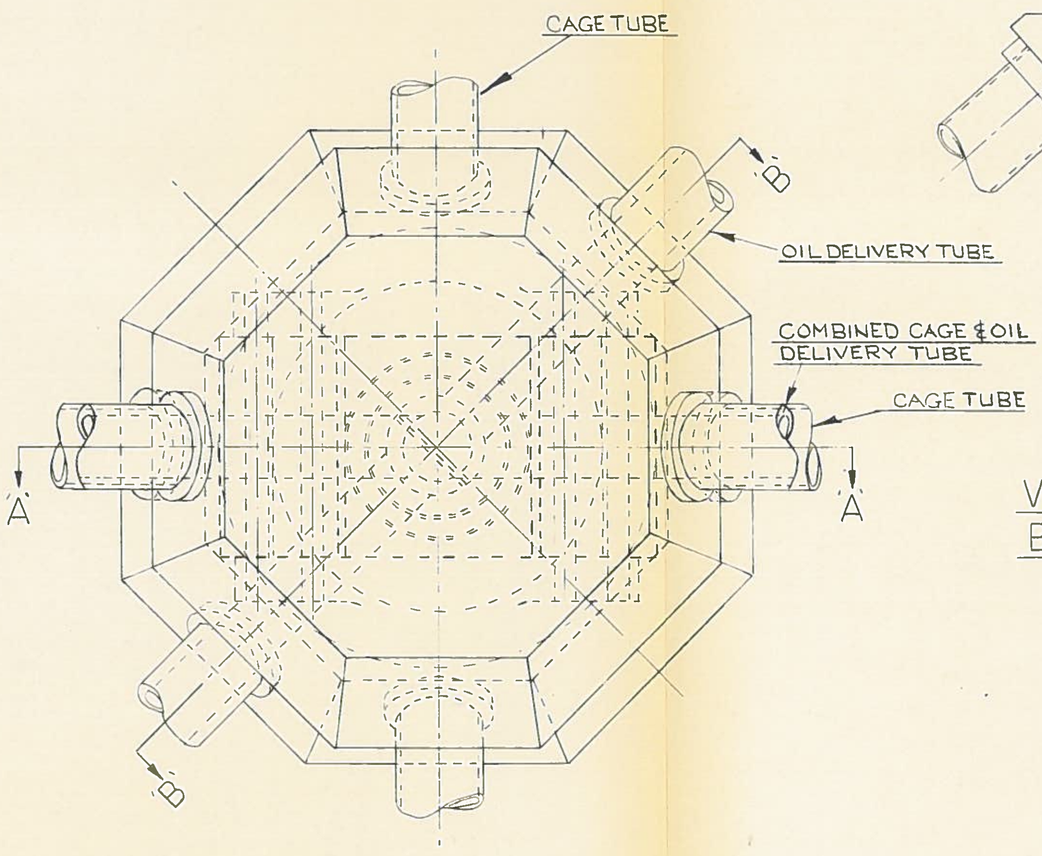
SECTION 'A'-'A'



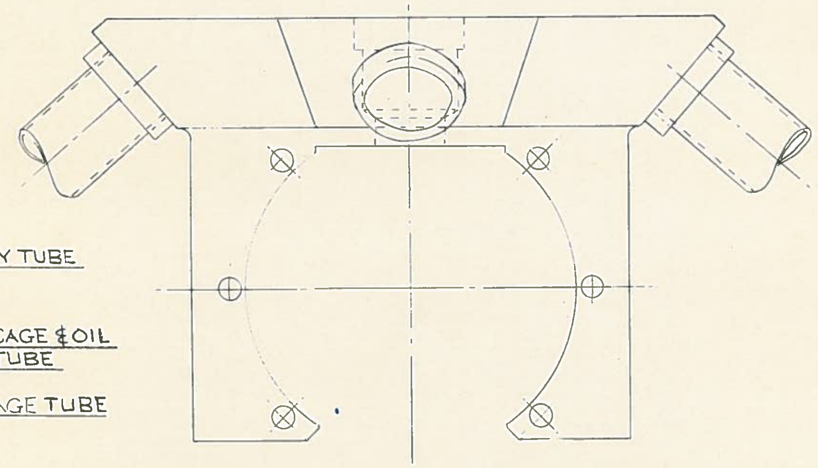
SIDE ELEVATION



SECTION 'B'-'B'



INVERTED PLAN VIEW



VIEW ON UPPER BEARING HOUSING WITH BEARING, SPIGOT & BEARING CAPS REMOVED

THIRD ANGLE PROJECTION

COMMERCIAL IN CONFIDENCE

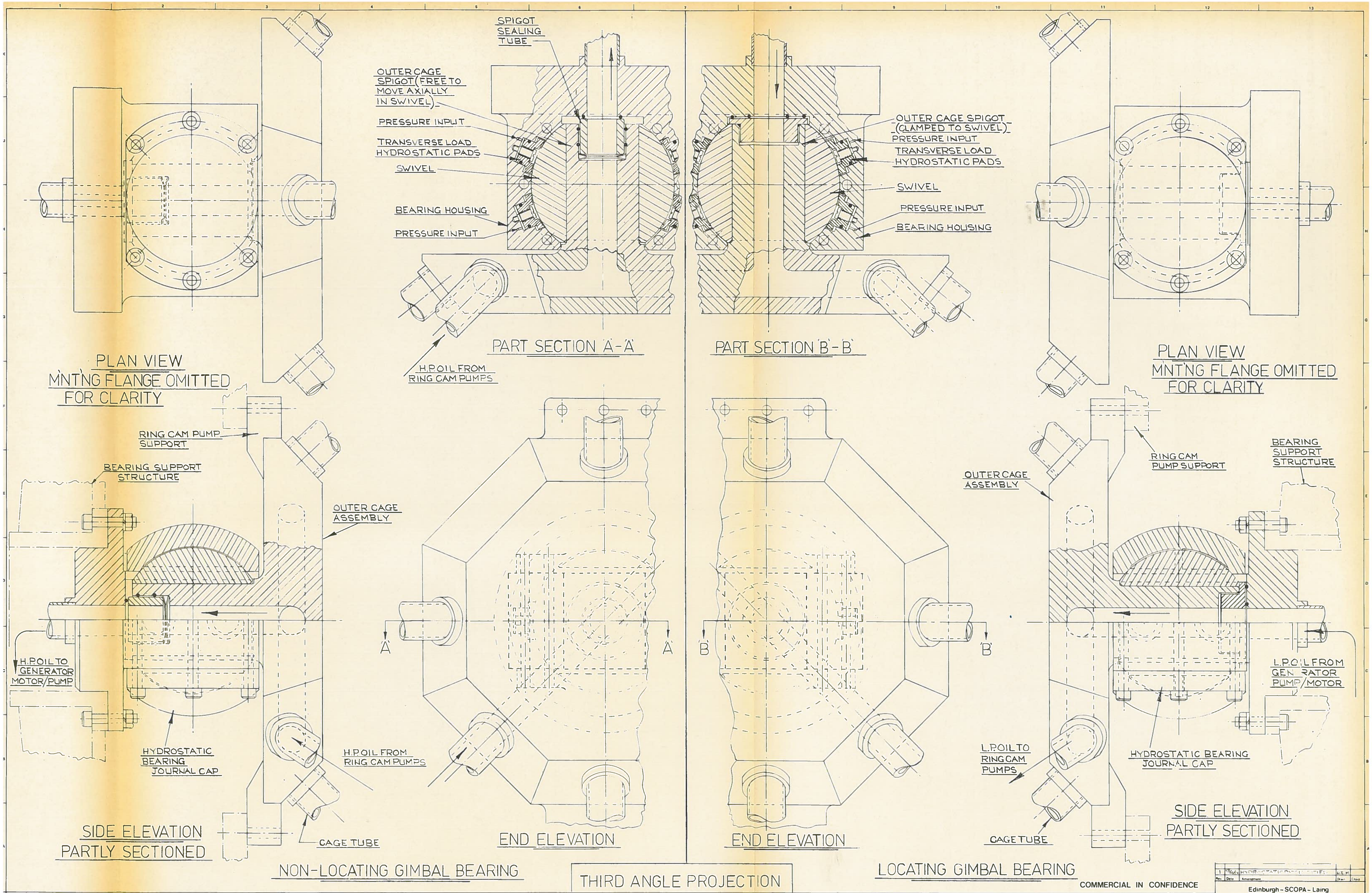
Rev.	Date	Amendment	Drawn	Chkd
1	13/11/81	HYDRO STATIC PADS MODIFIED	N.E.M.	

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title
GYRO CAGE POLAR BEARING - NON LOCATING (NORTH).

Project No. PAD 100 | L 9187-10034 | Rev. 1

Scale 1:5 | Drawn N.E.M. Date 3/11/81 | Chkd | Date



COMMERCIAL IN CONFIDENCE

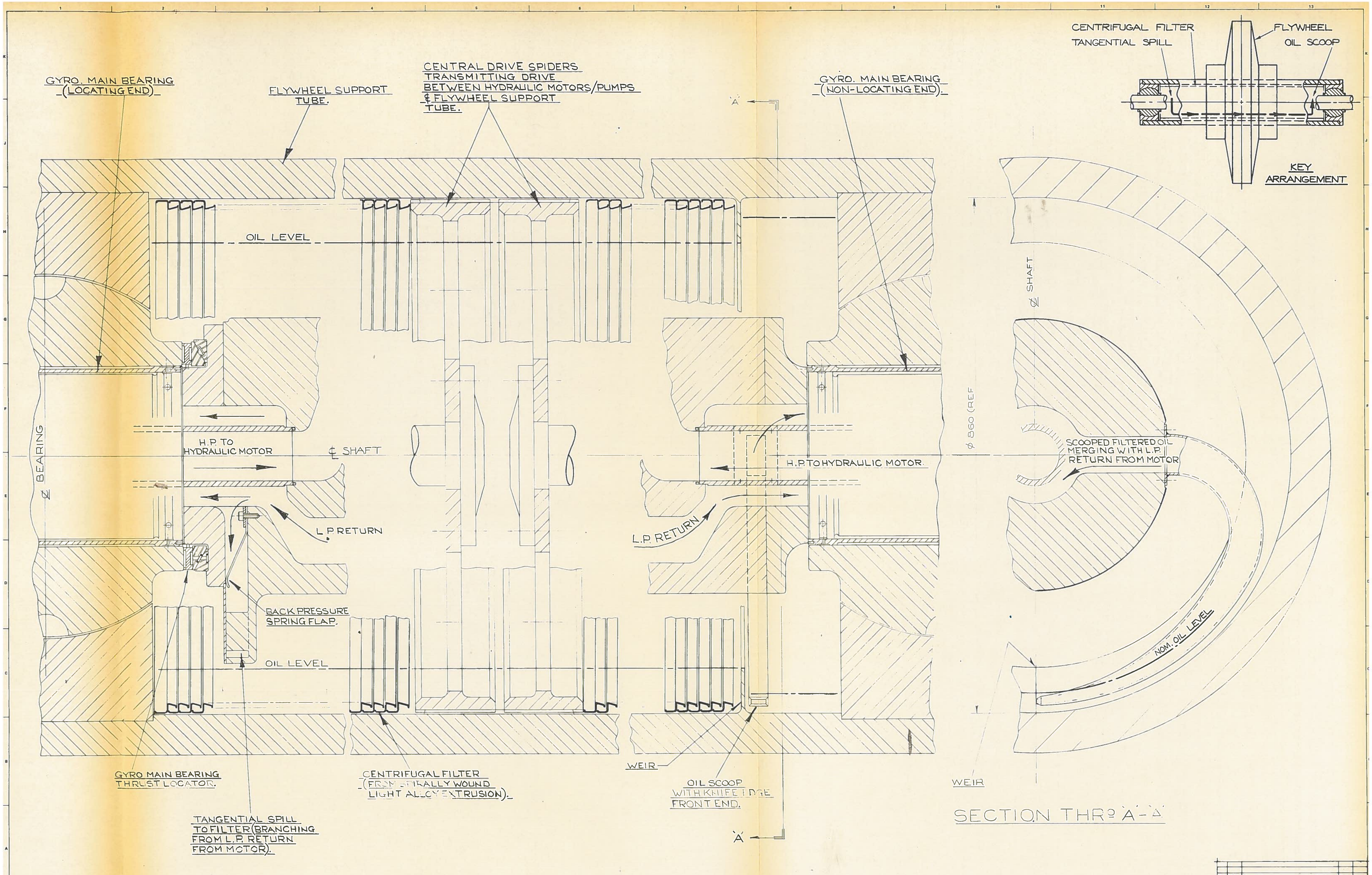
Rev	Date	Amendment	Drwn	Chkd
1	13/4/...	HYDROSTATIC PADS ...	N.E.M	

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title: GYRO OUTER GIMBAL BEARINGS

Project No: PAD 100 — L 9187-10035 — Rev 1

Scale 1:5 — Drwn N.E.M. 4-11-81



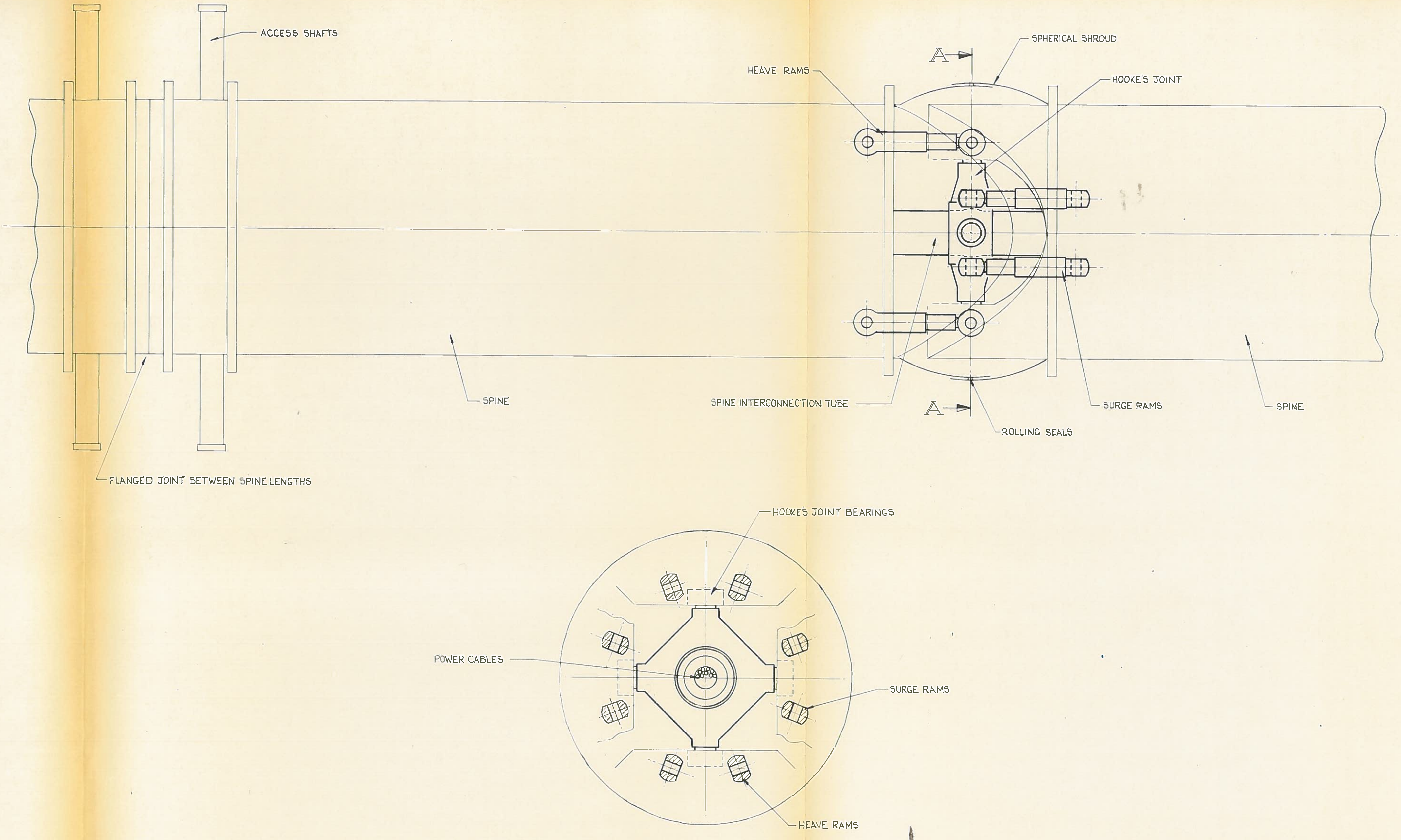
COMMERCIAL IN CONFIDENCE

Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

Drawing Title: GENERAL ARRANGEMENT OF CENTRIFUGAL FILTER.

Project No. PAD 100 | L 9187-10036 | 0

Scale 1 : 2 | Drawn AJH | 28-10-81



SECTION AA

COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

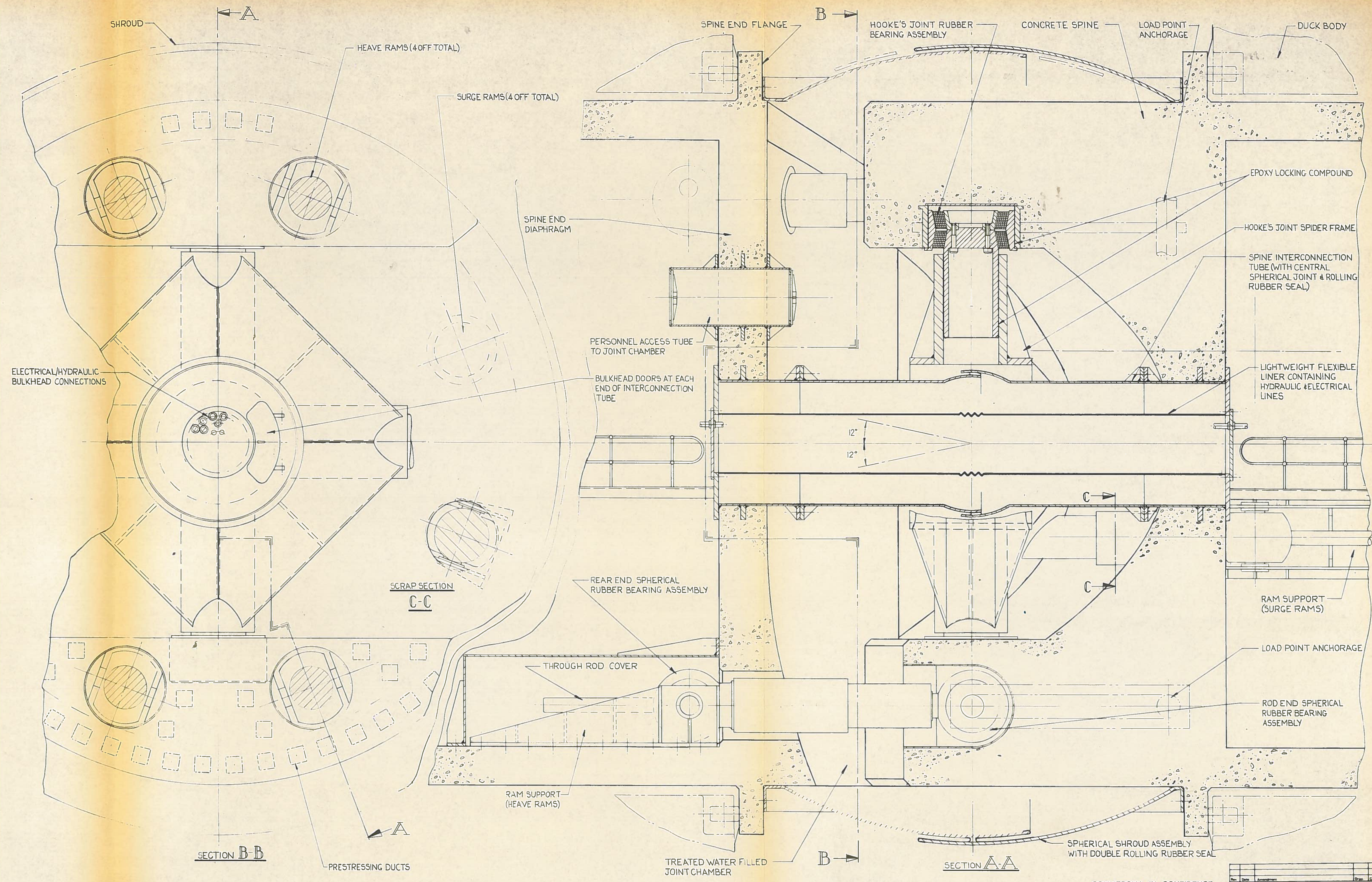
Drawing Title: SCHEMATIC ARRANGEMENT OF SPINE JOINT

Project No: PAD 100

L 9187-10040 Rev. 0

Scale: 1:75

Drawn: MSG Nov 81



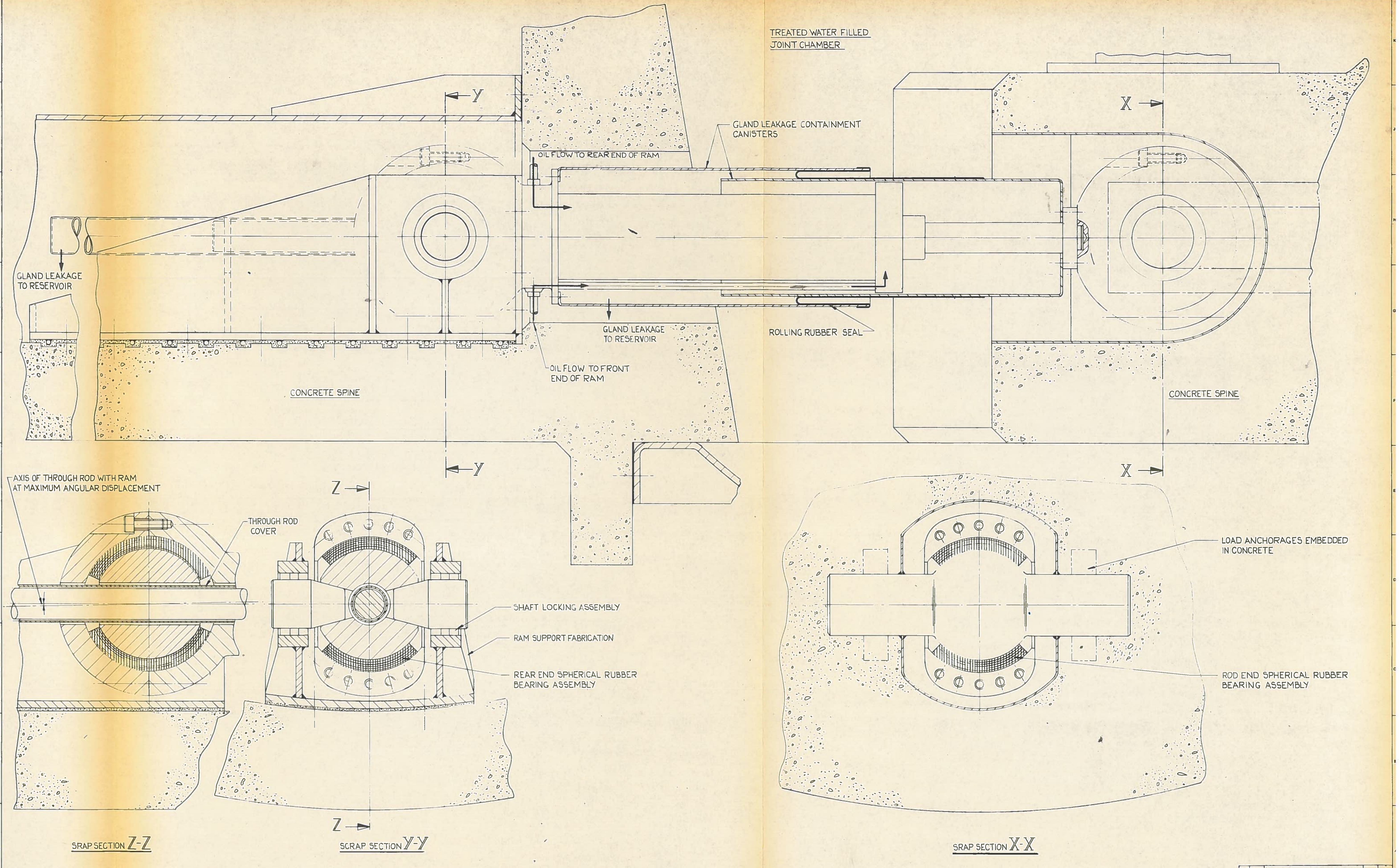
COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title: GENERAL ARRANGEMENT OF SPINE JOINT

Project No: PAD 100 | L 9187-10041 | Rev: 0

Scale: 1:25 | Drawn: MSG | Date: Nov 81



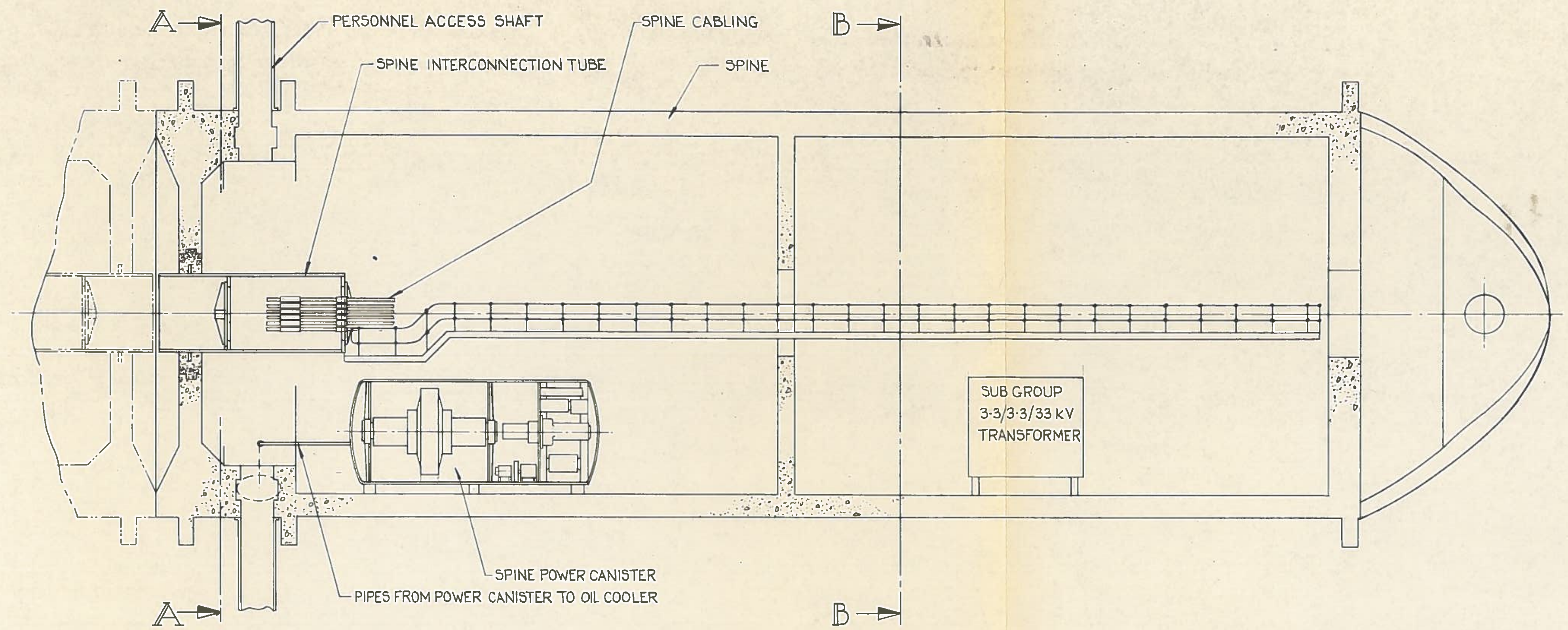
COMMERCIAL IN CONFIDENCE

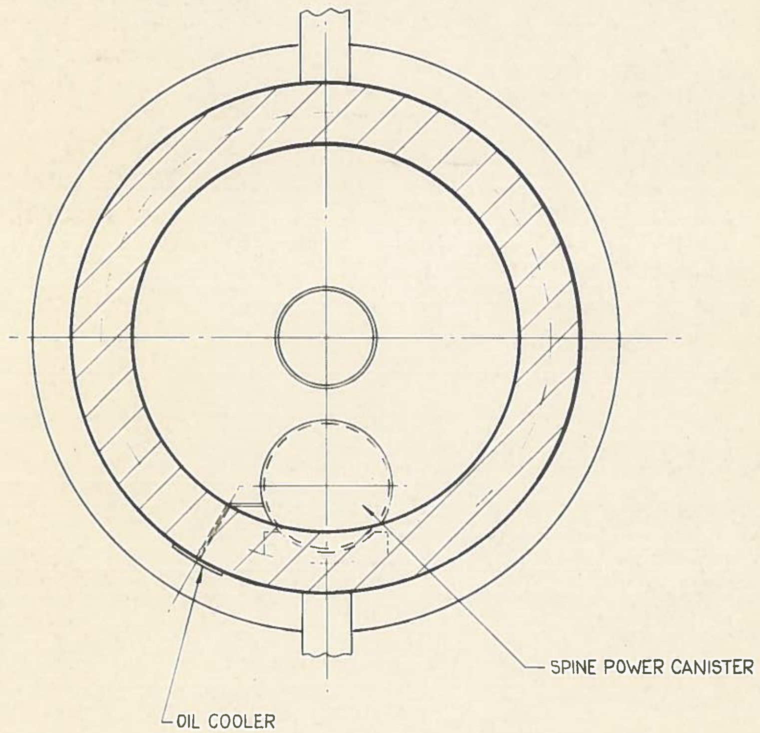
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

SPINE JOINT HYDRAULIC RAM

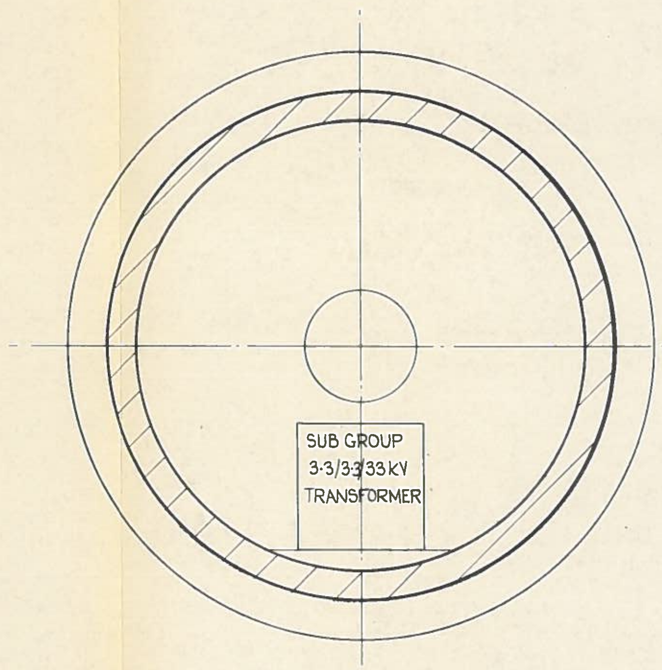
PAD 100 | L 9187-10042 | 0

1:10 | MSG | Nov81





SECTION A-A

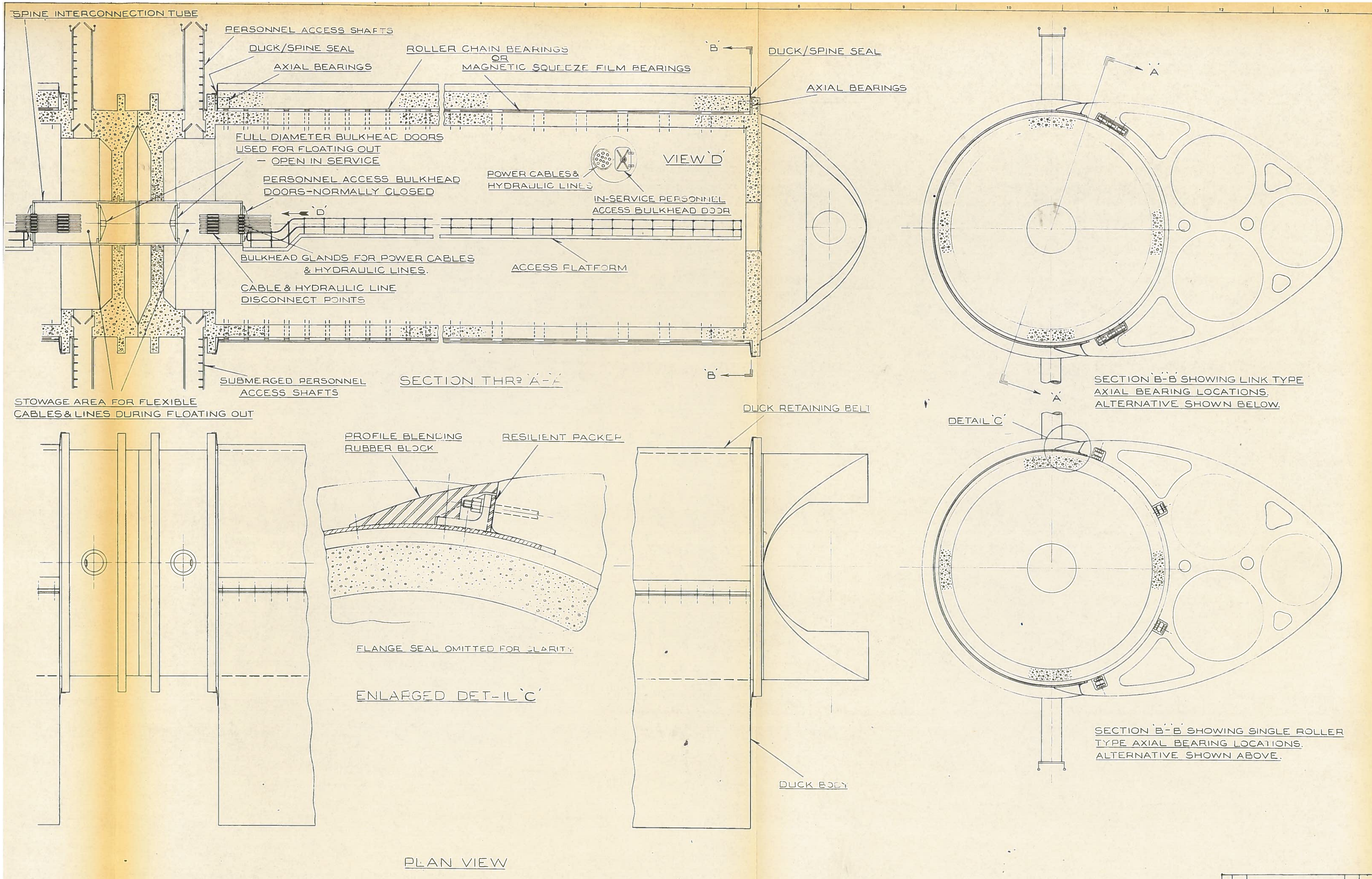


SECTION B-B

COMMERCIAL IN CONFIDENCE

ARRANGEMENT OF SPINE POWER CANISTER

1:75



COMMERCIAL IN CONFIDENCE

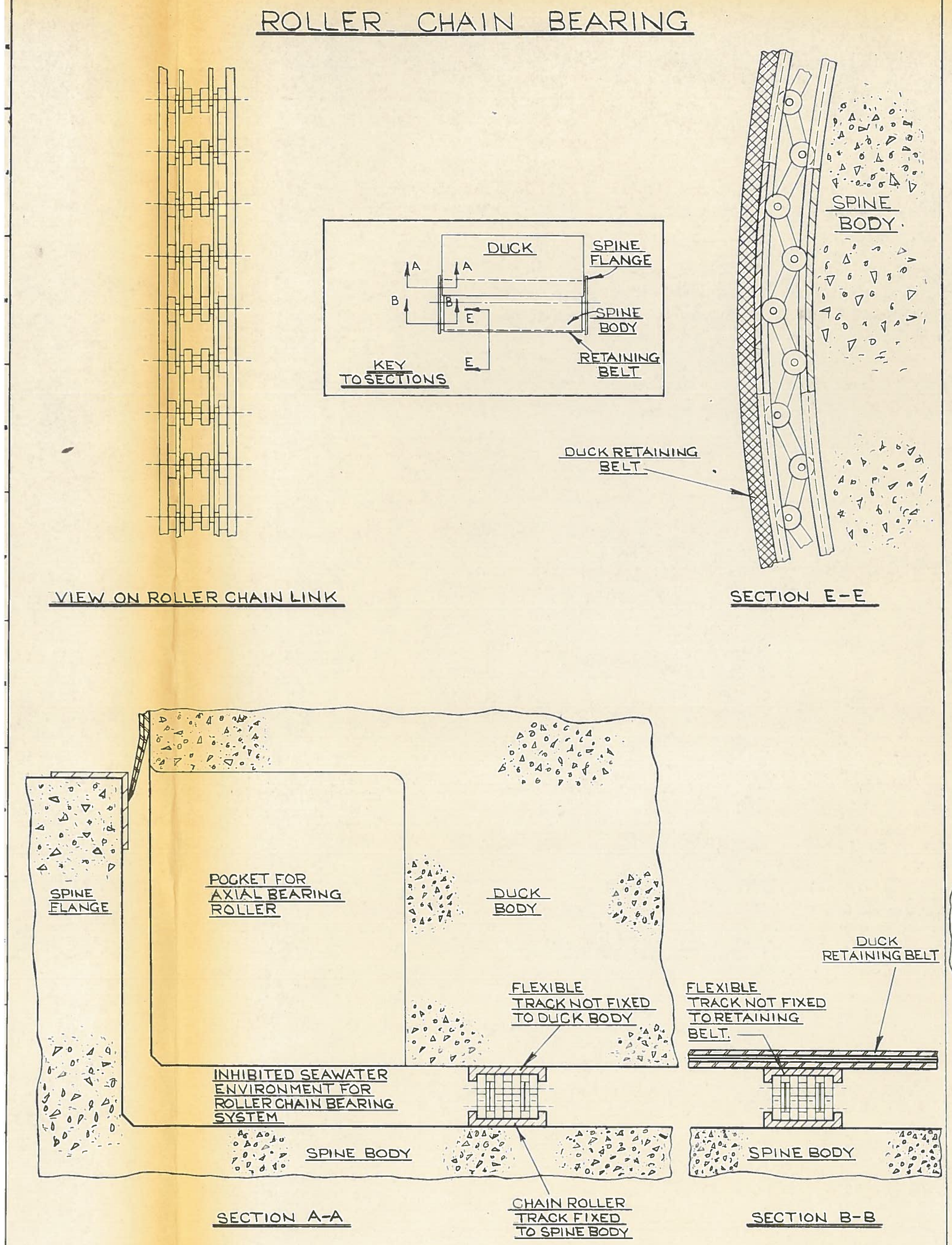
Rev	Date	Amendment	Drwn	Chkd

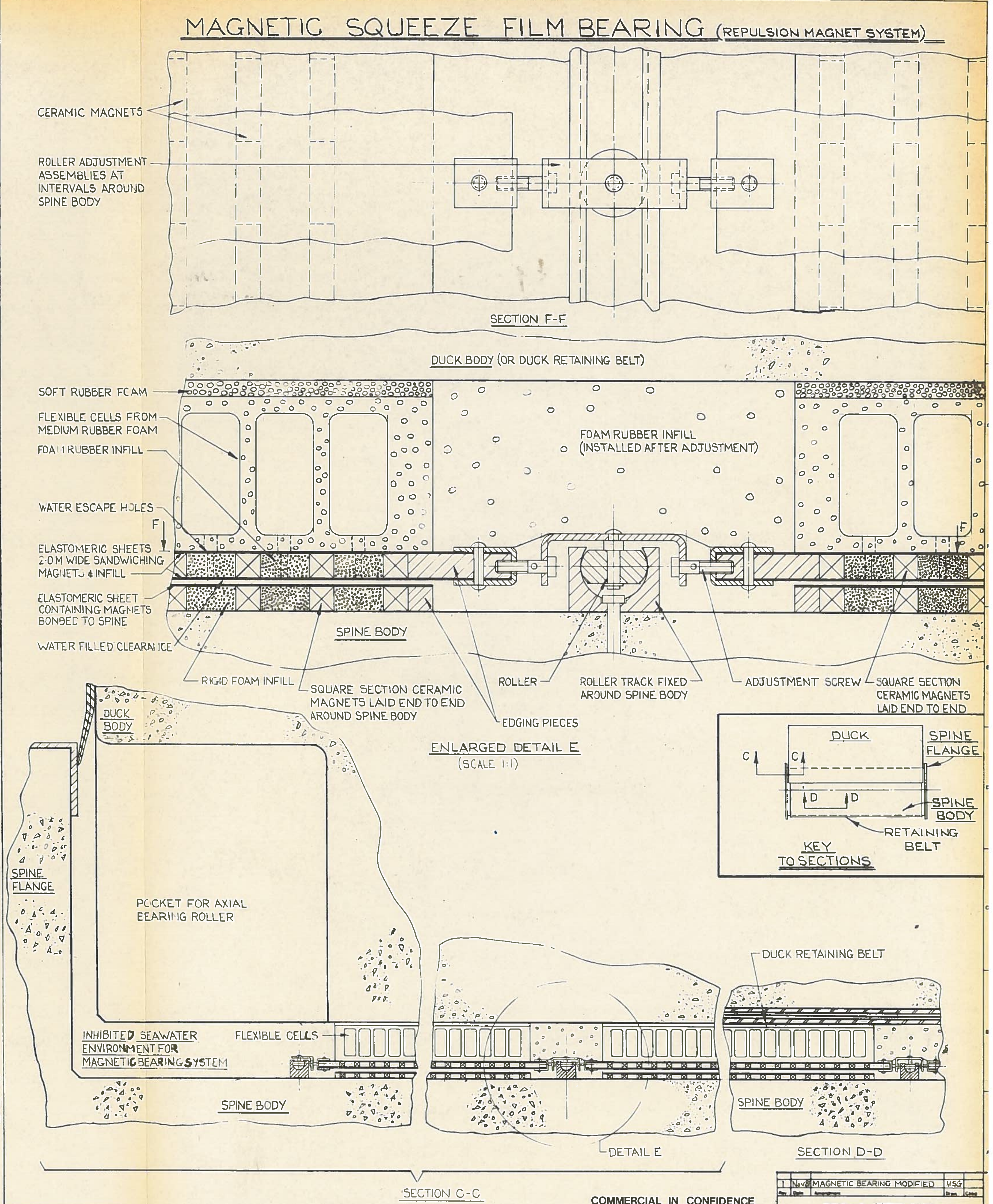
Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

Drawing Title: GENERAL ARRANGEMENT OF SPINE/DUCK BEARINGS

Project No: PAD 100 | L 9187-10050 | Rev: 0

Scale: 1:75 | Drwn: A.J.H | Date: 5/11/81





COMMERCIAL IN CONFIDENCE

1	Nov 82	MAGNETIC BEARING MODIFIED

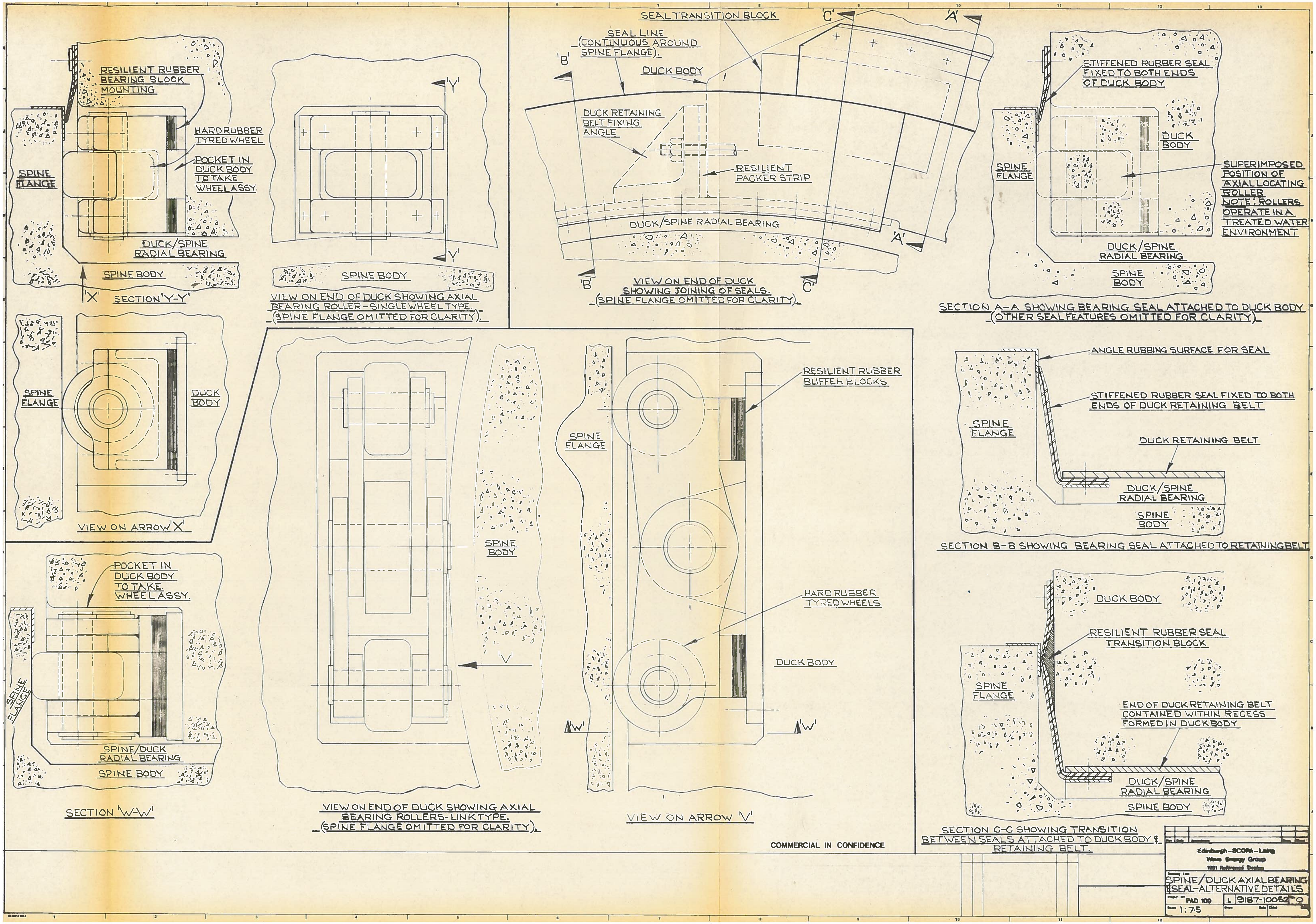
Edinburgh - SCOPA - Laing Wave Energy Group

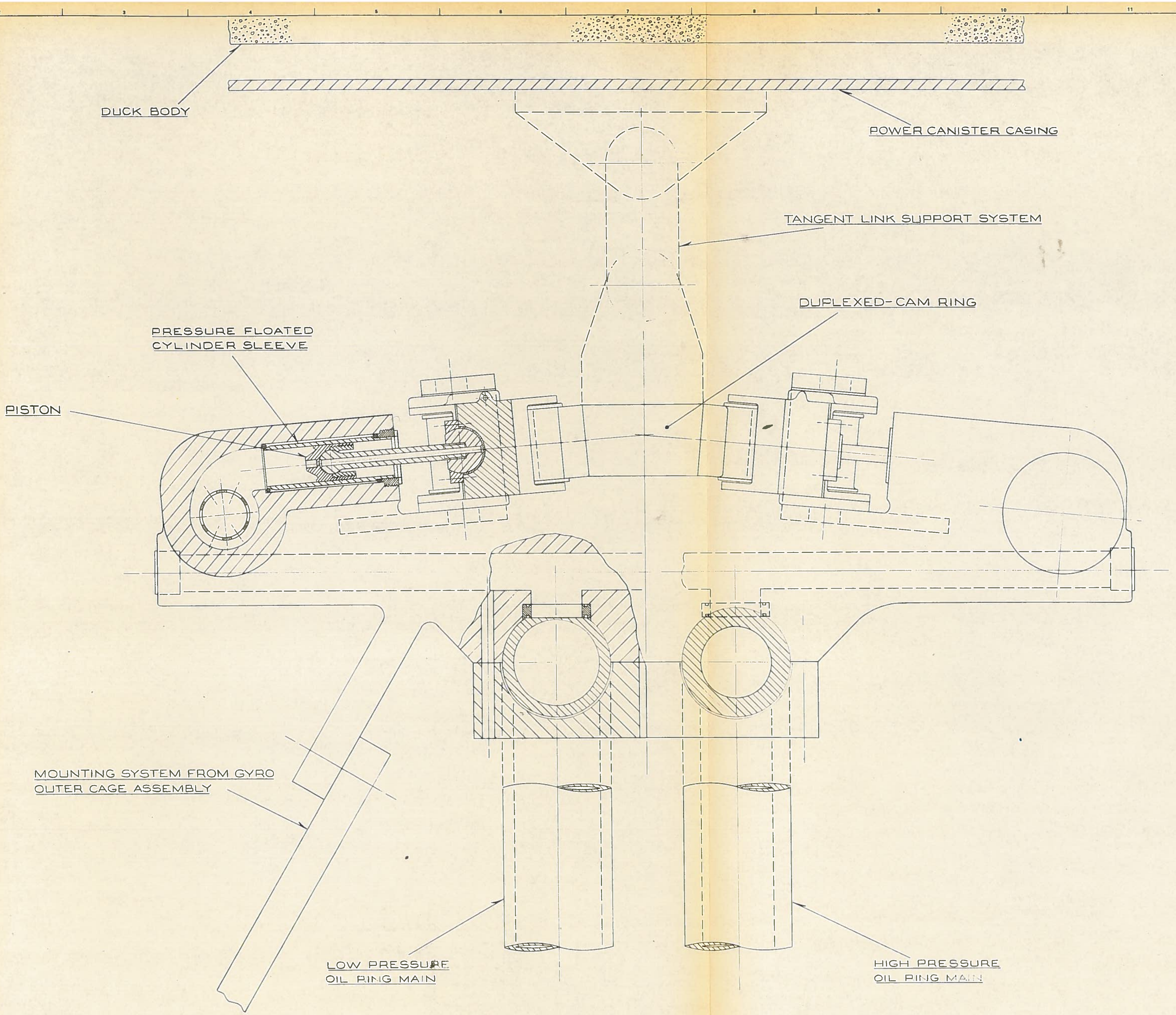
1981 Reference Design

Drawing Title: SPINE/DUCK RADIAL BEARING - ALTERNATIVE DETAILS

Project No	Drawing No	Rev
PAD 100	L 3187-10051	1

Scale: 1:5 — Drawn: A.J.H. Oct 81


RESILIENT RUBBER BEARING BLOCK MOUNTING
HARD RUBBER TYRED WHEEL
POCKET IN DUCK BODY TO TAKE WHEEL ASSY.
SPINE FLANGE
DUCK/SPINE RADIAL BEARING
SPINE BODY
'X'
SECTION 'Y-Y'
'Y'
'Y'
SPINE BODY
VIEW ON END OF DUCK SHOWING AXIAL BEARING ROLLER - SINGLE WHEEL TYPE. (SPINE FLANGE OMITTED FOR CLARITY).
SEAL TRANSITION BLOCK
'C'
'A'
SEAL LINE (CONTINUOUS AROUND SPINE FLANGE).
'B'
DUCK BODY
DUCK RETAINING BELT FIXING ANGLE
RESILIENT PACKER STRIP
DUCK/SPINE RADIAL BEARING
'A'
'B'
'C'
VIEW ON END OF DUCK SHOWING JOINING OF SEALS. (SPINE FLANGE OMITTED FOR CLARITY).
STIFFENED RUBBER SEAL FIXED TO BOTH ENDS OF DUCK BODY
DUCK BODY
SPINE FLANGE
SUPERIMPOSED POSITION OF AXIAL LOCATING ROLLER
NOTE: ROLLERS OPERATE IN A TREATED WATER ENVIRONMENT
DUCK/SPINE RADIAL BEARING
SPINE BODY
SECTION A-A SHOWING BEARING SEAL ATTACHED TO DUCK BODY (OTHER SEAL FEATURES OMITTED FOR CLARITY)
SPINE FLANGE
DUCK BODY
VIEW ON ARROW 'X'
RESILIENT RUBBER BUFFER BLOCKS
SPINE FLANGE
SPINE BODY
'V'
HARD RUBBER TYRED WHEELS
DUCK BODY
'W'
'W'
ANGLE RUBBING SURFACE FOR SEAL
STIFFENED RUBBER SEAL FIXED TO BOTH ENDS OF DUCK RETAINING BELT
SPINE FLANGE
DUCK RETAINING BELT
DUCK/SPINE RADIAL BEARING
SPINE BODY
SECTION B-B SHOWING BEARING SEAL ATTACHED TO RETAINING BELT
POCKET IN DUCK BODY TO TAKE WHEEL ASSY.
SPINE FLANGE
SPINE/DUCK RADIAL BEARING
SPINE BODY
SECTION 'W-W'
VIEW ON END OF DUCK SHOWING AXIAL BEARING ROLLERS - LINK TYPE. (SPINE FLANGE OMITTED FOR CLARITY).
VIEW ON ARROW 'V'
DUCK BODY
RESILIENT RUBBER SEAL TRANSITION BLOCK
SPINE FLANGE
END OF DUCK RETAINING BELT CONTAINED WITHIN RECESS FORMED IN DUCK BODY
DUCK/SPINE RADIAL BEARING
SPINE BODY
SECTION C-C SHOWING TRANSITION BETWEEN SEALS ATTACHED TO DUCK BODY & RETAINING BELT.
COMMERCIAL IN CONFIDENCE
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design
Drawing Title
SPINE/DUCK AXIAL BEARING & SEAL - ALTERNATIVE DETAILS
Project No
PAD 100
9187-10052
0
Scale
1:7.5



COMMERCIAL IN CONFIDENCE

1	9/11/8	PUMP AXES WERE IN LINE	AJH

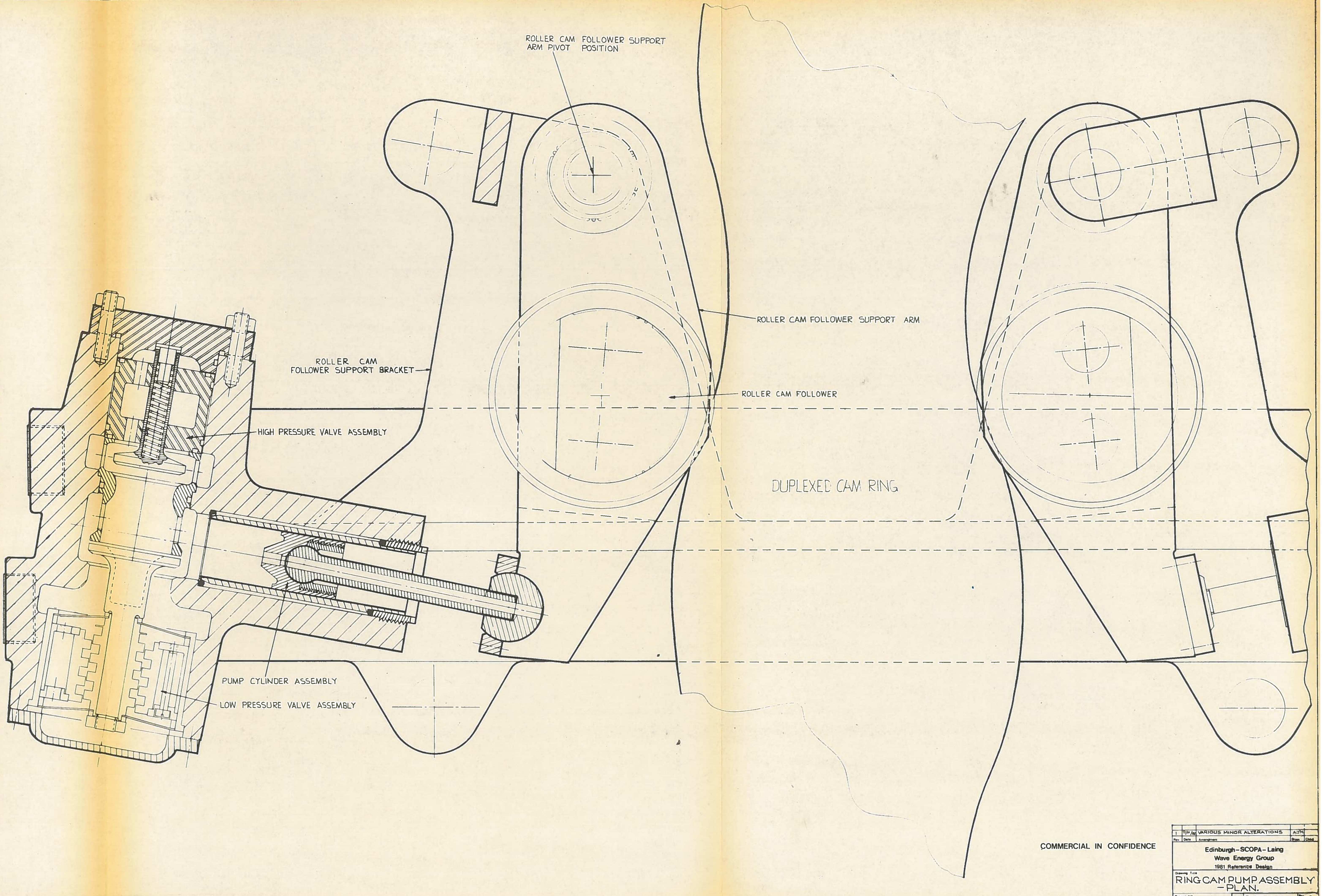
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

RING CAM PUMP ASSEMBLY
- ELEVATION -

PAD 100 | L | 9187-10060 | 1

Scale 1:2

AJ.H. 29-10-81



COMMERCIAL IN CONFIDENCE

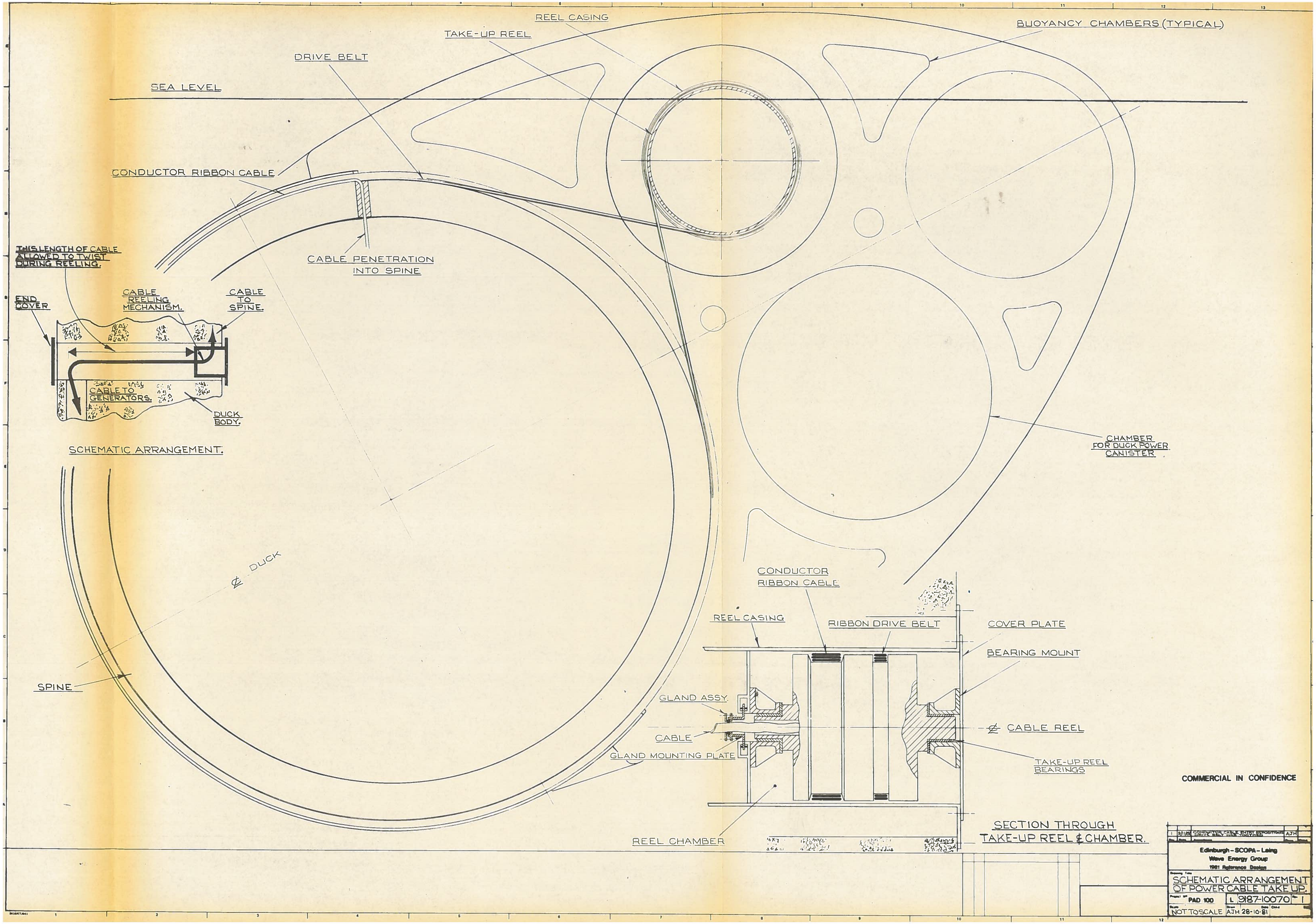
Rev	Date	Amendment	Drwn	Chkd
1	5/11/81	VARIOUS MINOR ALTERATIONS	AJH	

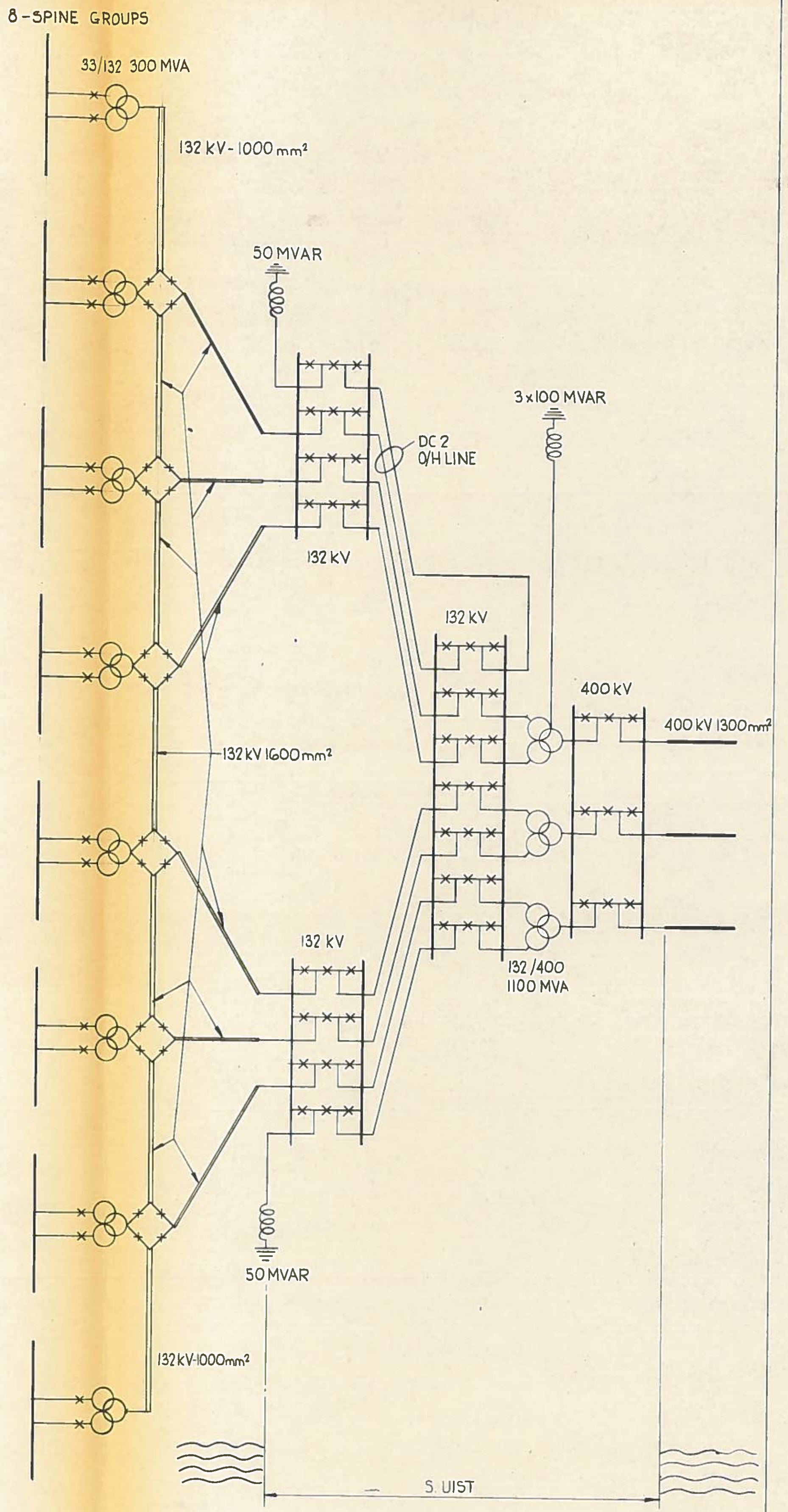
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title
RING CAM PUMP ASSEMBLY - PLAN.

Project No PAD 100 | L 9187-10061 | Rev 1

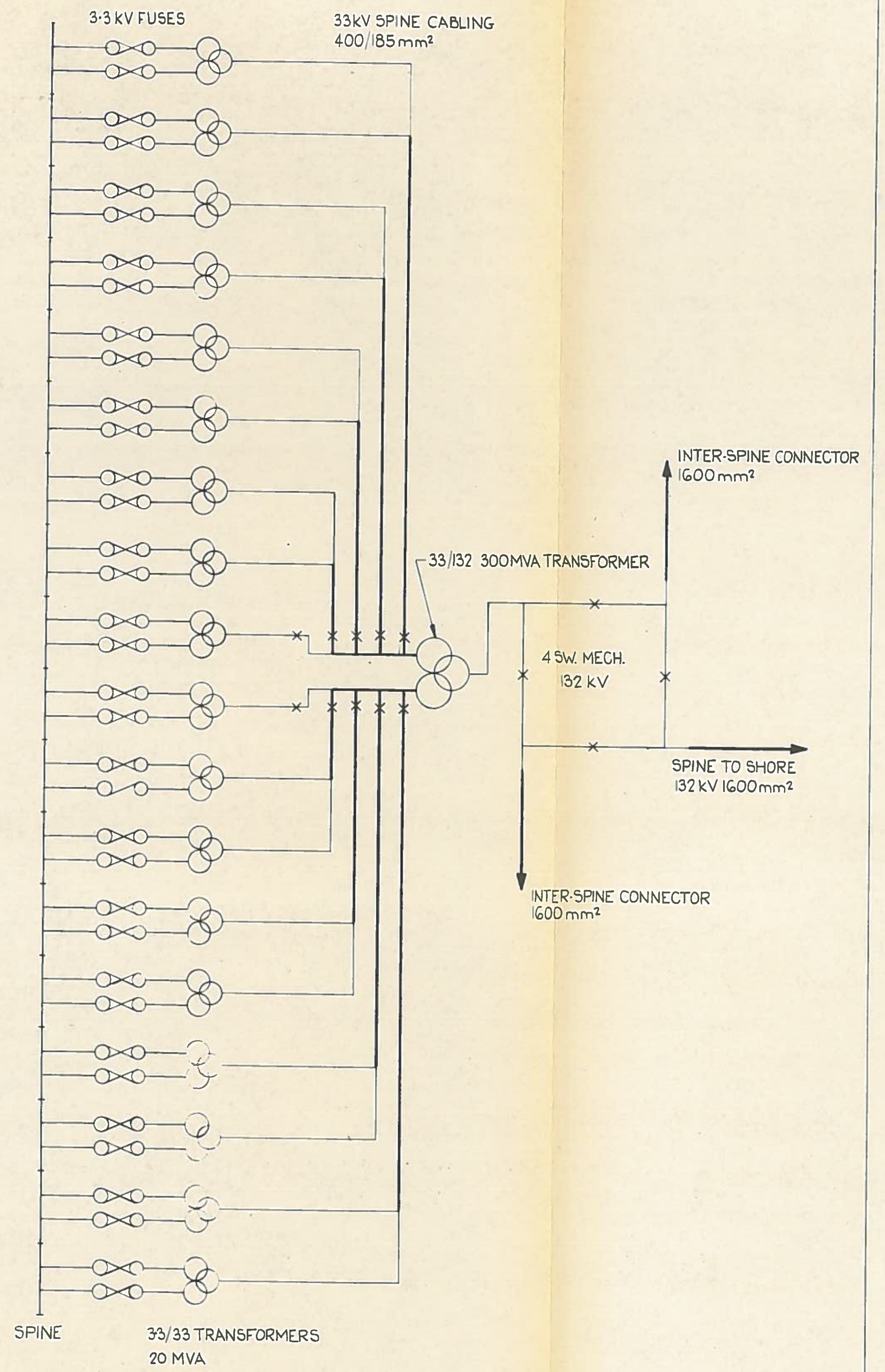
Scale FULL SIZE | Drwn P.D.A.H. | Date 2/11/81 | Chkd


REEL CASING
TAKE-UP REEL
BUOYANCY CHAMBERS (TYPICAL)
DRIVE BELT
SEA LEVEL
CONDUCTOR RIBBON CABLE
THIS LENGTH OF CABLE ALLOWED TO TWIST DURING REELING.
CABLE PENETRATION INTO SPINE
END COVER
CABLE REELING MECHANISM.
CABLE TO SPINE.
CABLE TO GENERATORS.
DUCK BODY.
SCHEMATIC ARRANGEMENT.
CHAMBER FOR DUCK POWER CANISTER
℄ DUCK
CONDUCTOR RIBBON CABLE
REEL CASING
RIBBON DRIVE BELT
COVER PLATE
BEARING MOUNT
SPINE
GLAND ASSY.
CABLE
℄ CABLE REEL
GLAND MOUNTING PLATE
TAKE-UP REEL BEARINGS
COMMERCIAL IN CONFIDENCE
SECTION THROUGH TAKE-UP REEL & CHAMBER.
REEL CHAMBER
1 9/11/81 AJH
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design
SCHEMATIC ARRANGEMENT OF POWER CABLE TAKE UP.
PAD 100
L 9187-10070
1
NOT TO SCALE
AJH
28-10-81



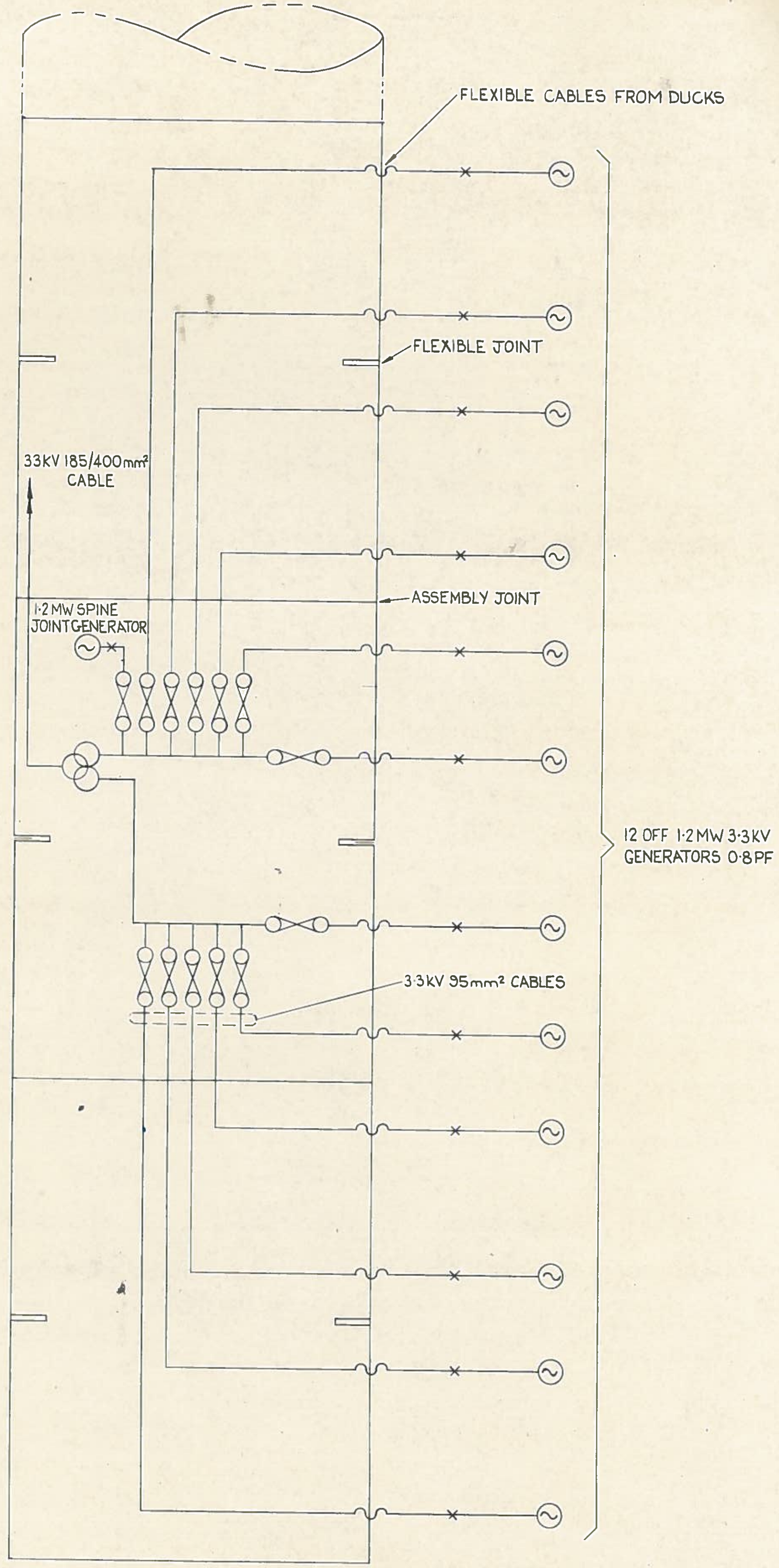
MAIN TRANSMISSION

FIG. 1



280 MW SPINE GROUP (4·86 km)
i.e. 18 SUBGROUPS

FIG. 2



15·6 MW SUB GROUP (270 m APPROX)

FIG. 3

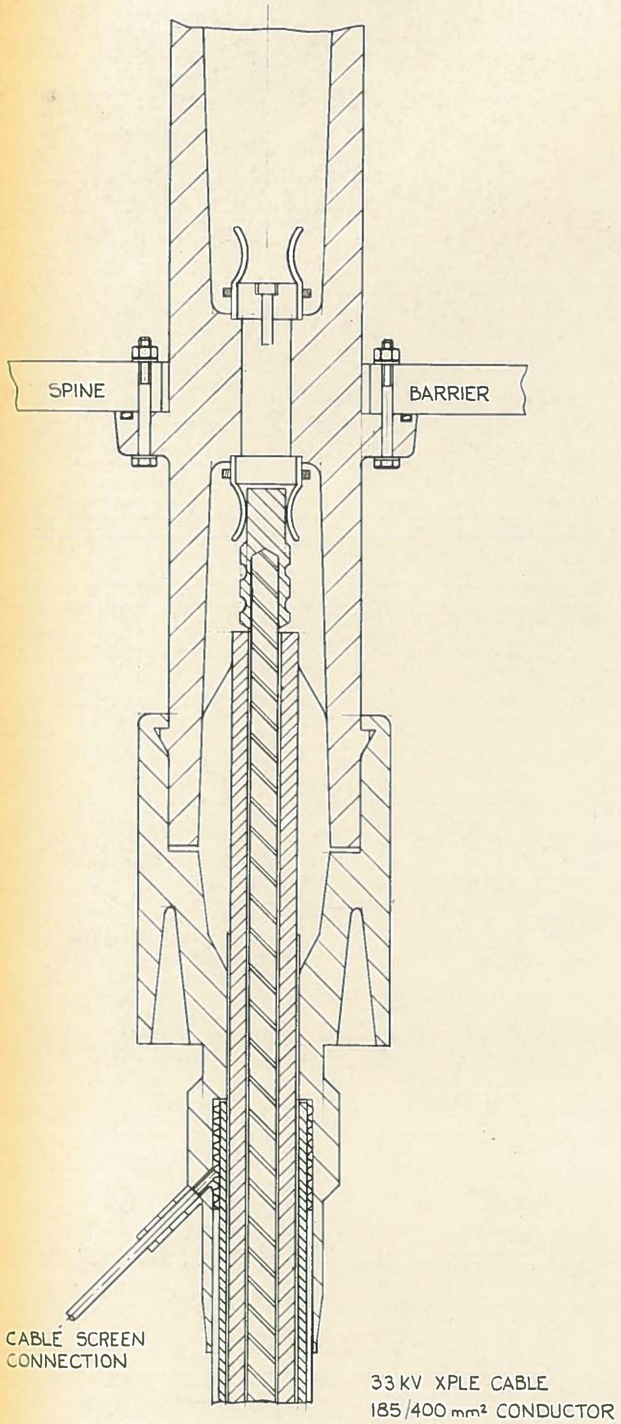
COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

SCHEMATIC ARRANGEMENT OF
POWER TRANSMISSION

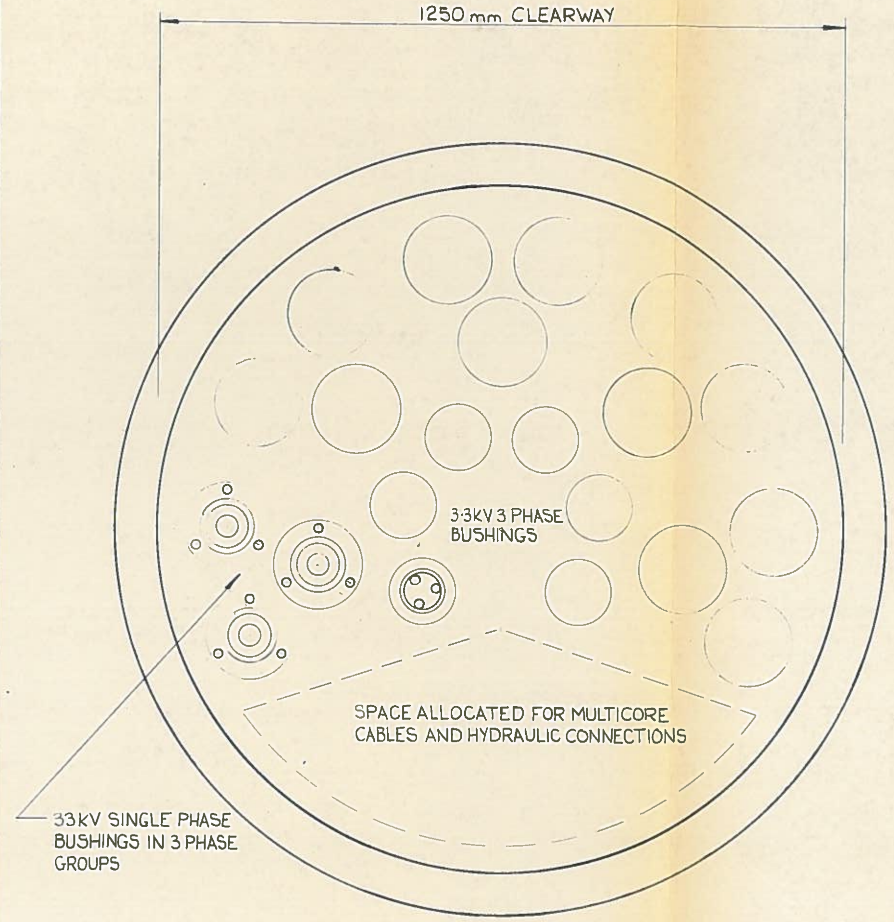
PAD 100 | L 9187-10071 | 0

NTS



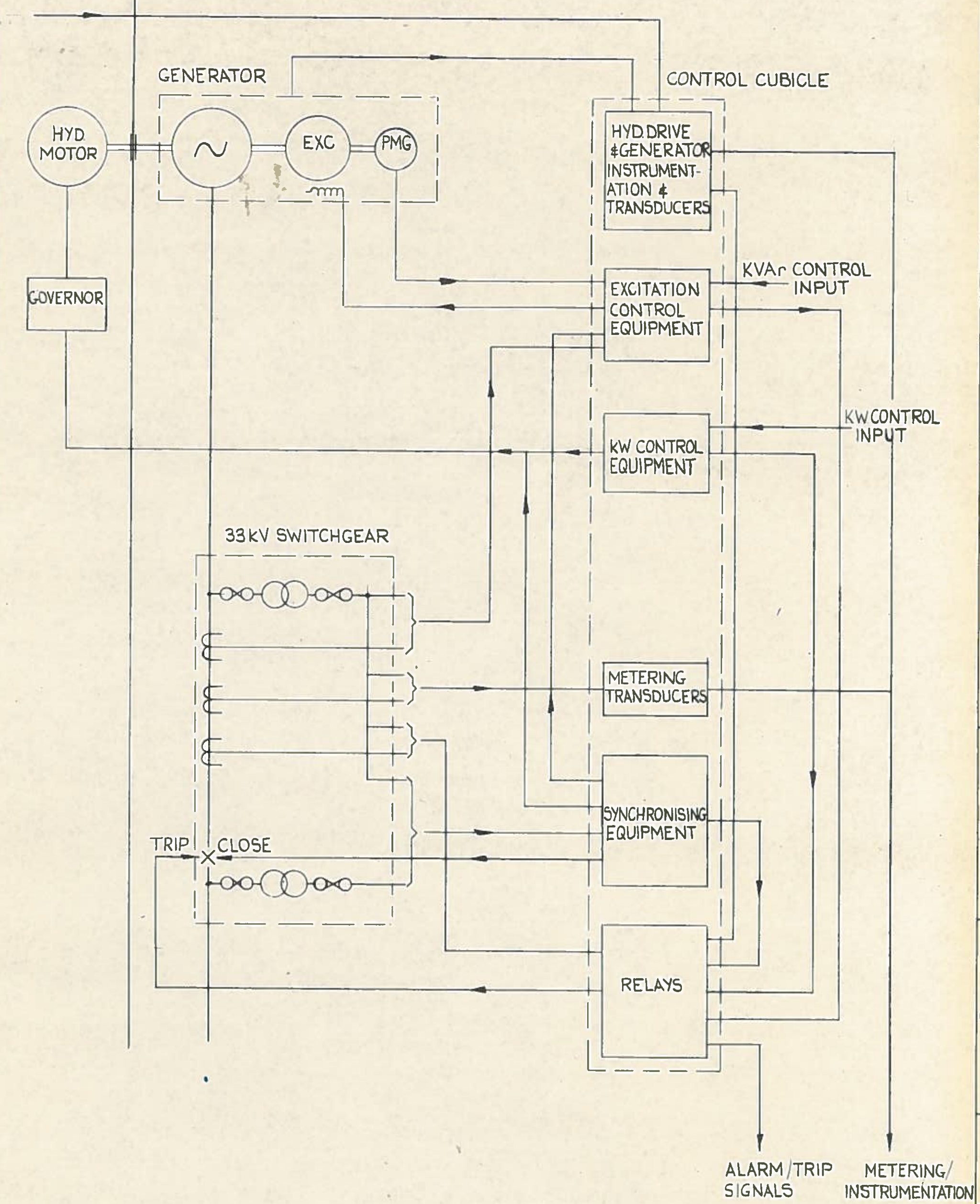
PROPOSED ARRANGEMENT OF 33kV CABLE CONNECTORS

FIG.5



PROPOSED ARRANGEMENT OF SPINE BARRIER BUSHING

FIG.6



SCHEMATIC DIAGRAM OF GENERATOR CONTROL, PROTECTION, METERING & INSTRUMENTATION CIRCUITS

FIG.7

COMMERCIAL IN CONFIDENCE

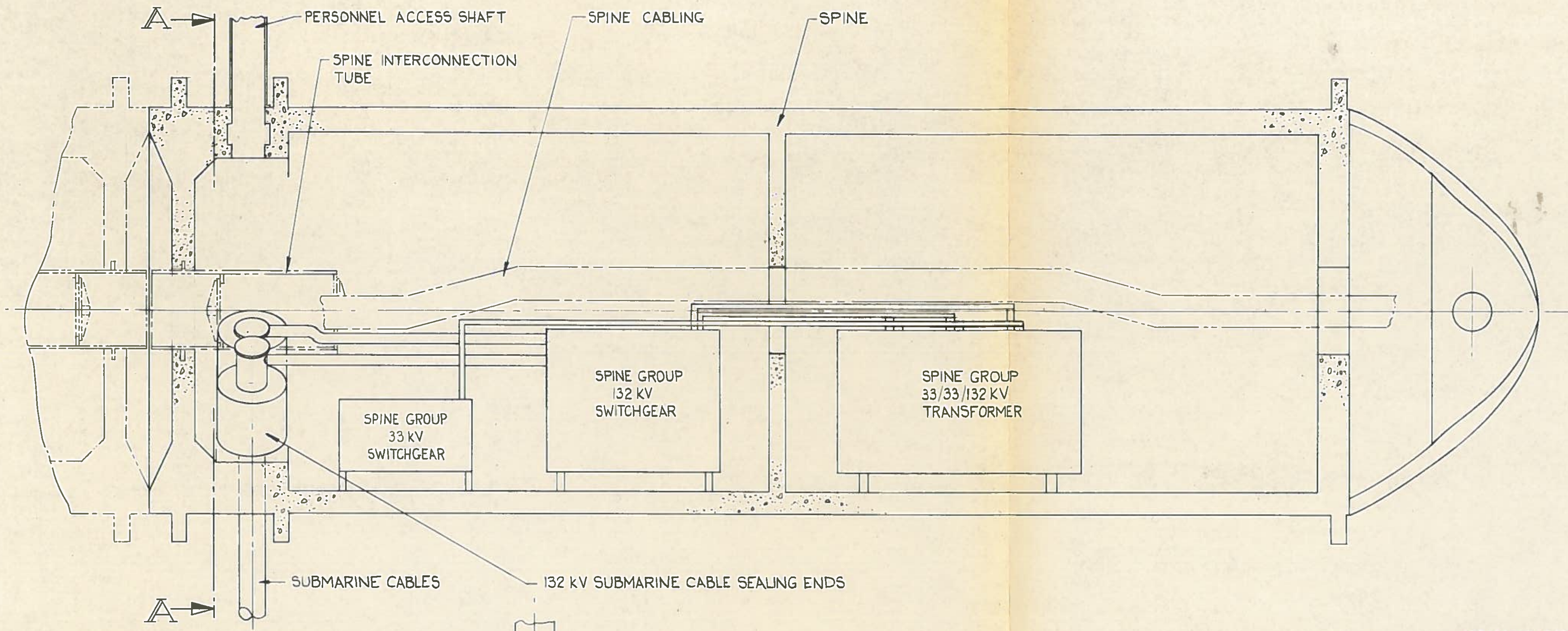
Rev	Date	Amendment	Drawn	Chkd

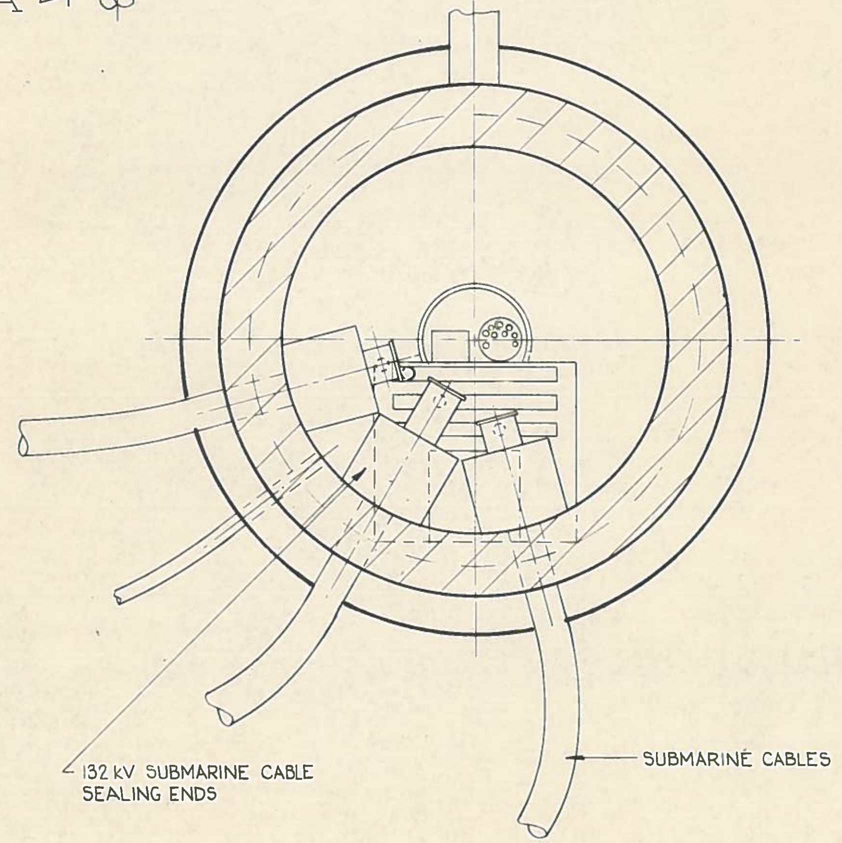
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SCHEMATIC DIAGRAM OF GENERATOR CIRCUITS INCLUDING CABLE DETAILS

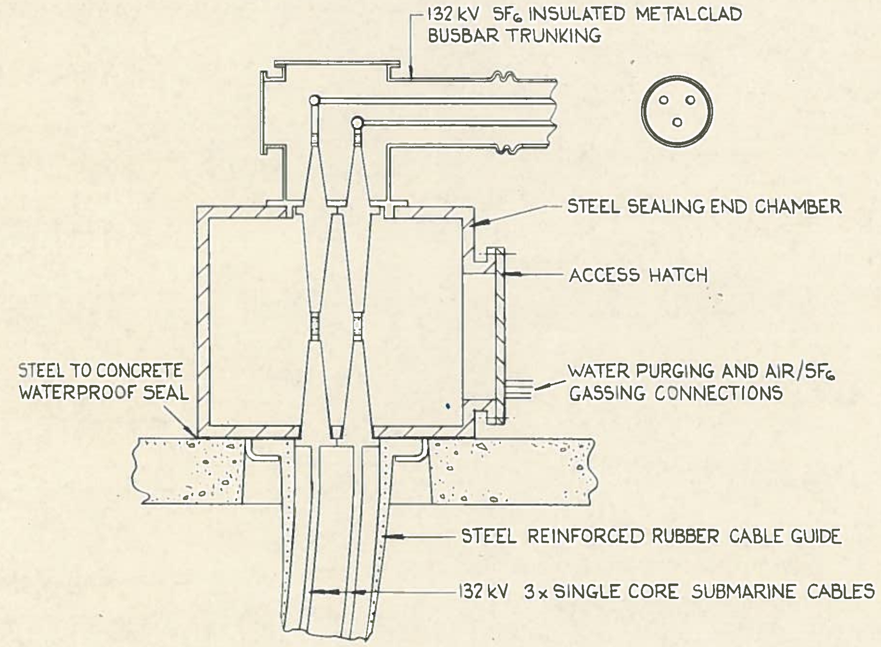
Project No: PAD 100 | L 9187-10072 | No. O

Scale: NTS | Drawn: MSG | Date: Nov81





SECTION A-A

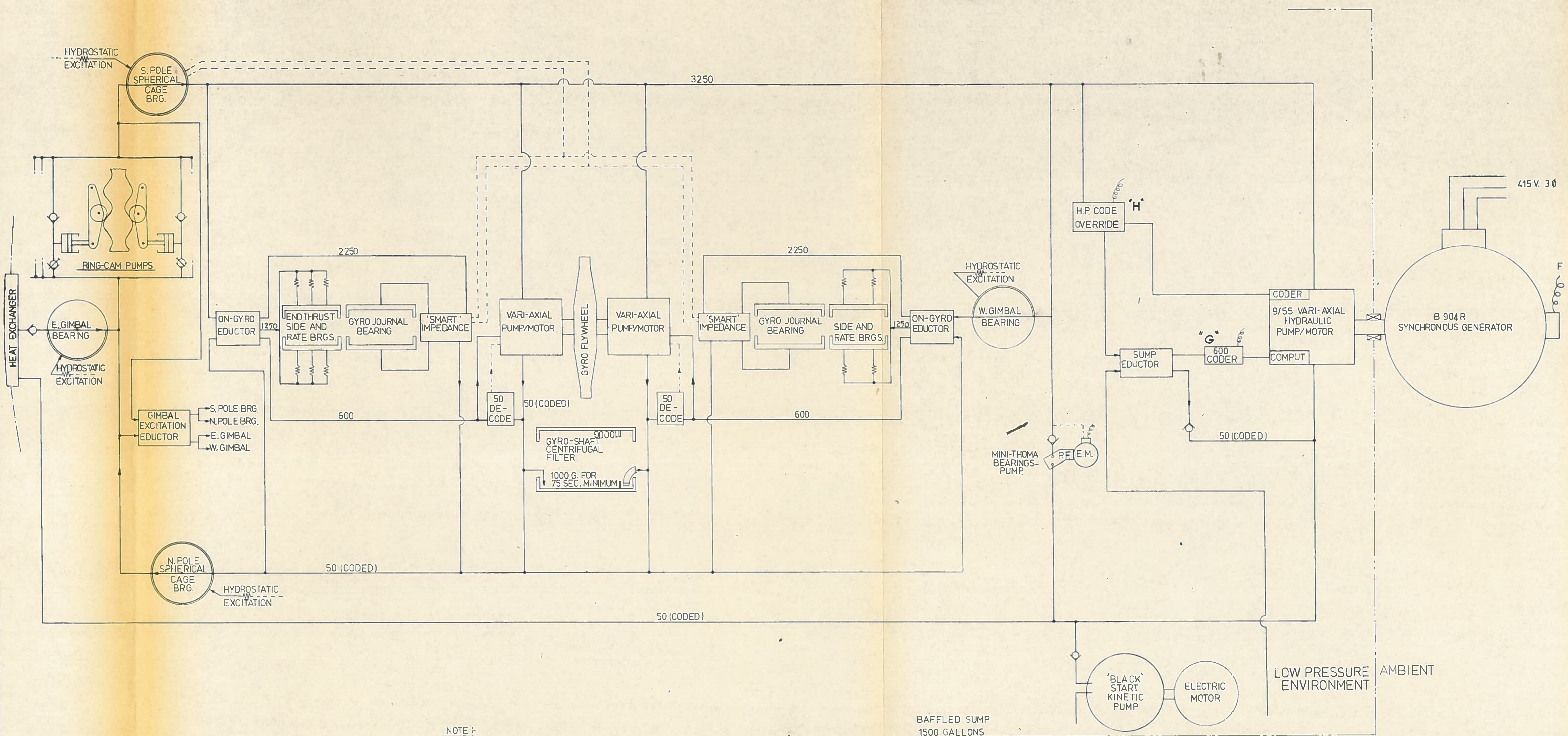


PROPOSED ARRANGEMENT OF 132 kV SUBMARINE CABLE SEALING ENDS

SCALE 1:25

COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing Wave Energy Group 1981 Reference Design	
Drawing Title: ARRANGEMENT OF SPINE GROUP ELECTRICAL EQUIPMENT	
Project No. PAD 100	L 9187-10073 Rev. 0
Scale 1:75 & 1:25	Drawn MSG Date Nov81



NOTE:-

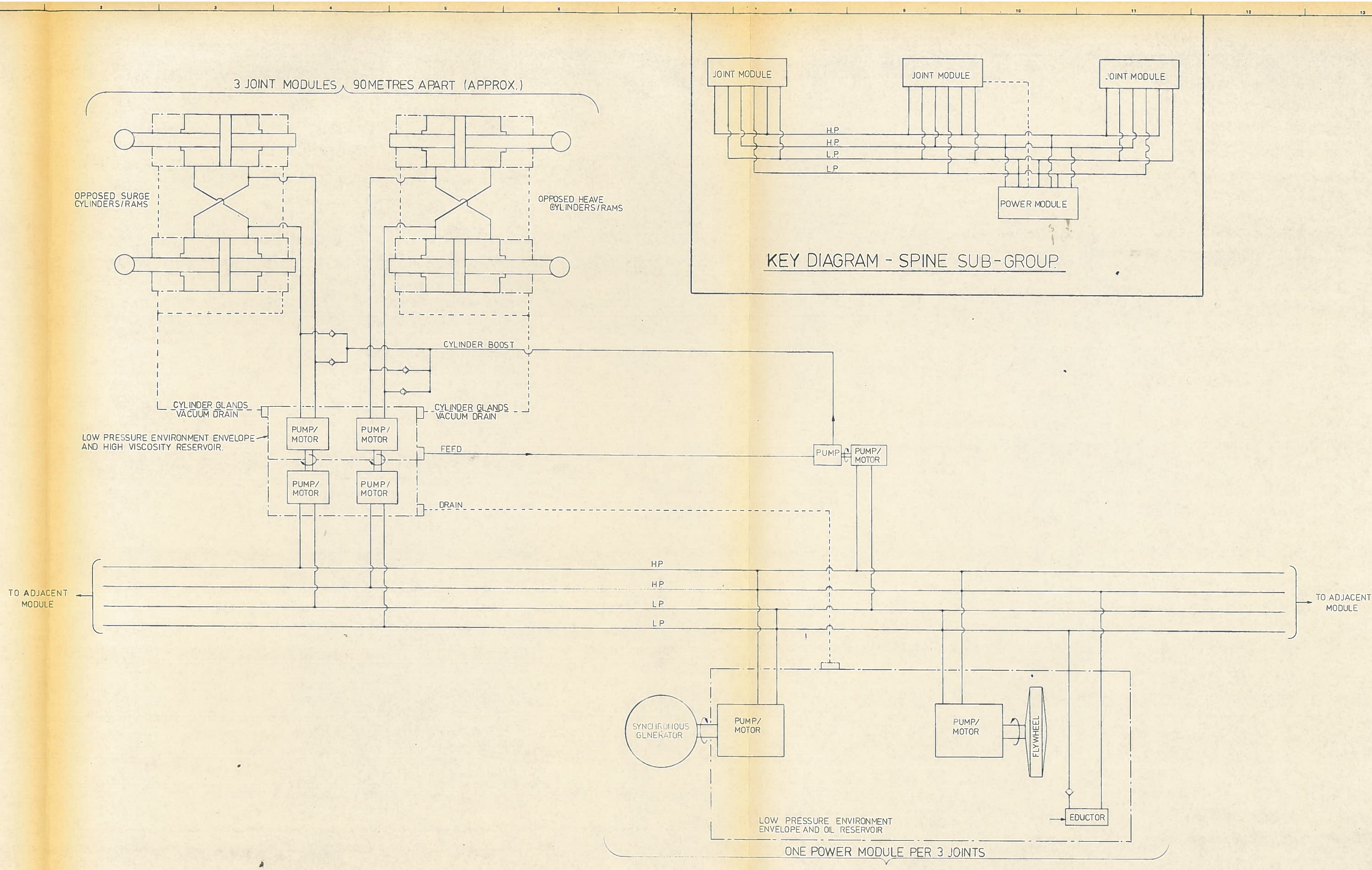
NUMBERS THUS:- 50 (CODED) ARE LINE PRESSURES IN P.S.I. AT MAXIMUM POWER (VARYING AS POWER^2)

G 600 CODE INCREASES/DECREASES SYNC-MOTOR STROKE DISPLACEMENT.

H H.P. CODE (VIA 50 CODED) CONTROLS GYRO SPEED AND SYSTEMS PRESSURE.

COMMERCIAL IN CONFIDENCE

	Edinburgh - SCOPA - Laing Wave Energy Group 1991 Reference Design
Drawing Title	DUCK POWER CANISTER HYDRAULIC CIRCUIT.
Project No	PAD 100	L 918/10080 D
Scale	N.T.S.	NE MOATE



COMMERCIAL IN CONFIDENCE

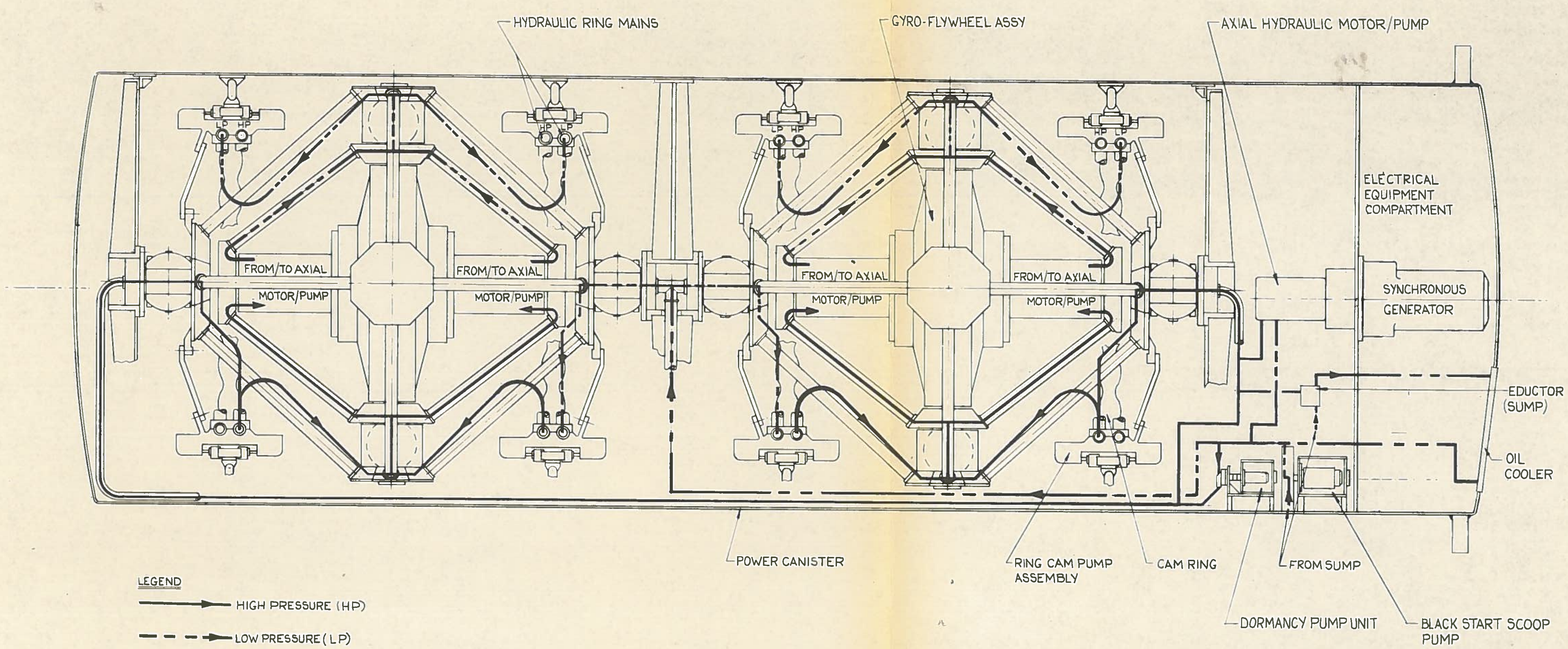
1	16-11-81	KEY DIAGRAM ADDED, SUB-GROUP COMPRISE 6 MODULES NOW 3

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Referenced Design

Drawing Title SPINE JOINT HYDRAULIC CIRCUIT

Project No PAD 100 | L 9187-10081 | Rev 1

Scale N.T.S. | Drawn N.E.MOATE



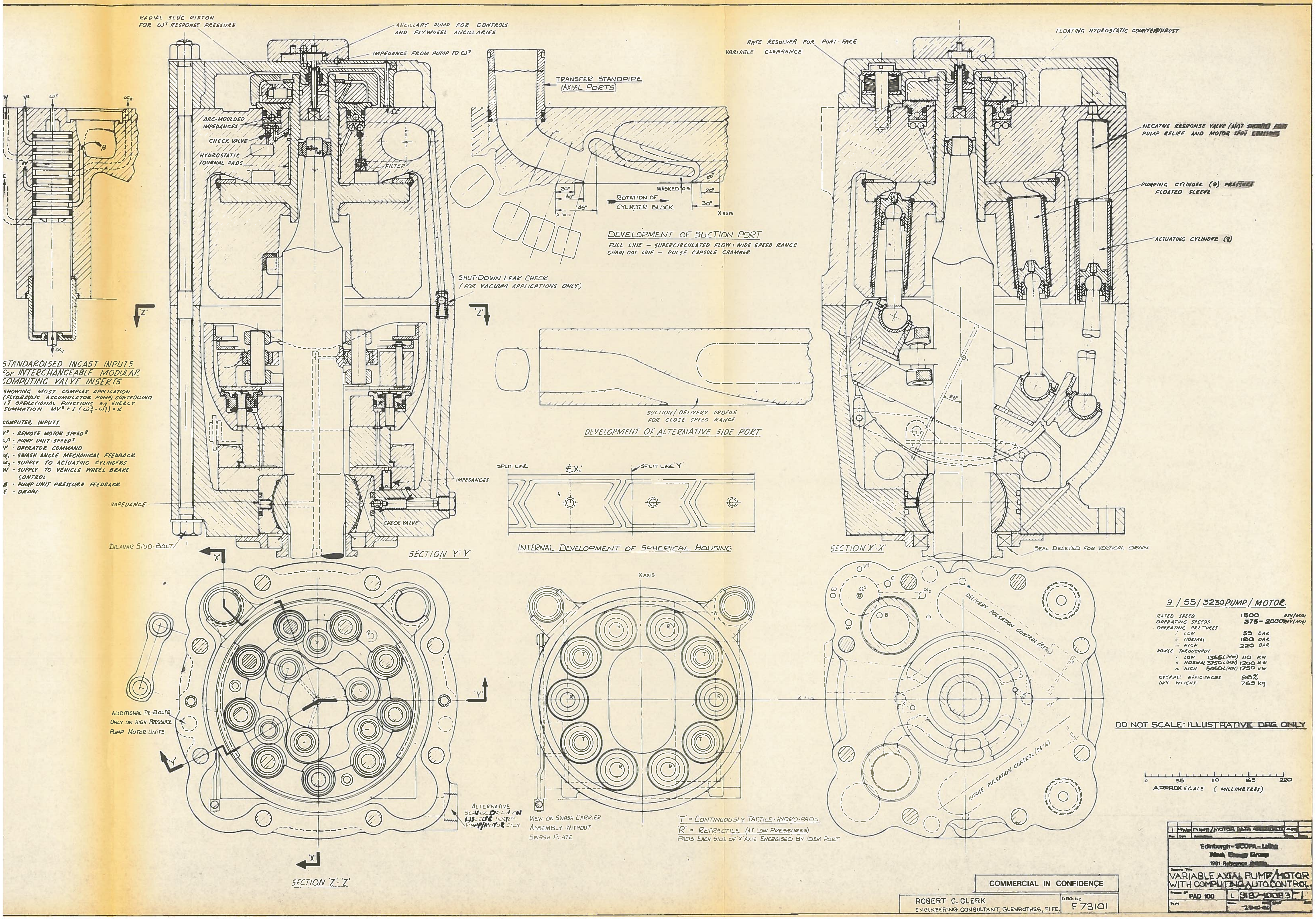
COMMERCIAL IN CONFIDENCE

Rev	Date	Amendment	Drawn	Chkd
1	Nov81	ELECT. EQUIP. COMPARTMENT ADDED	MSG	

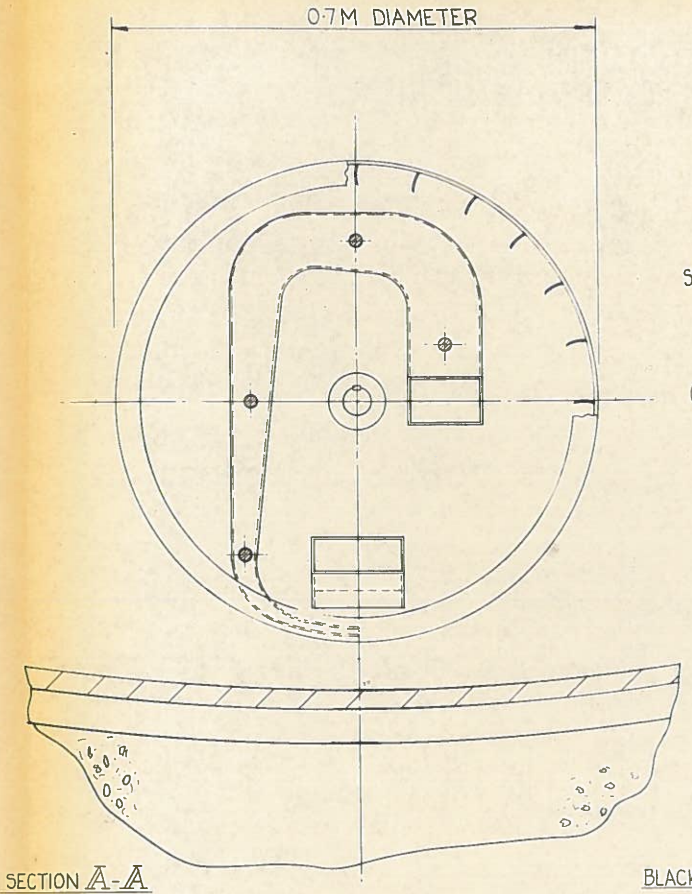
Edinburgh – SCOPA – Laing
Wave Energy Group
1981 Reference Design

Drawing Title: ARRANGEMENT OF DUCK POWER CANISTER HYDRAULIC MAINS

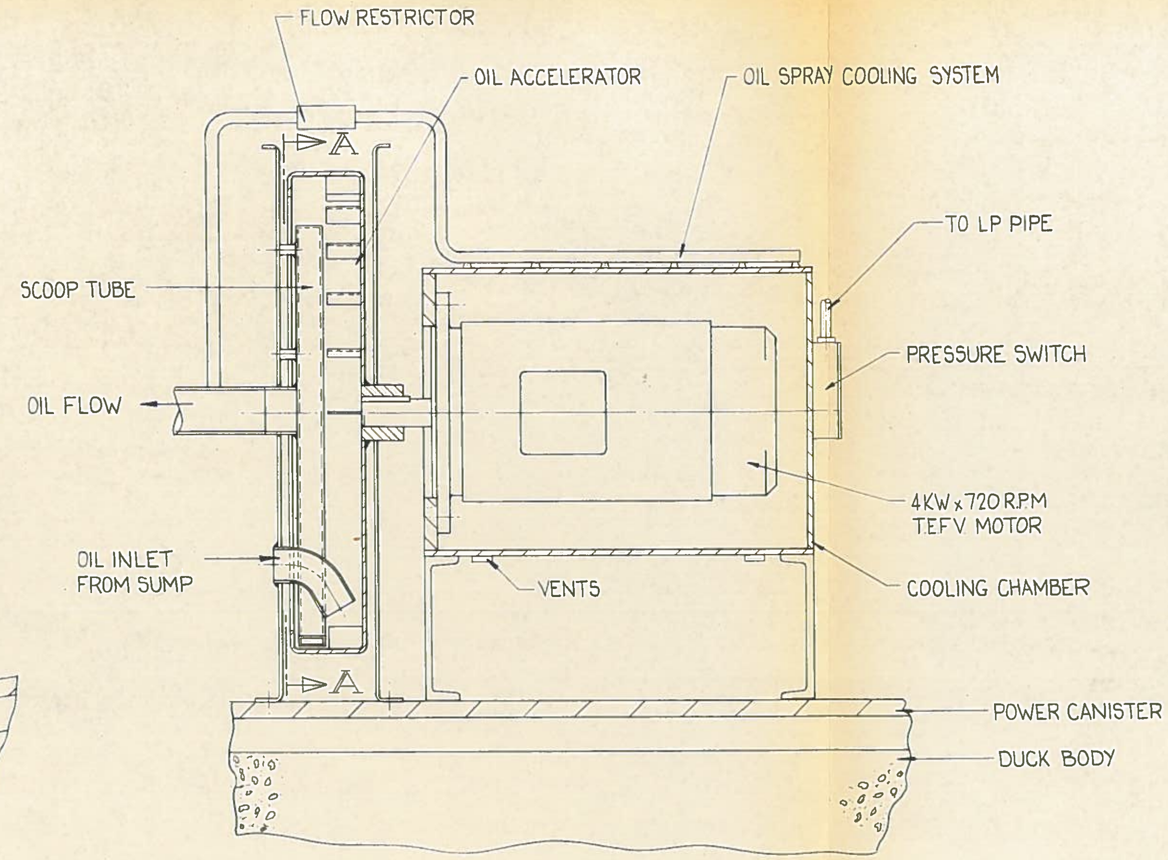
Project No PAD 100	L	9187-10082	Rev 1
Scale 1:25	Drawn M.S.G	Date OCT 81	Chkd / Date



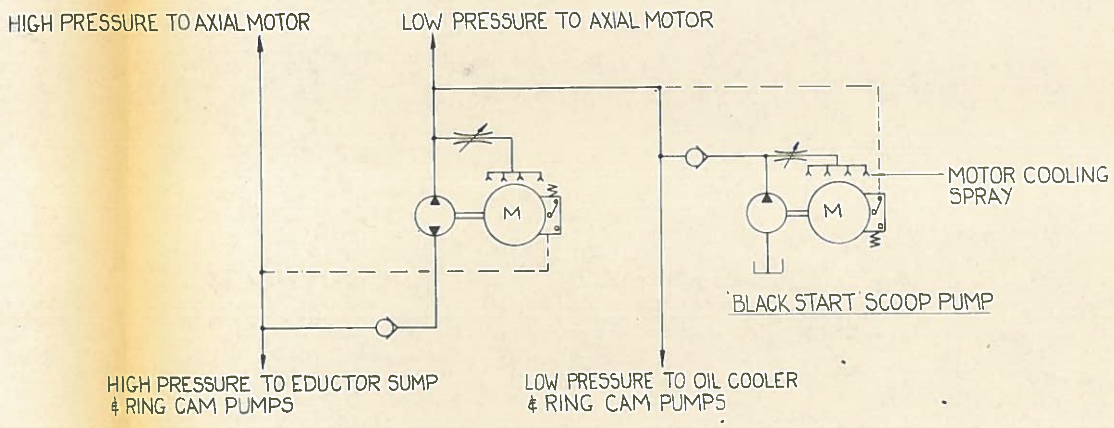
RADIAL SLUG PISTON FOR ω² RESPONSE PRESSURE
ANCILLARY PUMP FOR CONTROLS AND FLYWHEEL ANCILLARIES
IMPEDANCE FROM PUMP TO ω²
ARC-MOULDED IMPEDANCES
CHECK VALVE
HYDROSTATIC JOURNAL PADS
FILTER
TRANSFER STANDPIPE (AXIAL PORTS)
ROTATION OF CYLINDER BLOCK
MASKED
X AXIS
DEVELOPMENT OF SUCTION PORT
FULL LINE - SUPERCIRCULATED FLOW: WIDE SPEED RANGE
CHAIN DOT LINE - PULSE CAPSULE CHAMBER
SHUT-DOWN LEAK CHECK (FOR VACUUM APPLICATIONS ONLY)
'Z'
'Z'
SUCTION/DELIVERY PROFILE FOR CLOSE SPEED RANGE
DEVELOPMENT OF ALTERNATIVE SIDE PORT
SPLIT LINE
SPLIT LINE 'Y'
IMPEDANCES
IMPEDANCE
CHECK VALVE
DILAVAR STUD-BOLT
SECTION 'Y'-'Y'
INTERNAL DEVELOPMENT OF SPHERICAL HOUSING
SECTION 'X'-'X'
SEAL DELETED FOR VERTICAL DRAIN
RATE RESOLVER FOR PORT FACE VARIABLE CLEARANCE
FLOATING HYDROSTATIC COUNTERTHRUST
NEGATIVE RESPONSE VALVE (NOT SHOWN) FOR PUMP RELIEF AND MOTOR SPIN LIMITING
PUMPING CYLINDER (9) PRESSURE FLOATED SLEEVE
ACTUATING CYLINDER (2)
STANDARDISED INCAST INPUTS for INTERCHANGEABLE MODULAR COMPUTING VALVE INSERTS
SHOWING MOST COMPLEX APPLICATION (FLYDRAULIC ACCUMULATOR PUMP) CONTROLLING 17 OPERATIONAL FUNCTIONS e.g ENERGY SUMMATION $MV^2 + I(\omega_2^2 - \omega_1^2) = K$
COMPUTER INPUTS
V^2 - REMOTE MOTOR SPEED²
ω^2 - PUMP UNIT SPEED²
Ψ - OPERATOR COMMAND
α_1 - SWASH ANGLE MECHANICAL FEEDBACK
α_2 - SUPPLY TO ACTUATING CYLINDERS
W - SUPPLY TO VEHICLE WHEEL BRAKE CONTROL
β - PUMP UNIT PRESSURE FEEDBACK
ϵ - DRAIN
'X'
'Y'
ADDITIONAL TIE-BOLTS ONLY ON HIGH PRESSURE PUMP MOTOR UNITS
ALTERNATIVE SCAVENGE DRAIN ON DISCRETE HOUSED PUMP/MOTOR ONLY
SECTION 'Z'-'Z'
VIEW ON SWASH CARRIER ASSEMBLY WITHOUT SWASH PLATE
X AXIS
'T' = CONTINUOUSLY TACTILE HYDRO-PADS
'R' = RETRACTILE (AT LOW PRESSURES)
PADS EACH SIDE OF 'X' AXIS ENERGISED BY IDEM PORT
DELIVERY PULSATION CONTROL
INTAKE PULSATION CONTROL
9 / 55 / 3230 PUMP / MOTOR
RATED SPEED 1500 REV/MIN
OPERATING SPEEDS 375 - 2000 REV/MIN
OPERATING PRESSURES
LOW 55 BAR
NORMAL 180 BAR
HIGH 220 BAR
POWER THROUGHPUT
LOW 1365 L(MIN) 110 KW
NORMAL 3750 L(MIN) 1200 KW
HIGH 5460 L(MIN) 1750 KW
OVERALL EFFICIENCIES 98%
DRY WEIGHT 765 kg
DO NOT SCALE: ILLUSTRATIVE DRG ONLY
0 55 110 165 220
APPROX SCALE (MILLIMETRES)
COMMERCIAL IN CONFIDENCE
ROBERT C. CLERK
ENGINEERING CONSULTANT, GLENROTHES, FIFE.
DRG. No F 73101
PUMP/MOTOR DATA
Edinburgh-SCOPA
Group
1981 Reference
VARIABLE AXIAL PUMP/MOTOR WITH COMPUTING AUTO CONTROL.
PAD 100



SECTION A-A

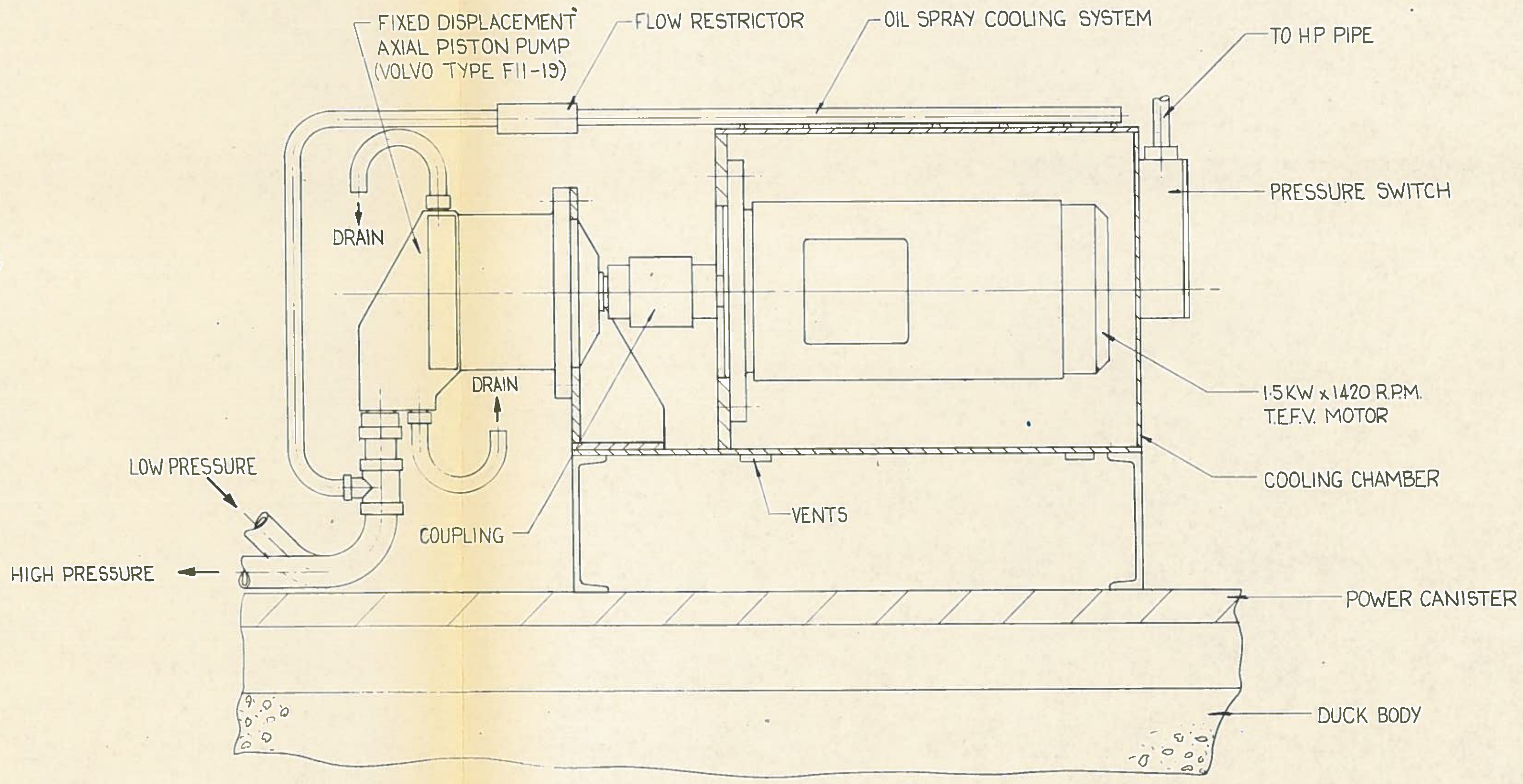


BLACK START SCOOP PUMP
SCALE 1:4



DORMANCY PUMP UNIT

PART CIRCUIT DIAGRAM FOR BLACK START EQUIPMENT



DORMANCY PUMP UNIT
SCALE 1:2

COMMERCIAL IN CONFIDENCE

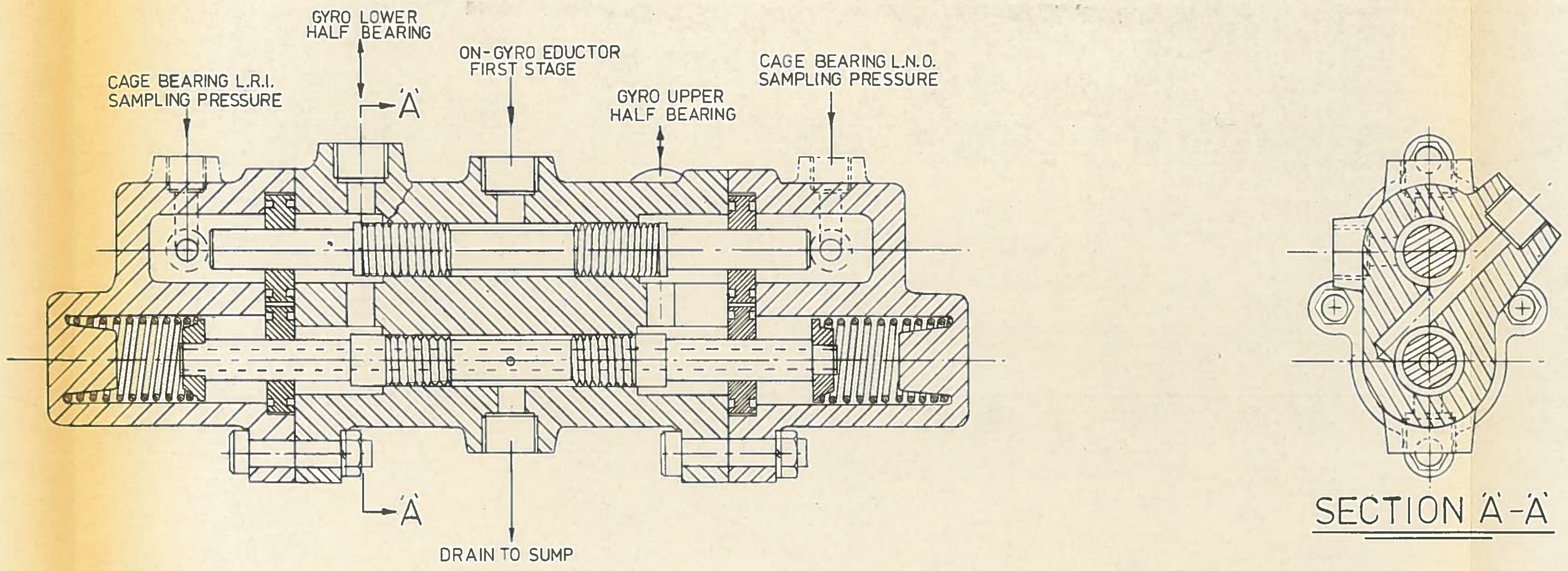
Rev	Date	Amendment	
1	Nov 81	TITLE CHANGED	MSG

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

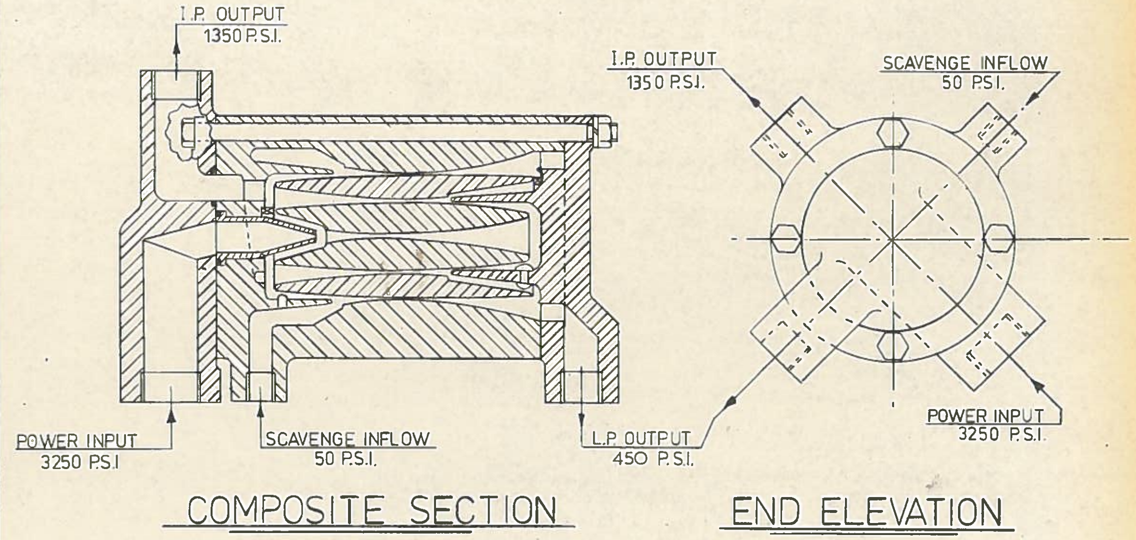
Drawing Title: START UP AUXILIARIES

Project No: PAD 100 | L 9187-10084 | Rev 1

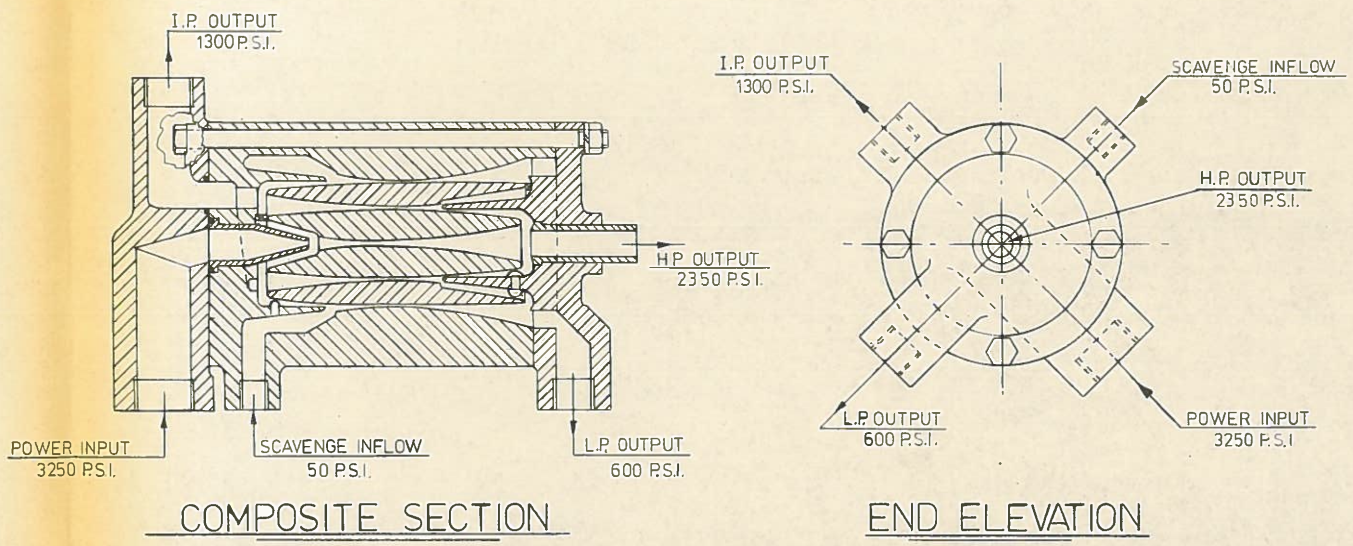
Scale: SEE DRAWING | Drawn: MSG | Date: OCT 81 | Chkd:



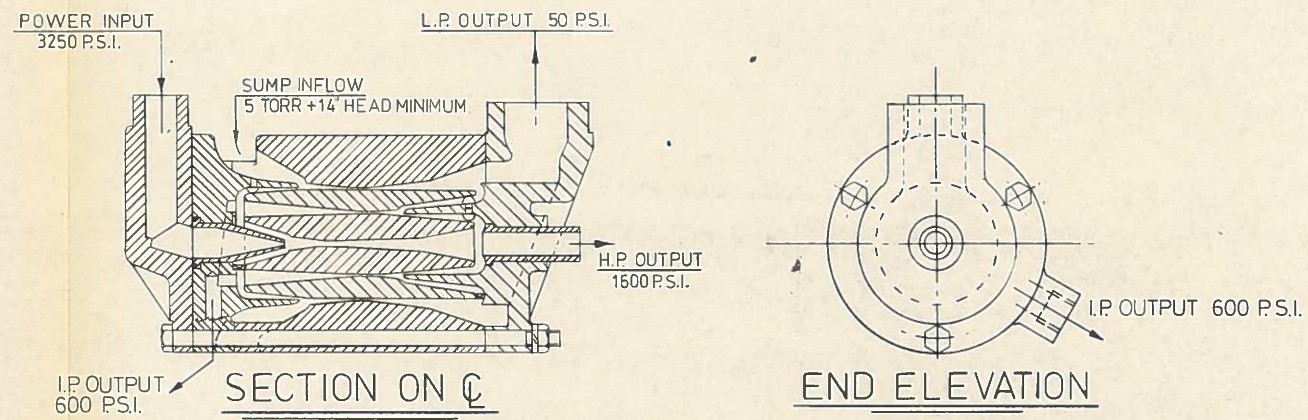
SMART IMPEDANCE - GYRO BEARINGS (2 PER GYRO)



GIMBAL EXCITATION EDUCTOR PUMP (1 PER GYRO)



ON-GYRO TRIPLE OUTPUT EDUCTOR PUMP (2 PER GYRO)



OFF-GYRO THREE STAGE EDUCTOR PUMP AND SCAVENGE (1 PER. POWER CANISTER)

THIRD ANGLE PROJECTION

COMMERCIAL IN CONFIDENCE

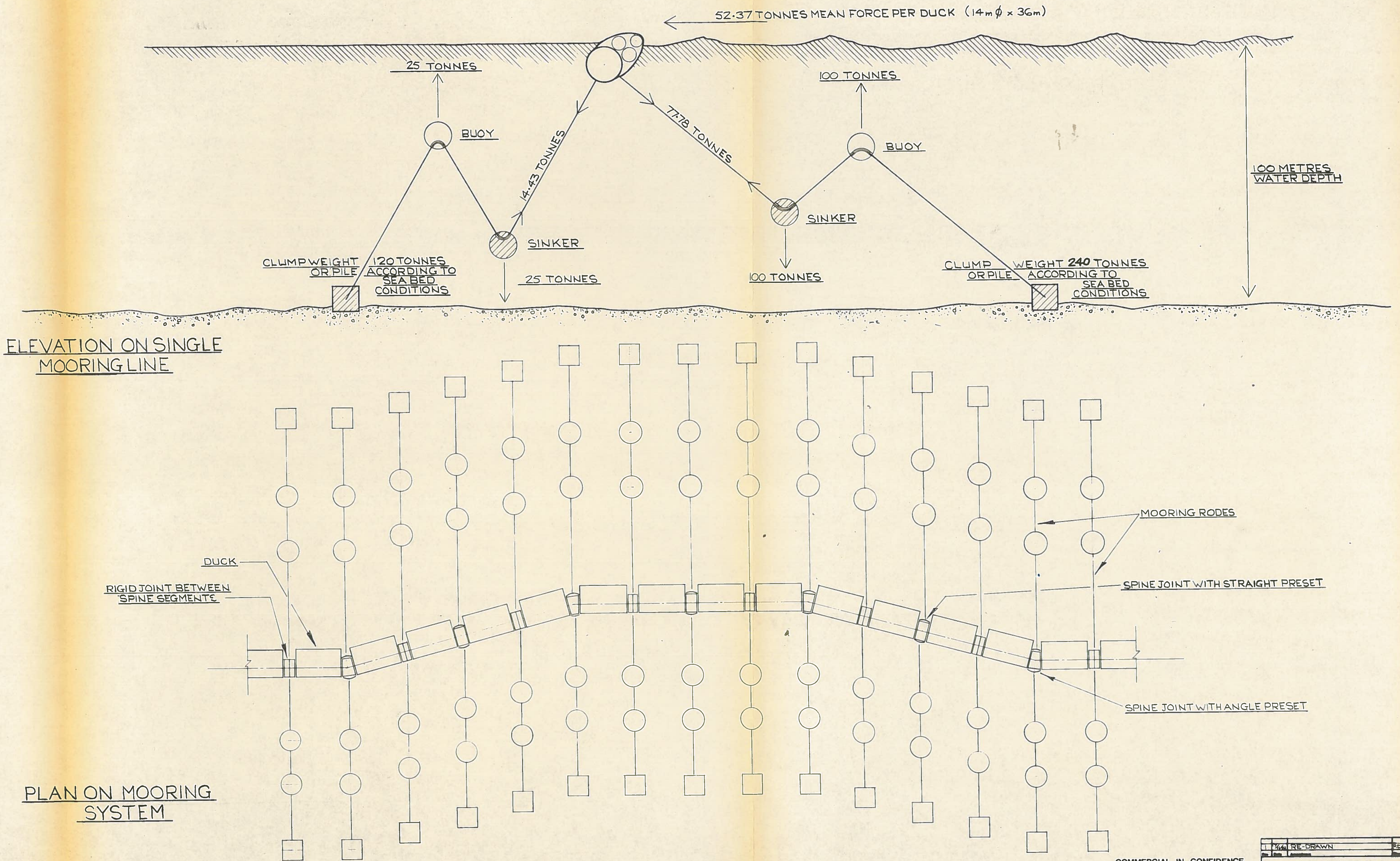
Rev	Date	Amendment	Drwn	Chkd
2	25.11.81	VARIOUS MODS. TO INPUT & OUTPUT PRESS.	M.A.S.	
1	16.11.81	GIMBAL EXCITATION EDUCTOR PUMP ADDED	M.E.M.	

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SMART IMPEDANCE AND EDUCTOR PUMPS

Project No: PAD 100 | L 9187-10085 | Rev. 2

Scale: FULL SIZE | Drwn: NE MOATE | Date: Nov 81 | Chkd



COMMERCIAL IN CONFIDENCE

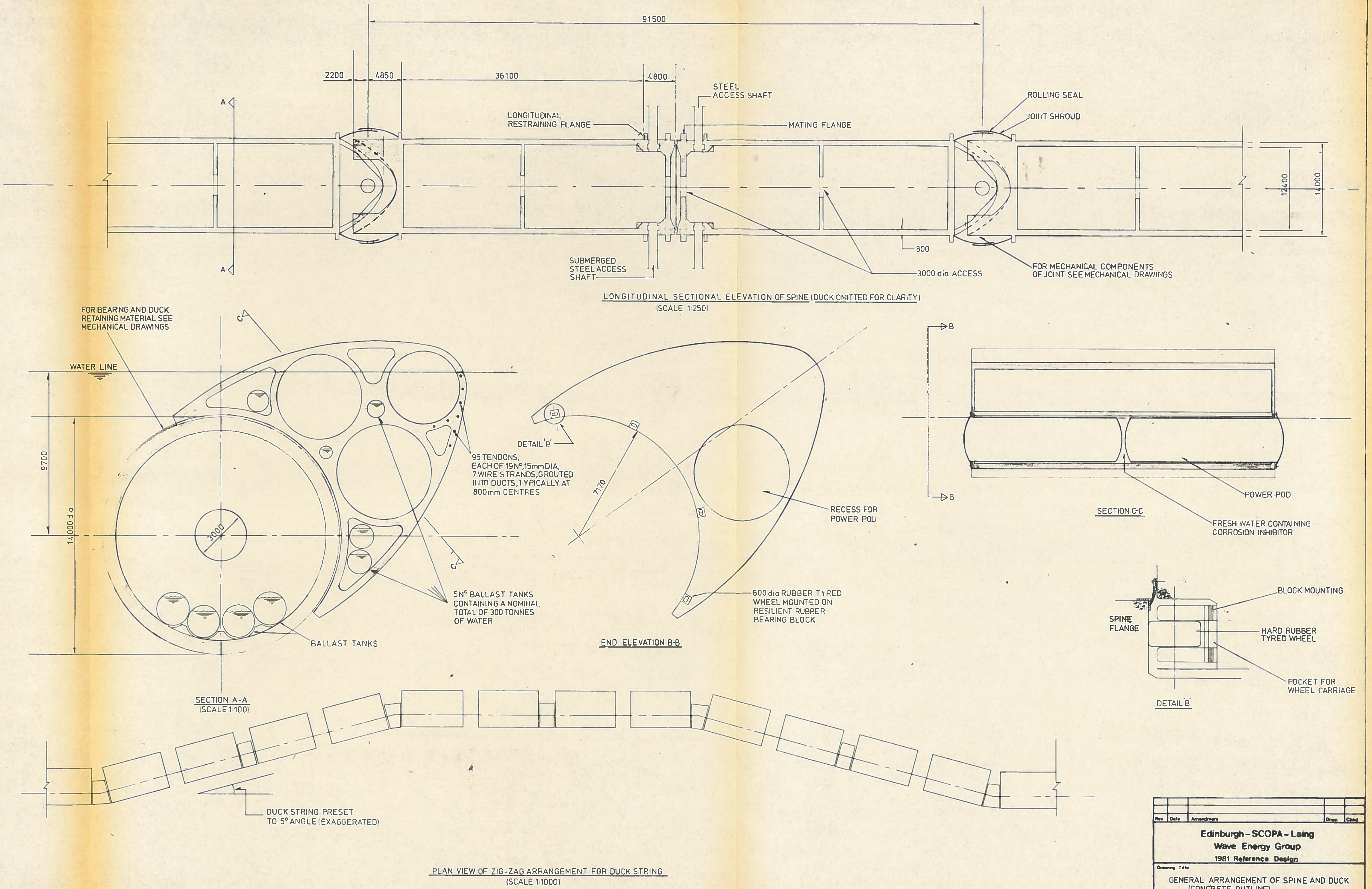
1	RE-DRAWN	P.D.H.	

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

SCHEMATIC ARRANGEMENT OF MOORING SYSTEM

PAD 100 | L 9187-10090 | 1

N.T.S. | P.D.H. 17-1-81



Rev	Date	Amendment	Drwn	Chkd

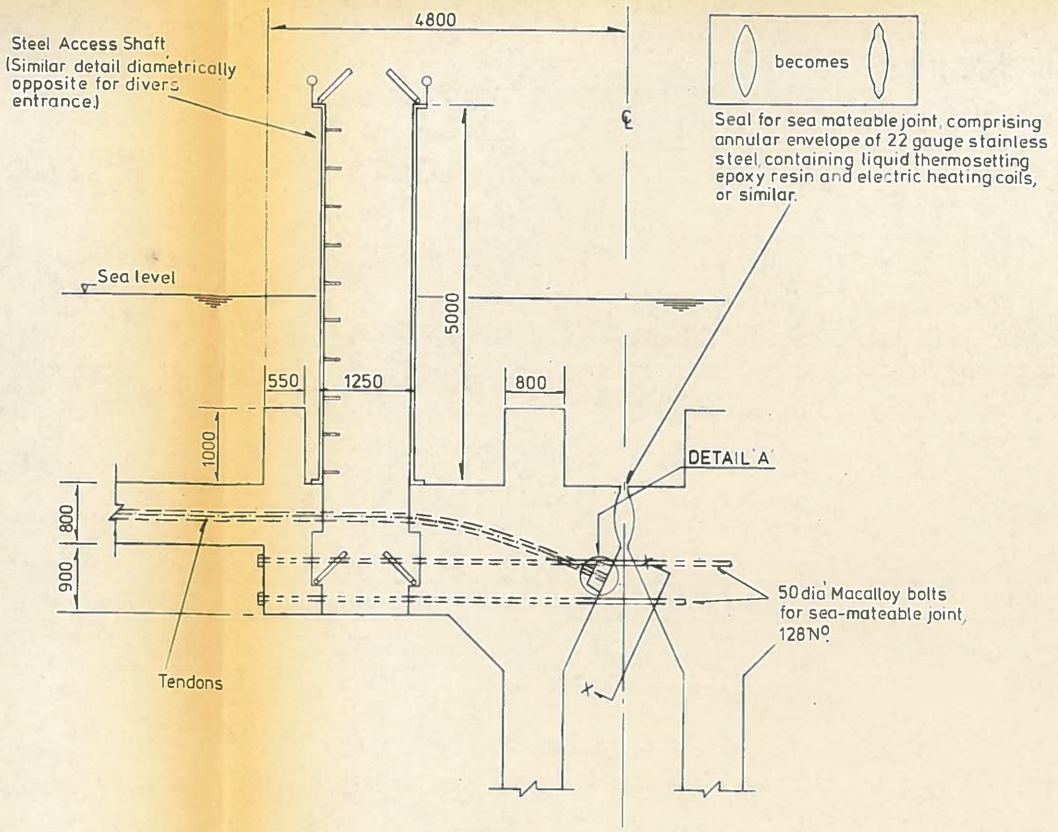
Edinburgh – SCOPA – Laing
Wave Energy Group
1981 Reference Design

Drawing Title
GENERAL ARRANGEMENT OF SPINE AND DUCK
(CONCRETE OUTLINE)

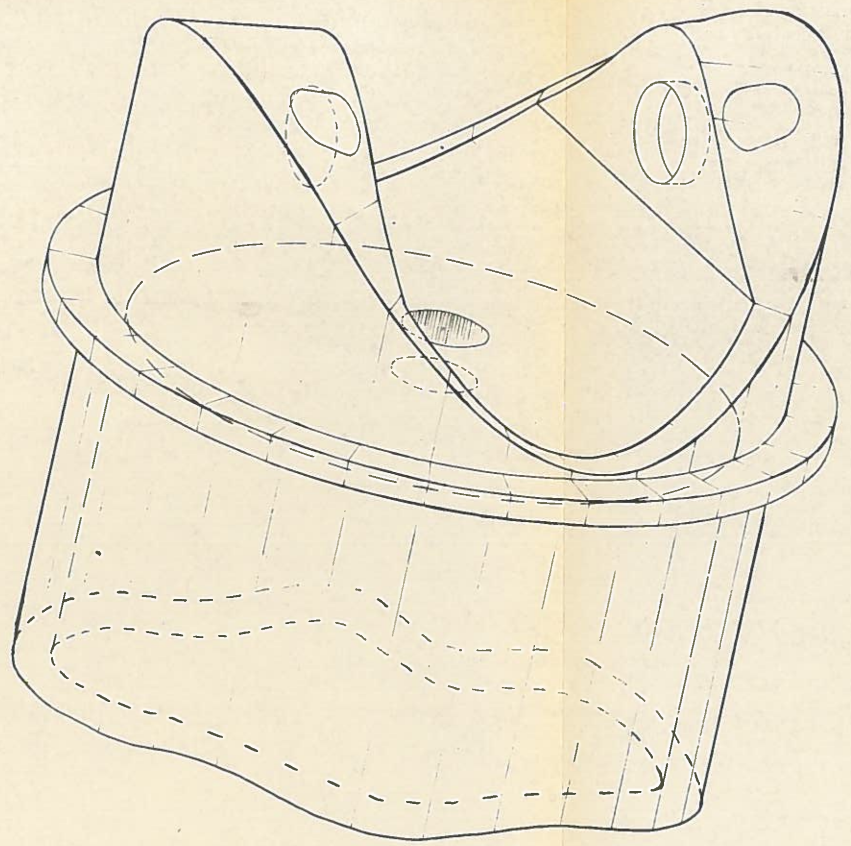
Project Nº		Rev
PAD 100	L 9187-10101	0

Scale	Drwn	Date	Chkd	Date
1:250 1:100 1:1000	JPL	OCT 81		

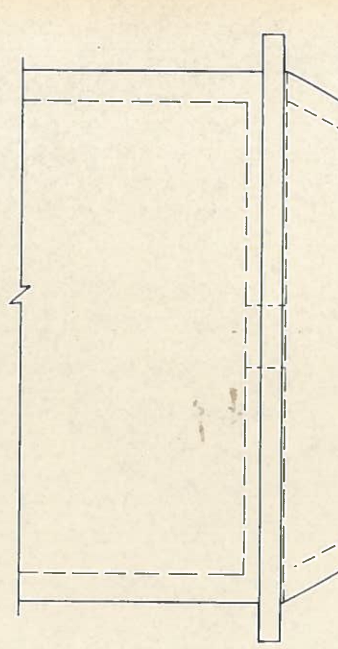
COMMERCIAL IN CONFIDENCE



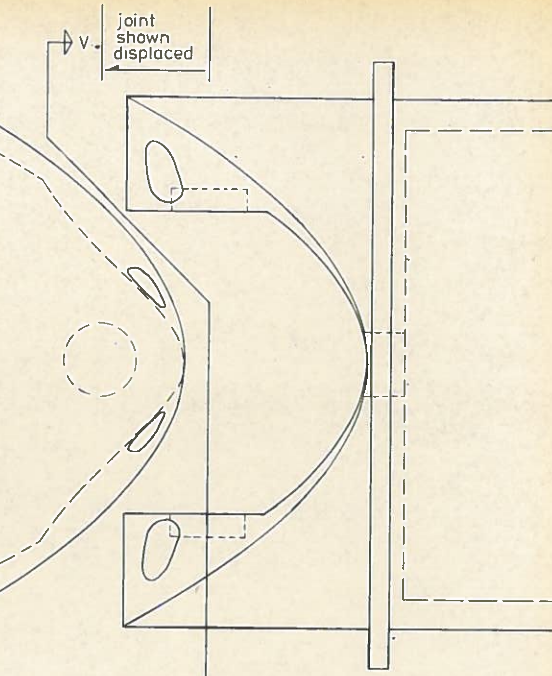
DETAILS AT CENTRE OF SPINE



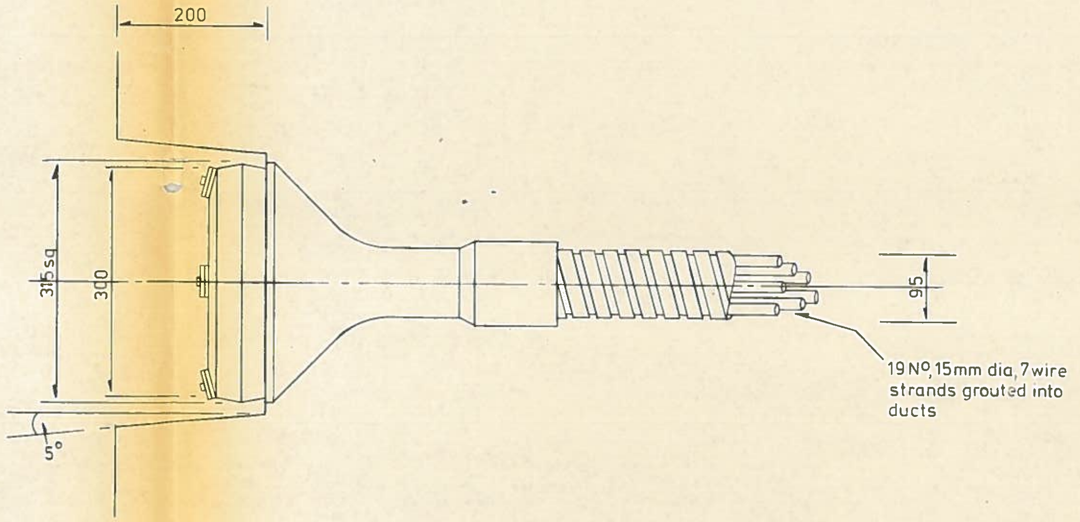
ISOMETRIC VIEW OF SPINE END



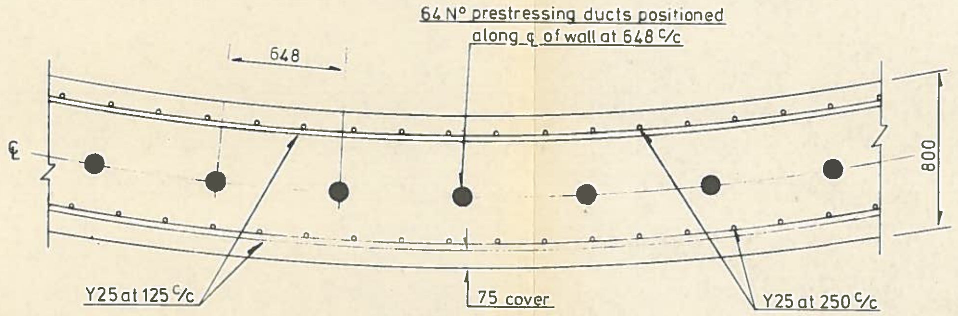
ELEVATION Y-Y



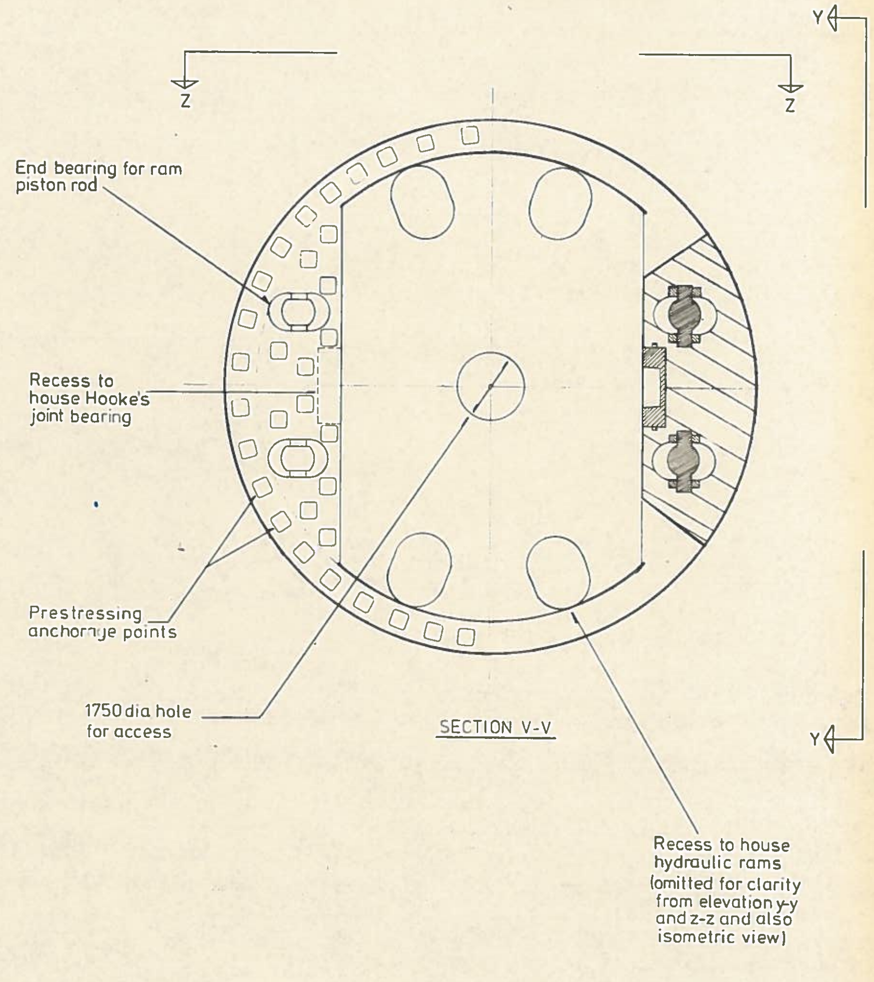
ELEVATION Z-Z



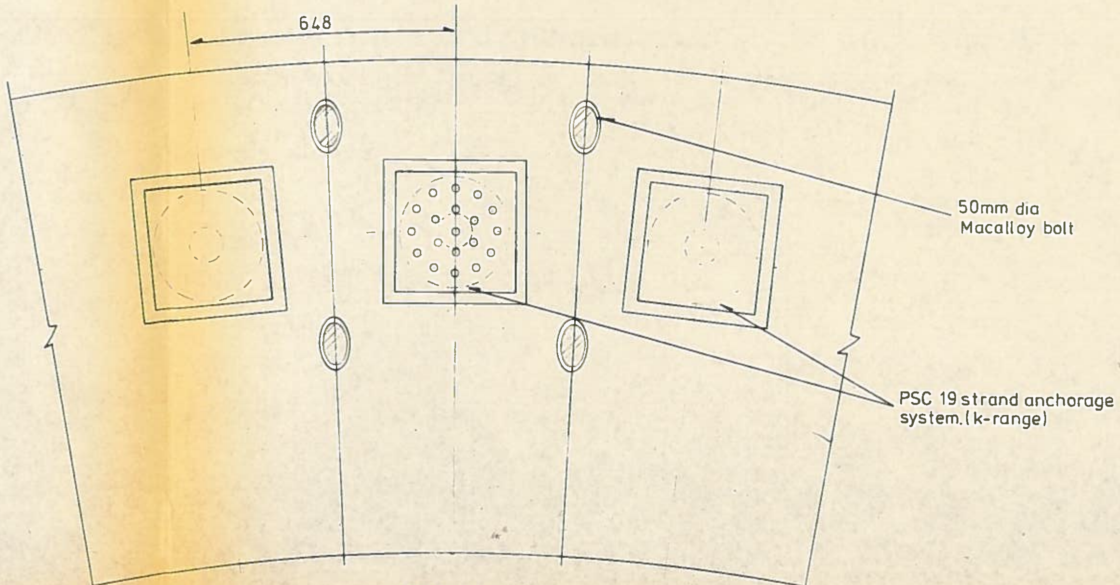
DETAIL 'A'



TYPICAL DETAIL OF SPINE WALL



SECTION V-V



ELEVATION X-X

TYPE	LOCATION	GRADE N/mm²	AGGREGATE	FINES	MIN CEMENT CONTENT Kg/m³	WATER/CEMENT RATIO
A	SPINE	50	CRUSHED ROCK/GRAVEL	NATURAL SANDS	400	0·4
B	DUCK	45	LYTAG	NATURAL SANDS	400	0·4

CONCRETE SPECIFICATION

Rev	Date	Amendment	Drawn	Chkd

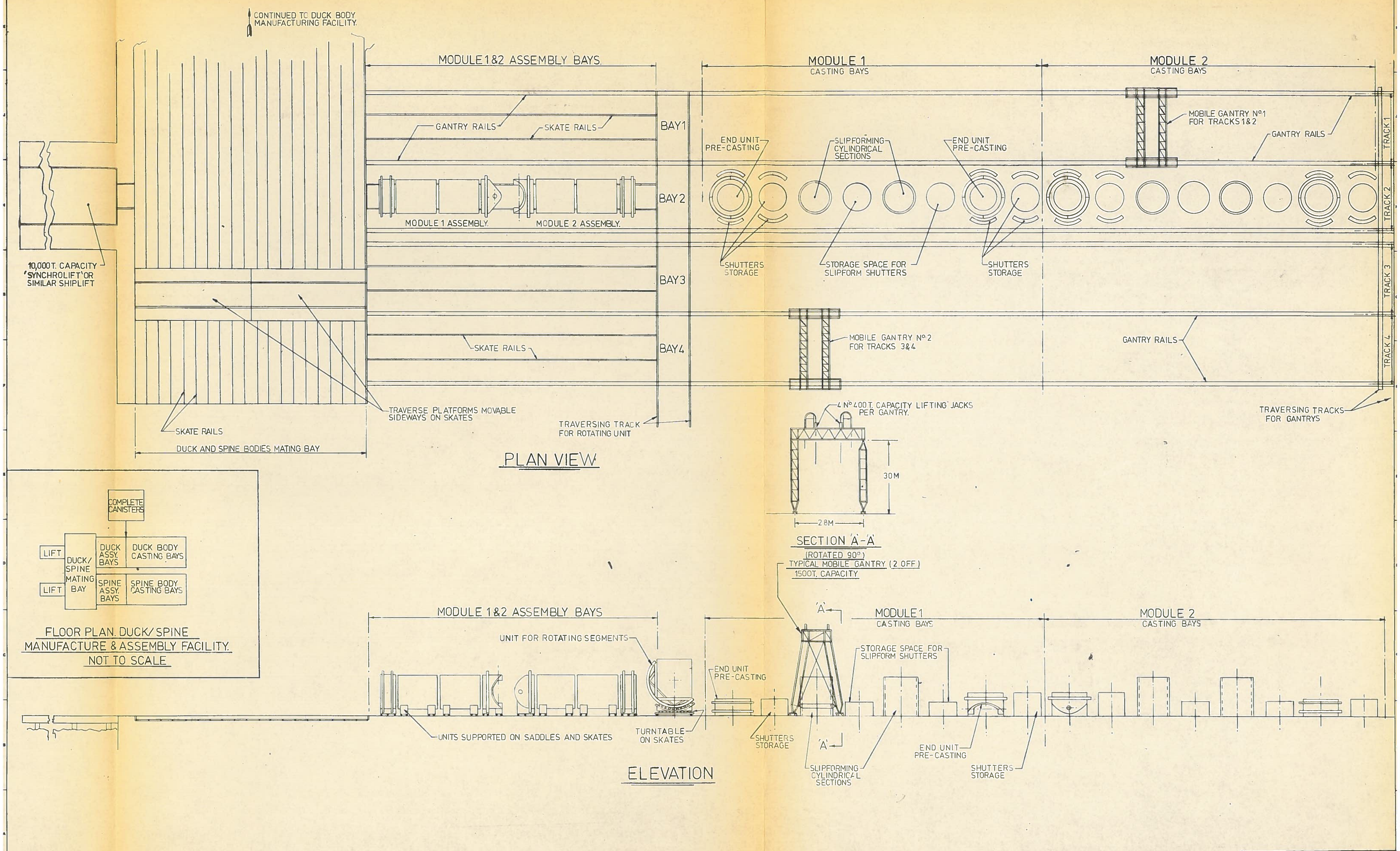
Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title
DETAILS OF SPINE

Project N° PAD 100 | L 9187-10102 | Rev O

Scale 1:50 1:5 1:10 1:20 1:100 | Drawn FPL | Date OCT 81 | Chkd

COMMERCIAL IN CONFIDENCE



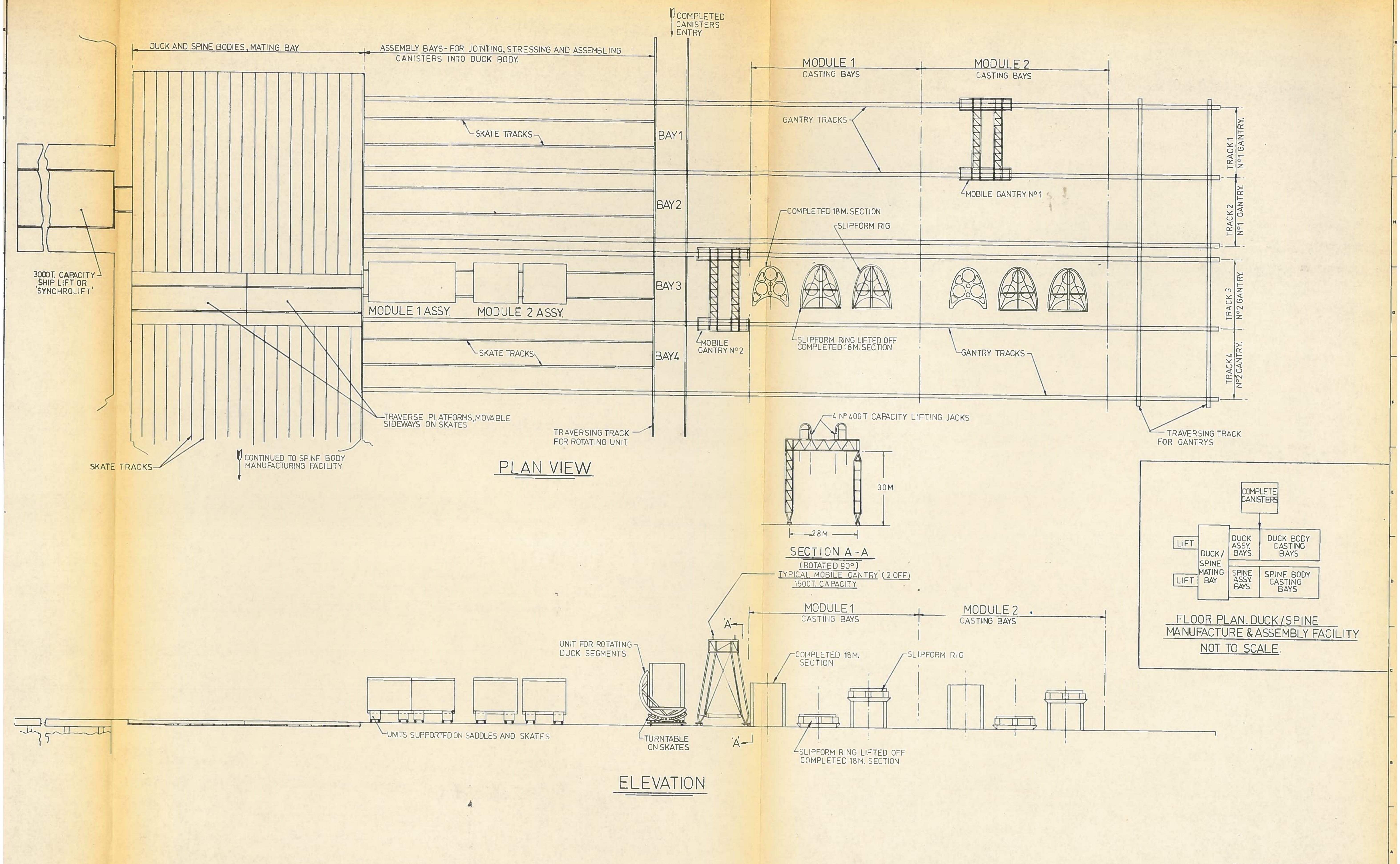
COMMERCIAL IN CONFIDENCE

Rev	Date	Amendment	Drawn	Chkd

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title
MANUFACTURING FACILITY FOR SPINE BODIES

Project Nº PAD 100
L 9187-10202 Rev 0
Scale 1:500
Drawn NEMOATE Date Nov '81 Chkd Date

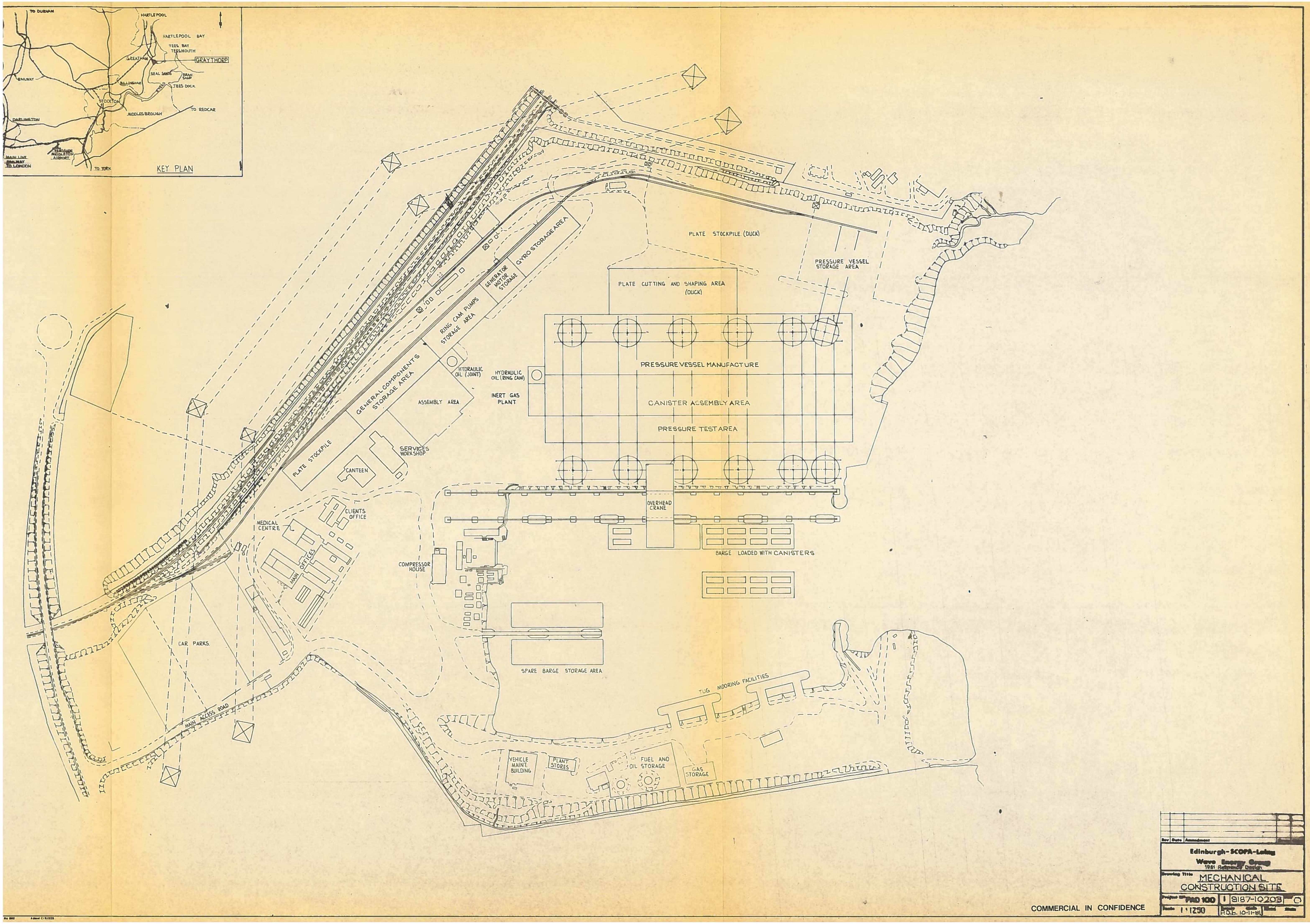


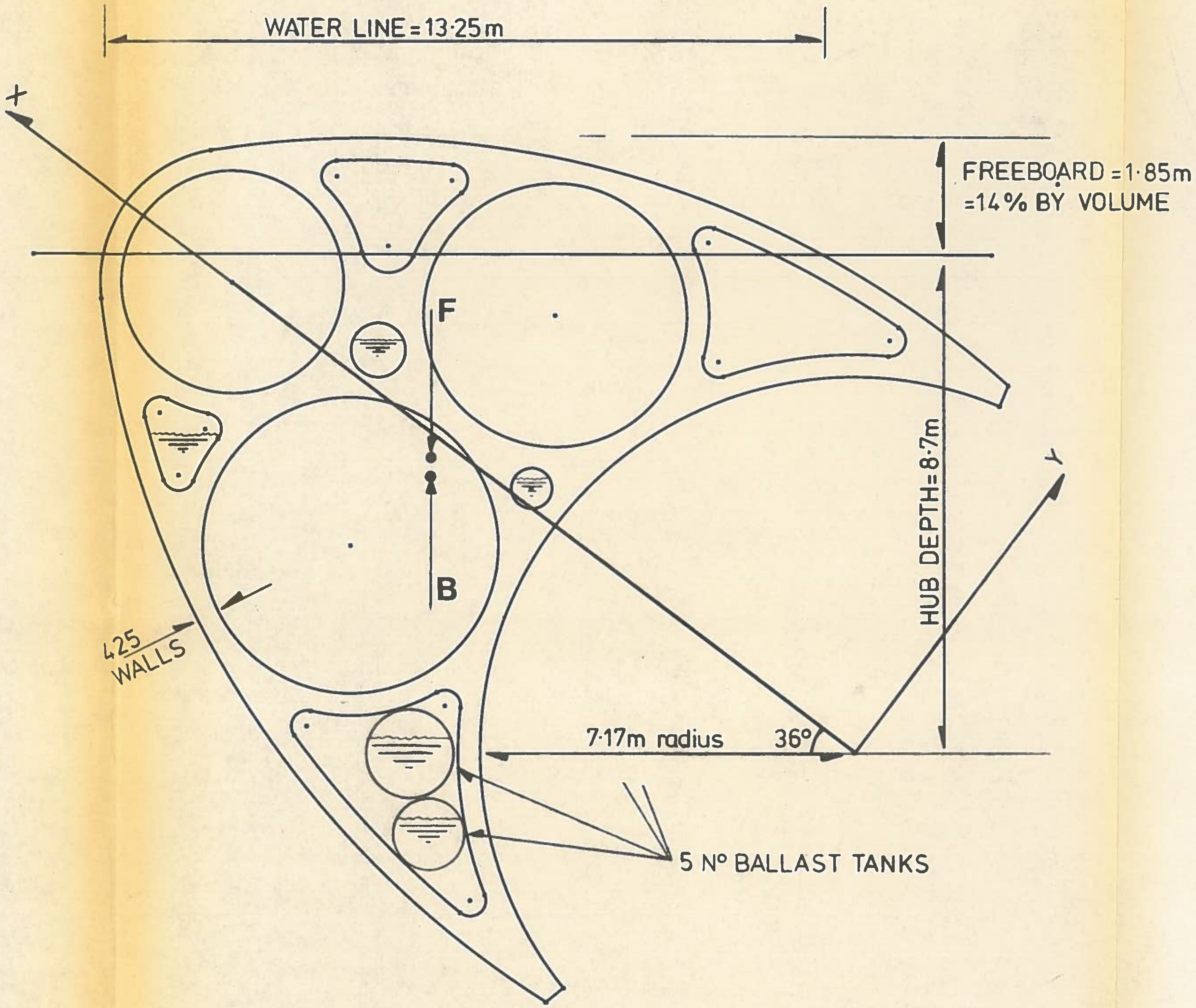
COMMERCIAL IN CONFIDENCE

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

MANUFACTURING FACILITY FOR DUCK BODIES

PAD 100 | L 9187-10201 | 0
1:500 | NEMOATE


TO DURHAM
HARTLEPOOL
HARTLEPOOL BAY
TEES BAY
TEESMOUTH
GREATHAM
GRAYTHORP
SEAL SANDS
BRAN SAND
BILLINGHAM
TEES DOCK
TEES
RAILWAY
STOCKTON
MIDDLESBROUGH
TO REDCAR
DARLINGTON
MAIN LINE RAILWAY TO LONDON
TEESSIDE MIDDLETON AIRPORT
TO YORK
KEY PLAN
PLATE STOCKPILE (DUCK)
PRESSURE VESSEL STORAGE AREA
PLATE CUTTING AND SHAPING AREA (DUCK)
GYRO STORAGE AREA
GENERATOR MOTOR STORAGE
RING CAM PUMPS STORAGE AREA
HYDRAULIC OIL (JOINT)
HYDRAULIC OIL (RING CAM)
INERT GAS PLANT
PRESSURE VESSEL MANUFACTURE
CANISTER ASSEMBLY AREA
PRESSURE TEST AREA
GENERAL COMPONENTS STORAGE AREA
ASSEMBLY AREA
PLATE STOCKPILE
SERVICES WORKSHOP
CANTEEN
CLIENTS OFFICE
MEDICAL CENTRE
MAIN OFFICES
COMPRESSOR HOUSE
OVERHEAD CRANE
BARGE LOADED WITH CANISTERS
CAR PARKS
SPARE BARGE STORAGE AREA
TUG MOORING FACILITIES
MAIN ACCESS ROAD
VEHICLE MAINT. BUILDING
PLANT STORES
FUEL AND OIL STORAGE
GAS STORAGE
COMMERCIAL IN CONFIDENCE
Rev Date Amendment
Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design
Drawing Title MECHANICAL CONSTRUCTION SITE
PAD 100 I 9187-10203 O
Scale 1:1250
P.D.B. 10-11-81



NOTE:- CONCRETE TO BE LYTAG-SAND-CEMENT, GRADE 45, MAX SATURATED DENSITY = 1·8 TONNE/m^2

HENCE, SATURATED DENSITY OF REINFORCED CONCRETE = 1·9 TONNE/m^2

FIXED WEIGHT 'F'

Concrete Shell	2750 Tonne	@ (9·627, 0·077)
Concrete End Plates	137 Tonne	@ (10·131, 0·691)
Power Pods + End Caps	500 Tonne	@ (10·077, -2·653)
Fresh Water Between Pod & Shell	56 Tonne	@ (10·077, -2·653)
TOTAL F	3443 Tonne	@ (9·720, -0·339)

ie, (R = 9·726, θ = -2°)

STILL WATER BUOYANCY 'B'

Buoyancy B = 3733 Tonne @ (9·622, -0·587)

ie, (R = 9·640, θ = -3·5°)

BALLAST

Ballast = B - F = 290 Tonne

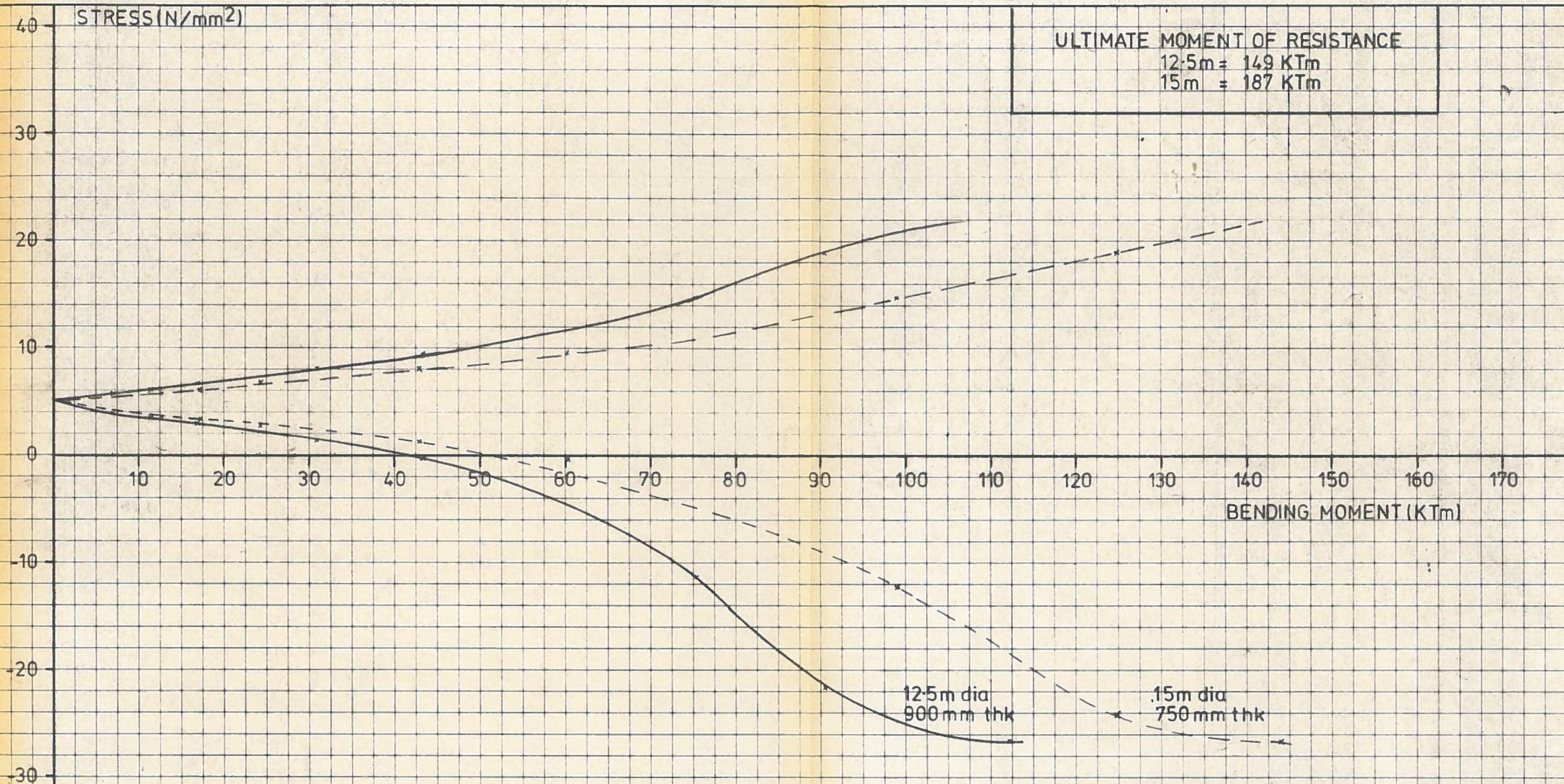
COMMERCIAL IN CONFIDENCE

Rev	Date	Amendment	Drwn	Chkd

Edinburgh - SCOPA - Laing
Wave Energy Group
1981 Reference Design

Drawing Title: DUCK BUOYANCY CALCULATIONS

Project N° PAD 100	L	9187 E 10111	Rev
Scale 1:100	Drwn F.P.L. Date NOV81	Chkd RAR	Date Nov 81



COMMERCIAL IN CONFIDENCE

BENDING MOMENT KTm	12·5 m DIA' MAX' COMP' STRESS N/mm²	12·5 m DIA' MIN' COMP' STRESS N/mm²	15m DIA' MAX' COMP' STRESS N/mm²	15m DIA' MIN' COMP' STRESS N/mm²
5	5·6	4·0	5·2	4·4
10	6·0	3·5	5·6	3·9
15	6·4	3·2	5·9	3·6
20	6·9	2·6	6·2	3·2
25	7·4	2·0	6·6	2·8
30	7·9	1·6	6·9	2·4
33	8·1	1·2	7·2	2·2

CHARACTERISTIC CUBE STRENGTH = 50 N/mm²

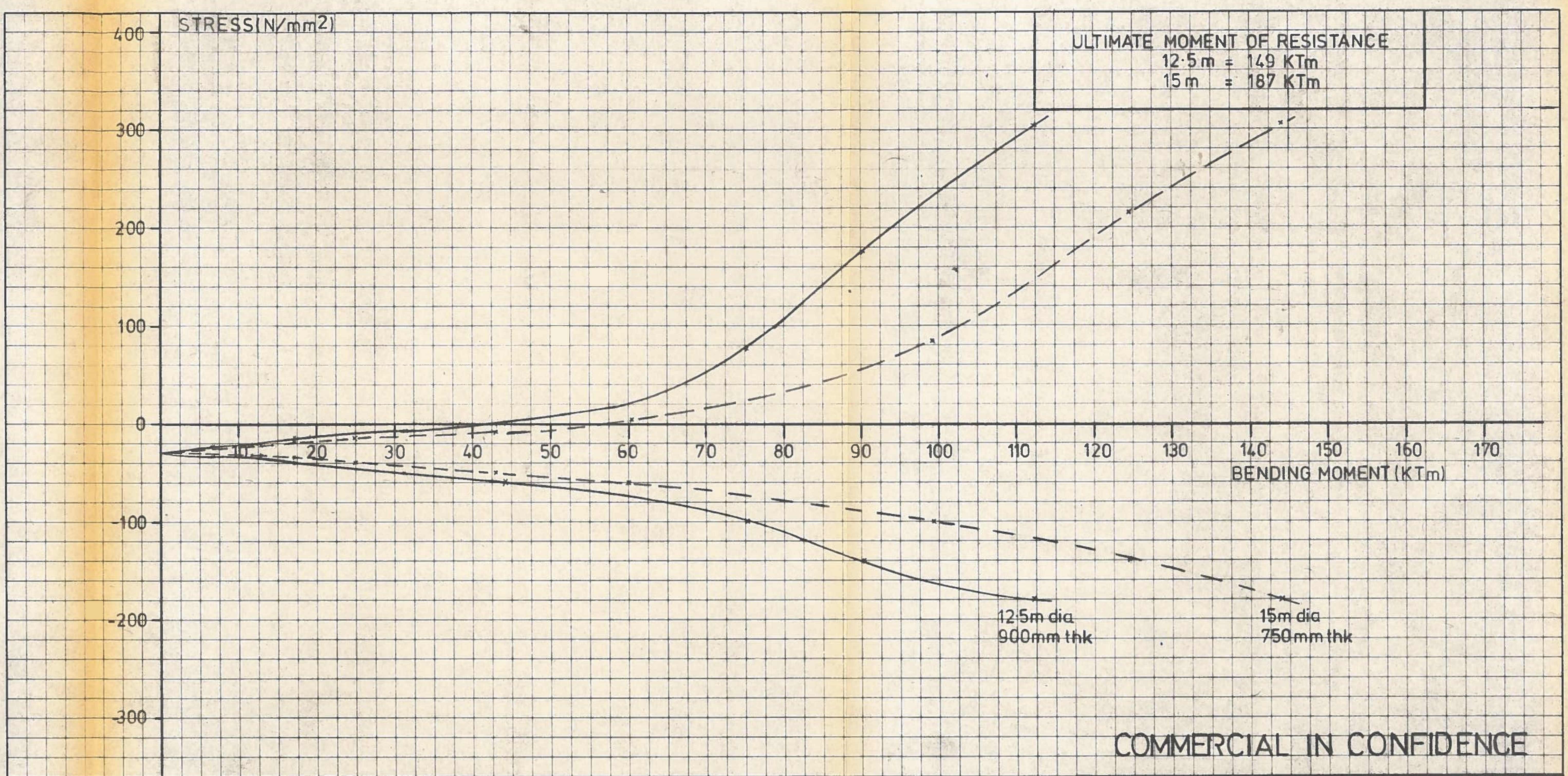
Rev	Date	Ammendment	Drwn	Chkd

Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SPINE DESIGN
CHARACTERISTIC BENDING MOMENT vs STRESS CURVES FOR CONCRETE

Project No.		Rev
PAD 100	L 9187 E 10103	

Scale	Drwn	Date	Chkd	Date
	J.P.L.	NOV'81	RW	Nov'81



BENDING MOMENT N/mm²	12·5m DIA' MAX' COMP' STRESS N/mm²	12·5m DIA' MIN' COMP' STRESS N/mm²	15m DIA' MAX' COMP' STRESS N/mm²	15m DIA' MIN' COMP' STRESS N/mm²
5	34	26	31	28
10	34	22	32	24
15	37	18	32	22
20	42	14	36	18
25	46	10	40	15
30	50	8	42	14
33	52	7	44	12

CHARACTERISTIC YIELD STRESS = 425 N/mm²

COMMERCIAL IN CONFIDENCE

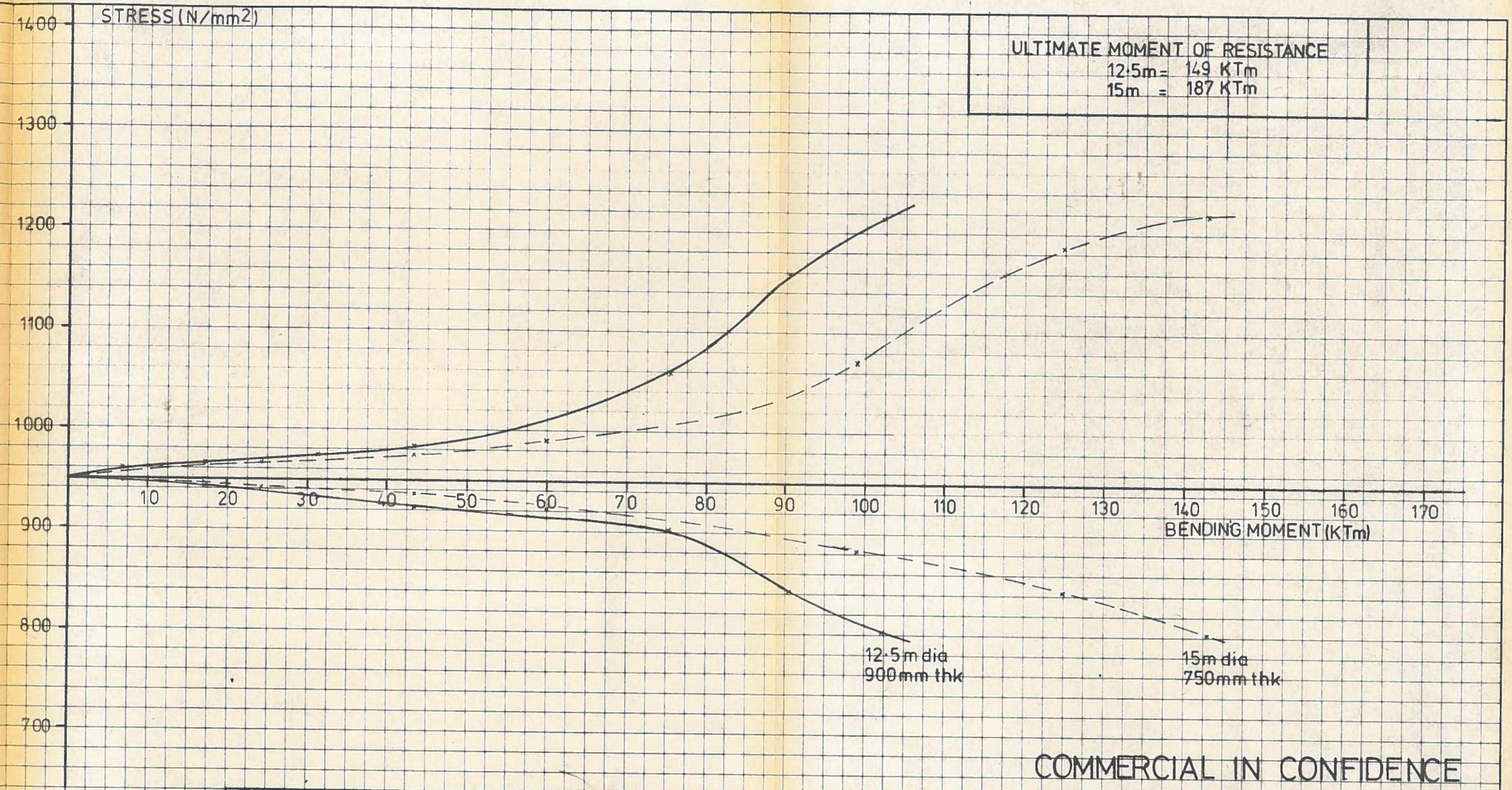
Rev	Date	Ammendment	Drwn	Chkd

Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SPINE DESIGN
CHARACTERISTIC BENDING MOMENT vs STRESS CURVES FOR REBAR

Project No. PAD 100 | L | 9187: E 10104 | Rev

Scale | Drwn J.P.L. | Date NOV'81 | Chkd [illegible] | Date Nov '81



COMMERCIAL IN CONFIDENCE

BENDING MOMENT KTm	12·5m DIA MAX' TENS' STRESS N/mm^2	12·5m DIA MIN' TENS' STRESS N/mm^2	15m DIA MAX' TENS' STRESS N/mm^2	15m DIA MIN' TENS' STRESS N/mm^2
5	957	947	954	947
10	962	946	957	946
15	965	943	961	946
20	968	941	964	944
25	970	936	965	941
30	974	934	968	940
33	975	932	969	940

GUARANTEED BREAKING STRENGTH = 1700 N/mm^2

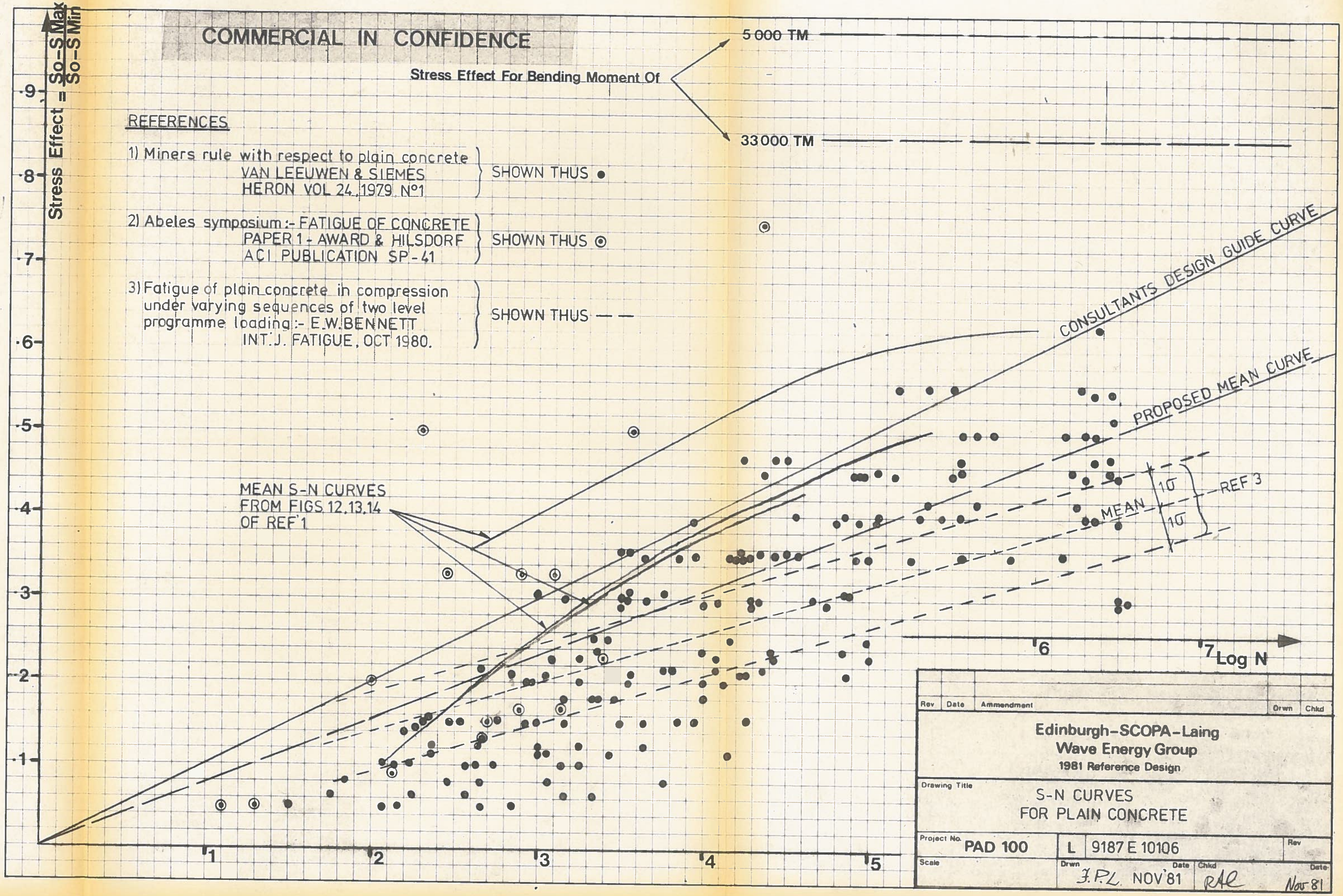
Rev	Date	Ammendment	Drwn	Chkd

Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SPINE DESIGN
CHARACTERISTIC BENDING MOMENT vs STRESS CURVES FOR TENDONS

Project No.			Rev
PAD 100	L	9187 E 10105	

Scale	Drwn	Date	Chkd	Date
	J.P.L.	NOV'81	RLe	Nov '81


COMMERCIAL IN CONFIDENCE
Stress Effect = So-S Max / So-S Min
Stress Effect For Bending Moment Of
5 000 TM
33 000 TM
REFERENCES
1) Miners rule with respect to plain concrete
VAN LEEUWEN & SIEMES
HERON VOL 24, 1979 N°1
SHOWN THUS ●
2) Abeles symposium :- FATIGUE OF CONCRETE
PAPER 1 - AWARD & HILSDORF
ACI PUBLICATION SP-41
SHOWN THUS ⊙
3) Fatigue of plain concrete in compression
under varying sequences of two level
programme loading :- E.W. BENNETT
INT. J. FATIGUE, OCT 1980.
SHOWN THUS — —
CONSULTANTS DESIGN GUIDE CURVE
PROPOSED MEAN CURVE
MEAN S-N CURVES
FROM FIGS 12, 13, 14
OF REF 1
MEAN
1σ
1σ
REF 3
·9
·8
·7
·6
·5
·4
·3
·2
·1
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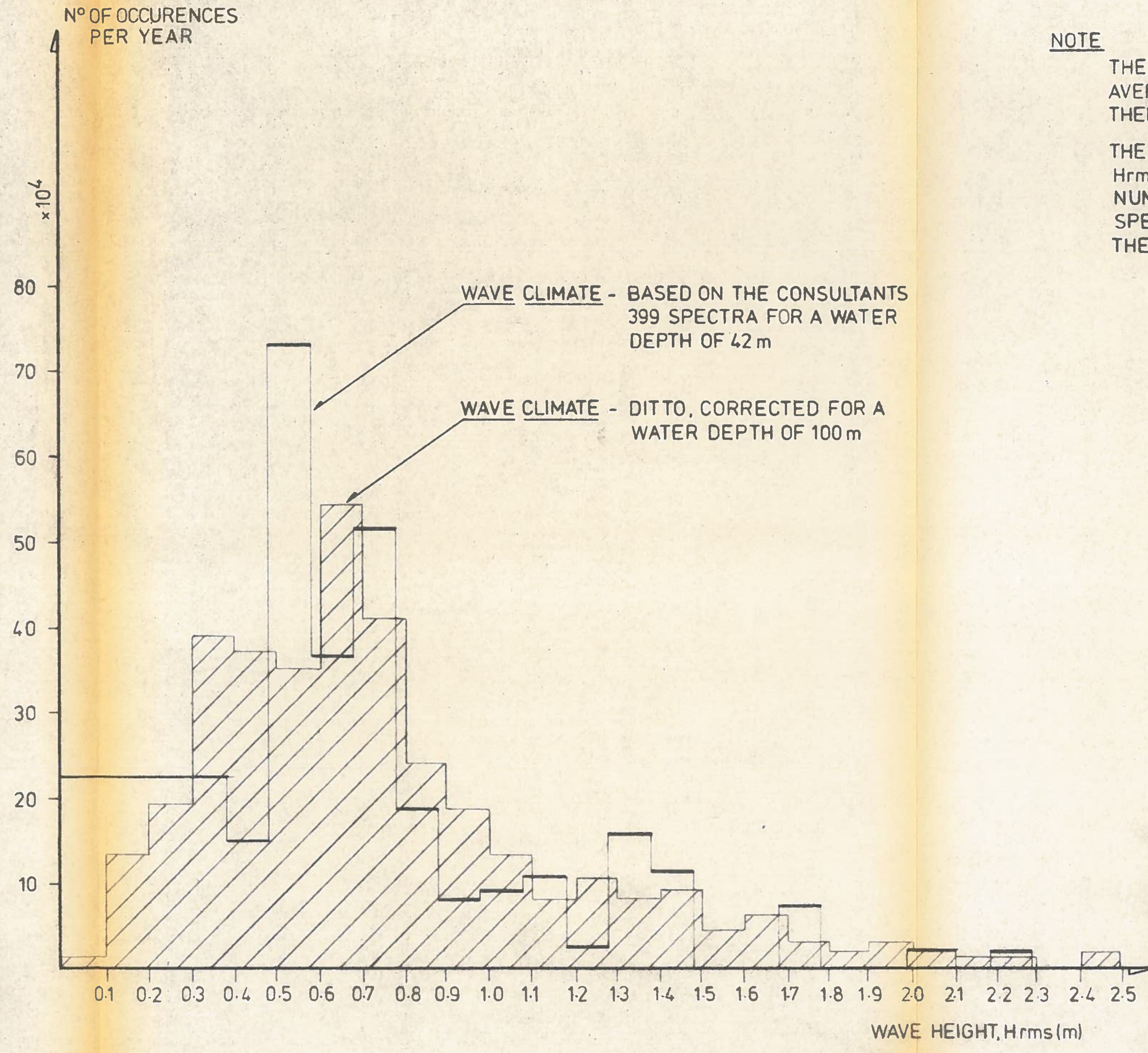
Log N
Rev
Date
Ammendment
Drwn
Chkd
Edinburgh-SCOPA-Laing
Wave Energy Group
1981 Reference Design
Drawing Title
S-N CURVES
FOR PLAIN CONCRETE
Project No.
PAD 100
L
9187 E 10106
Rev
Scale
Drwn
J.P.L.
Date
NOV '81
Chkd
RAL
Date
Nov 81



NOTE

THE 399 SPECIFIED WAVE SPECTRA HAVE AN AVERAGE PERIOD (Te) OF 8·89 SECONDS AND THEREFORE REPRESENT 1/8890 YEARS.

THE HISTOGRAM, WITH BAND WIDTH OF 0·1m Hrms, IS DRAWN BY MULTIPLYING THE NUMBER OF OCCURENCES (FROM 399 SPECTRA SPECIFIED) BY 8890 TO DETERMINE THE WAVE CLIMATE FOR 1 YEAR

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Rev	Date	Ammendment	Drwn	Chkd

Edinburgh–SCOPA–Laing
Wave Energy Group
1981 Reference Design

Drawing Title: WAVE CLIMATE FOR A ONE YEAR PERIOD

Project No	L	Rev
PAD 100	9187 E 10107	

Scale	Drwn	Date	Chkd	Date
	F.P.L.	NOV 81	rbl	Nov 81

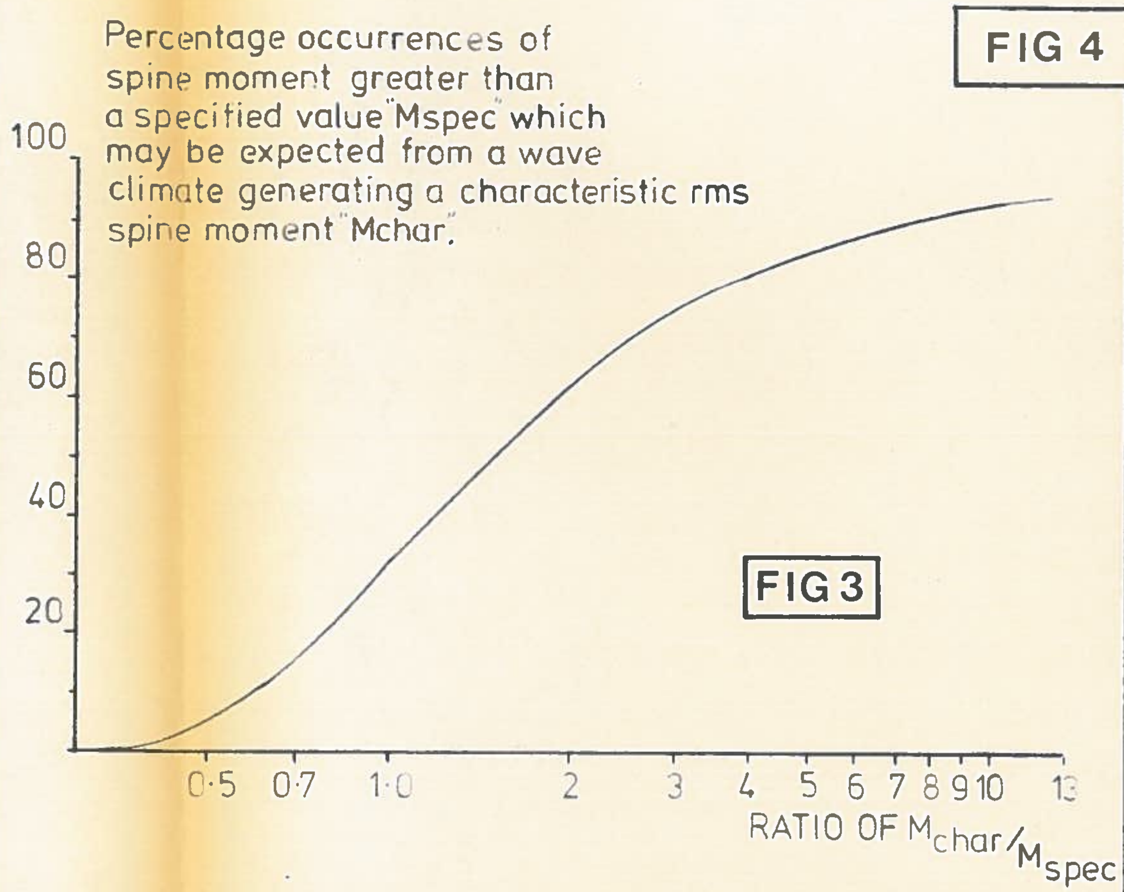

FIG 3

FIG 4

SPINE DIAMETER	BENDING MOMENT KTm	SURGE $n/1000$	SURGE $N/10^6$		COMBINED $n/1000$	COMBINED $N/10^6$	
12·5	0 – 5	408	2973	MINER'S SUM= $777/10^5$	749	2973	MINER'S SUM= $896/10^5$
	5 – 10	386	1652		630	1652	
	10 – 15	362	1069		490	1069	
	15 – 20	326	571		380	571	
	20 – 25	2070	318		296	318	
	25 – 30	0	-		228	199	
	30 – 33	0	-		774	151	
15	0 – 5	267	5284	MINER'S SUM= $390/10^5$	452	5284	MINER'S SUM= $529/10^5$
	5 – 10	188	2786		408	2786	
	10 – 15	219	1876		367	1876	
	15 – 20	215	1219		319	1219	
	20 – 25	2659	762		276	762	
	25 – 30	0	-		237	519	
	30 – 33	0	-		1490	393	

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THE BENDING MOMENT IN THE SPINE IS LIMITED TO 23000 Tm IN SURGE OR HEAVE AND TO 33000 Tm IN "COMBINED SURGE AND HEAVE" BY THE ACTION OF THE RAMS IN THE SPINE JOINT. (20000 Tm AT THE JOINT PLUS A POSSIBLE 3000 Tm INCREASE BETWEEN THE JOINTS, IN EITHER SURGE OR HEAVE DIRECTION)

FIG 2:-
THE THEORETICAL NUMBER OF OCCURRENCES OF A GIVEN CHARACTERISTIC MOMENT WAS DETERMINED BY COMBINING THE CHARACTERISTIC MOMENT / Hrms CURVE (FIG 1) & THE WAVE CLIMATE HISTOGRAM, (Drwg. N° 10107)

FIGS 3,4:-
FROM THIS CURVE A TABLE, (FIG 4), OF ACTUAL SPINE MOMENT OCCURRENCE PER YEAR, n, WAS DRAWN, ASSUMING A NORMAL DISTRUBUTION OF MOMENT EXCEEDENCE, (FIG 3). THE NUMBER OF OCCURRENCES TO FAILURE, N, IS GIVEN FOR EACH MOMENT RANGE, (SEE S-N CURVE Drwg. N° 10106) HENCE THE MINER'S SUM FOR 1 YEAR IS CALCULATED.

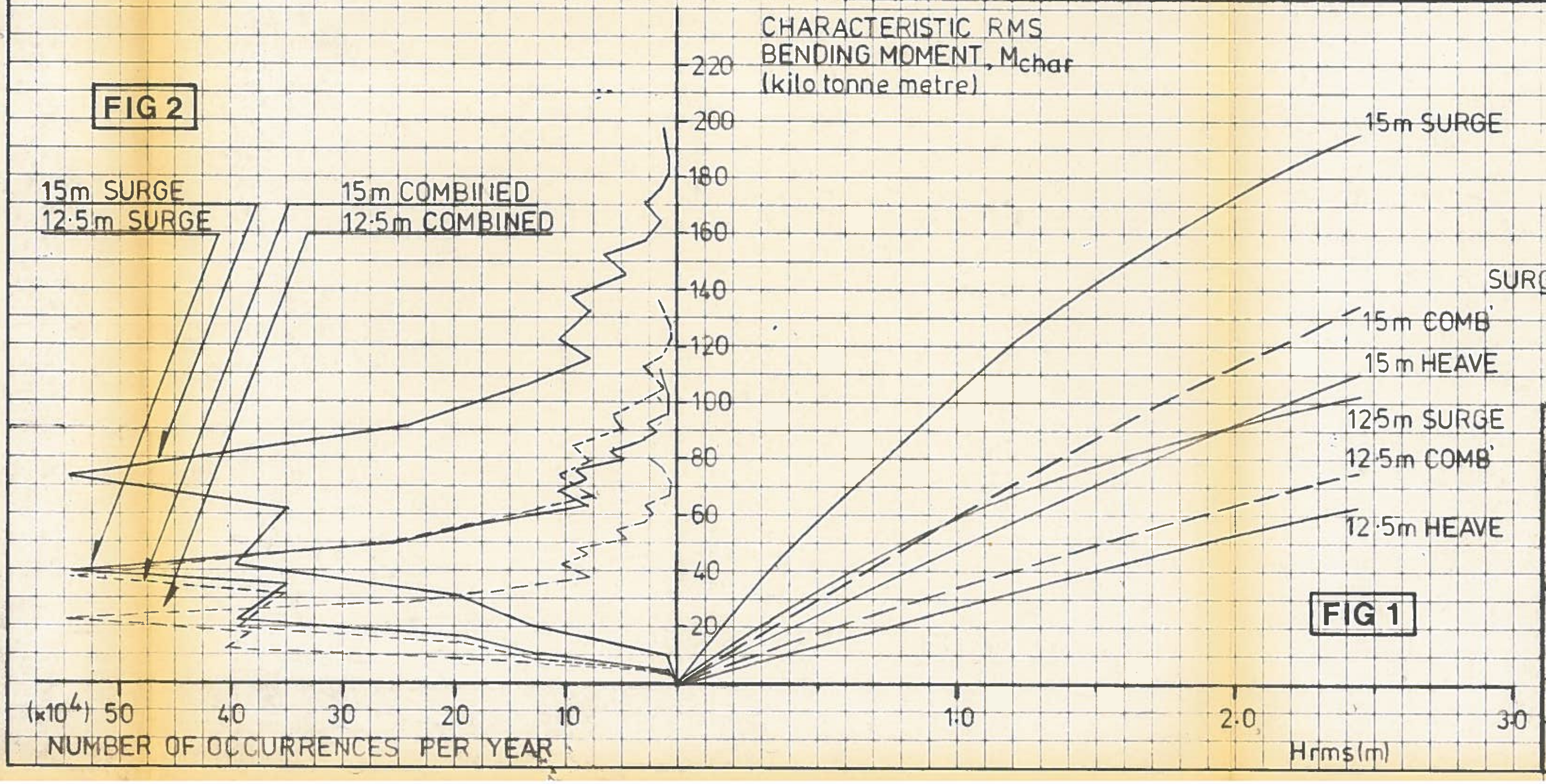
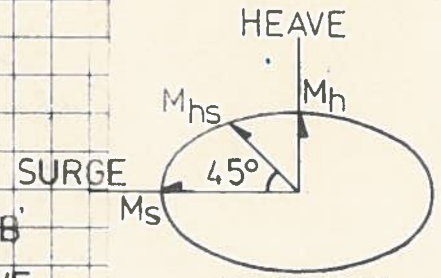


FIG 2

FIG 1



COMBINED SURGE & HEAVE MOMENT, $M_{hs} = \frac{M_s \times M_h}{\sqrt{M_s^2 + M_h^2}} \times \sqrt{2}$

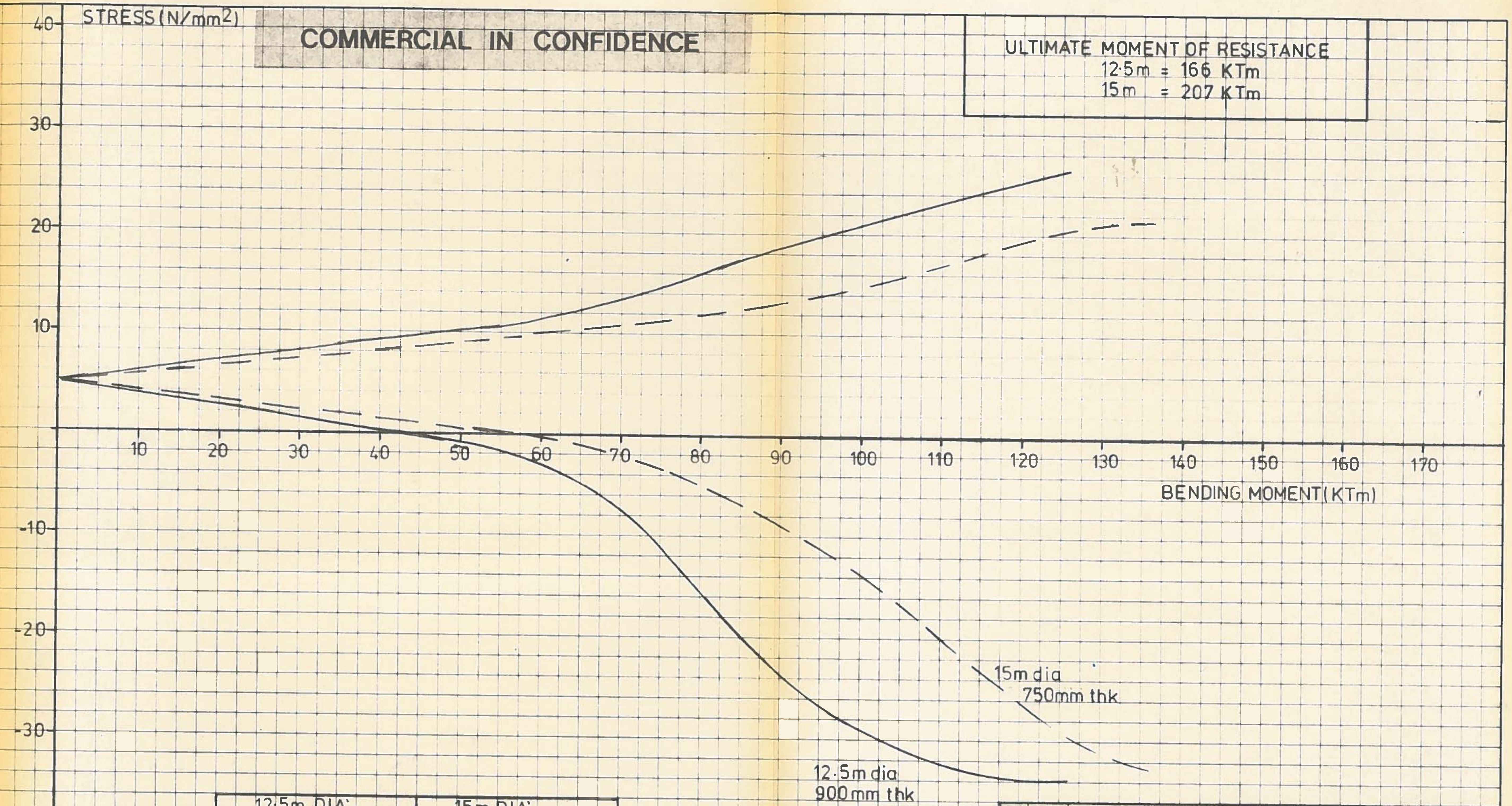
FIG 1a

Rev	Date	Ammendment	Drwn	Chkd

Edinburgh-SCOPA-Laing Wave Energy Group
1981 Reference Design

Drawing Title: CHARACTERISTIC BENDING MOMENT HISTOGRAM BUILD UP

Project No.			Rev
PAD 100	L	9187 E 10108	
Scale	Drwn J.P.L. Date NOV'81	Chkd RbL	Date Nov'81



ULTIMATE MOMENT OF RESISTANCE
12·5m = 166 KTm
15m = 207 KTm

BENDING MOMENT KTm	12·5m DIA' MAX' COMP' STRESS N/mm^2	12·5m DIA' MIN' COMP' STRESS N/mm^2	15m DIA' MAX' COMP' STRESS N/mm^2	15m DIA' MIN' COMP' STRESS N/mm^2
5	5·5	4·4	5·4	4·5
10	6·1	3·8	5·8	4·1
15	6·7	3·2	6·2	3·7
20	7·2	2·8	6·6	3·2
25	7·7	2·1	7·0	2·8
30	8·3	1·5	7·4	2·3
33	8·7	1·2	7·6	2·1

MEAN CUBE STRENGTH
$= 63\ N/mm^2$

Rev	Date	Ammendment	Drwn	Chkd

Edinburgh–SCOPA–Laing
Wave Energy Group
1981 Reference Design

Drawing Title: SPINE DESIGN
MEAN BENDING MOMENT vs STRESS CURVES
FOR CONCRETE

Project No. PAD 100 | L | 9187 E 10109 | Rev

Scale | Drwn J. P. L. | Date NOV81 | Chkd [signature] | Date Nov '81

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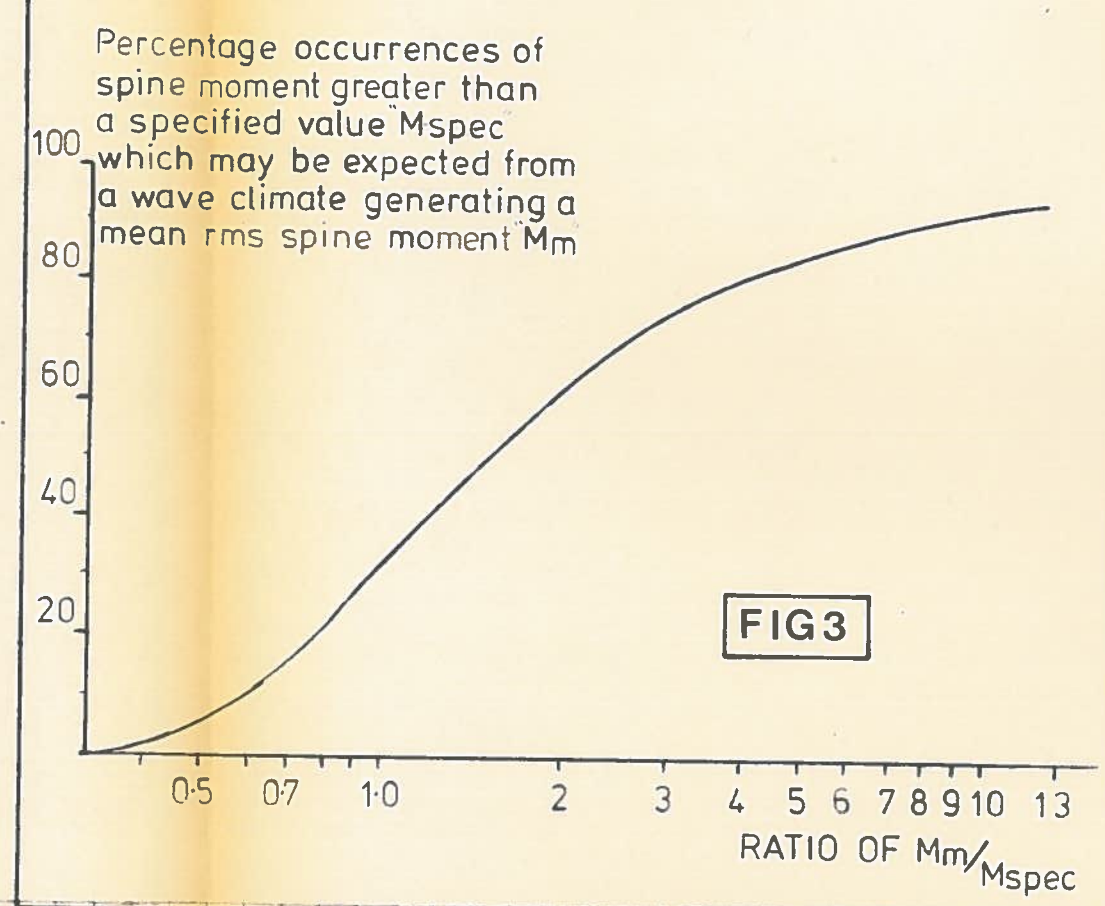


FIG 3

FIG 4

SPINE DIAMETER	BENDING MOMENT KTm	SURGE $n/1000$	SURGE $N/10^9$		COMBINED $n/1000$	COMBINED $N/10^9$	
12·5	0 – 5	538	3517	MINER'S SUM = $986/10^8$	997	3517	MINER'S SUM = $1064/10^8$
	5 – 10	489	1534		727	1534	
	10 – 15	418	722		514	722	
	15 – 20	351	423		366	423	
	20 – 25	1750	220		259	220	
	25 – 30	0	-		181	117	
	30 – 33	0	-		502	81	
15	0 – 5	340	4078	MINER'S SUM = $572/10^8$	623	4078	MINER'S SUM = $727/10^8$
	5 – 10	279	2296		525	2296	
	10 – 15	285	1347		429	1347	
	15 – 20	260	771		349	771	
	20 – 25	2385	480		286	480	
	25 – 30	0	-		234	291	
	30 – 33	0	-		1102	234	

THE BENDING MOMENT IN THE SPINE IS LIMITED TO 23000 Tm IN SURGE OR HEAVE AND TO 33000 Tm IN COMBINED SURGE AND HEAVE BY THE ACTION OF THE RAMS IN THE SPINE JOINT. (20 000 Tm AT THE JOINT PLUS A POSSIBLE 3000 Tm INCREASE BETWEEN THE JOINTS IN EITHER SURGE OR HEAVE DIRECTION)

FIG 2 :-

THE THEORETICAL NUMBER OF OCCURRENCES OF A GIVEN MEAN MOMENT WAS DETERMINED BY COMBINING THE MEAN MOMENT/Hrms CURVE (FIG 1) WITH THE WAVE CLIMATE HISTOGRAM (Drwg N° 10107)

FIGS 3, 4 :-

FROM THIS CURVE A TABLE (FIG 4) OF ACTUAL SPINE MOMENT OCCURRENCE PER YEAR WAS DRAWN, ASSUMING A NORMAL DISTRIBUTION OF MOMENT EXCEEDENCE (FIG 3).
THE NUMBER OF OCCURRENCES TO FAILURE, N, IS GIVEN FOR EACH MOMENT RANGE (SEE S-N CURVE, Drwg N°
HENCE THE MINER'S SUM FOR 1 YEAR IS CALCULATED.

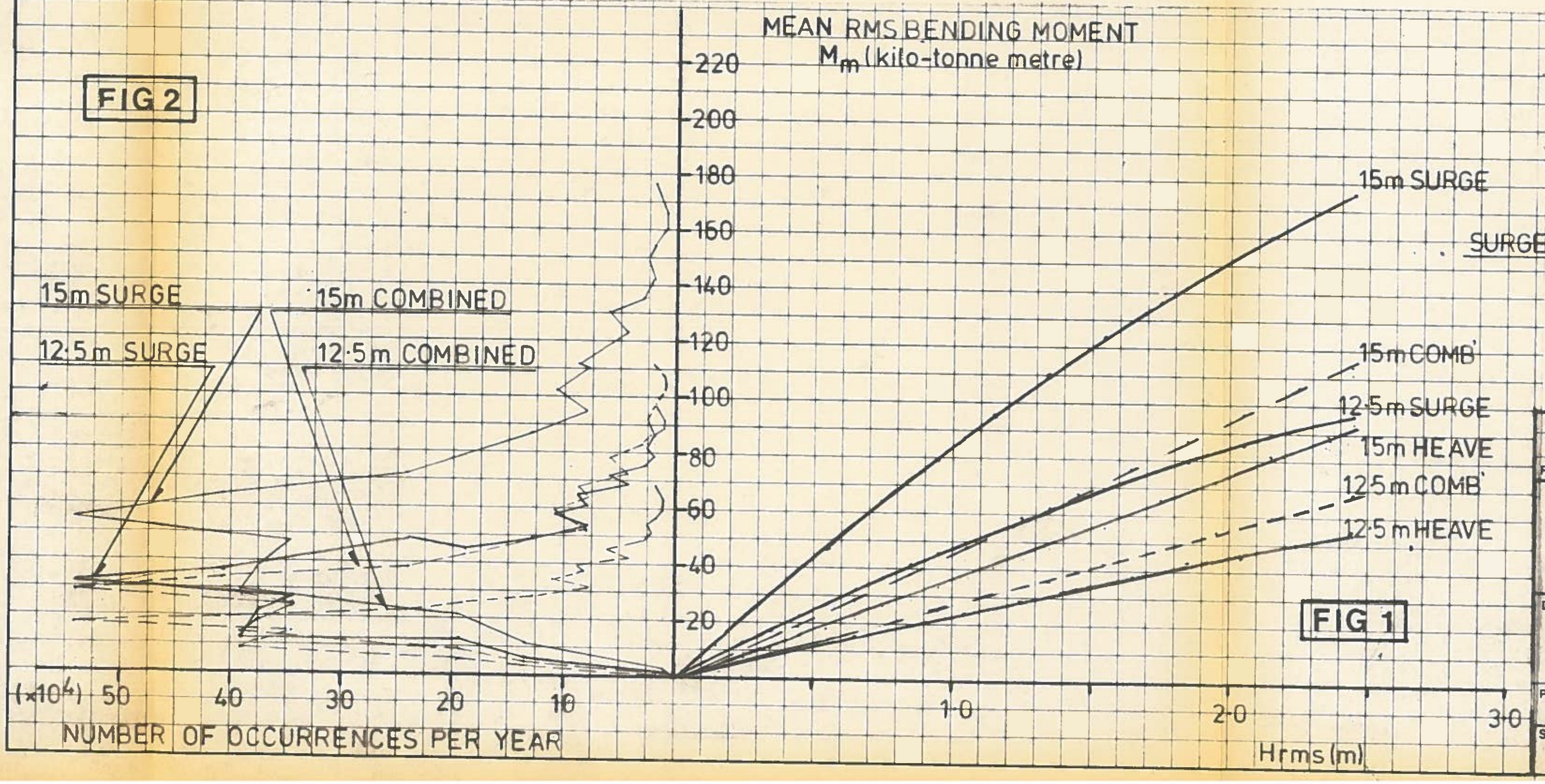
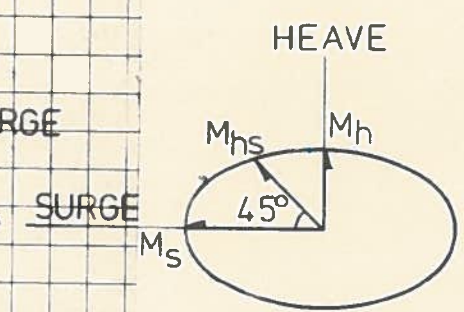


FIG 2

FIG 1



COMBINED SURGE & HEAVE MOMENT, $M_{hs} = \frac{M_s \times M_h}{\sqrt{M_s^2 + M_h^2}} \times \sqrt{2}$

FIG 1a

Rev	Date	Ammendment	Drwn	Chkd

Edinburgh–SCOPA–Laing Wave Energy Group
1981 Reference Design

Drawing Title: MEAN BENDING MOMENT HISTOGRAM BUILD UP

Project No.	PAD 100	L	9187 E 10110	Rev
Scale	Drwn J.P.L.	Date NOV'81	Chkd	Date Nov '81