WAVE ENERGY STEERING COMMITTEE
UNITED KINGDOM WAVE ENERGY PROGRAMME
CONSULTANTS SECOND REPORT
VOLUME 2
TECHNICAL APPRAISAL OF THE DEVICES
AUGUST 1978
PART I

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DEPARTMENT OF ENERGY

WAVE ENERGY STEERING COMMITTEE

UNITED KINGDOM WAVE ENERGY PROGRAMME

CONSULTANTS SECOND REPORT

VOLUME 2

Technical Appraisal of the Devices

AUGUST 1978

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[illegible]

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August 25th 1978

DRAFT VERSION OF VOLUME II

This document has been produced in draft to give guidance to WESC until the issue of the final version of the report. Production of the final version has been delayed to allow the incorporation of important information made available too late to meet the August 25th reporting date.

RENDEL, PALMER & TRITTON

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NEL Oscillating Water Column

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2	2 GW Scheme
3	Mooring
4	General Arrangement
5	Details
6	Construction and Flotation

Cockerell Raft

WP78/RAFT/1	Isometric
2	2 GW Scheme
3	Mooring
4	General Arrangement
5	Details of Power Take-off
6	Construction and Flotation

HRS Russell Rectifier

WP78/RECT/1	Isometric
2	2 GW Scheme
3	General Arrangement
4	Construction and Flotation

Salter Duck

WP78/SALT/1	Isometric
2	2 GW Scheme
3	Moorings
4	General Arrangement
5	Proposal for Full-Scale - Edinburgh University
6	Construction and Flotation

French Flexible Bag

WP78/FREN/1	Isometric
2	Mooring
3	General Arrangement
4	Details
5	Construction and Flotation

Wells Oscillator

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2	General Arrangement and Construction

Vickers Device

WP78/VICK/1	Isometric
2	Provisional Visualisation

CHAPTER 1.0 - INTRODUCTION AND GENERAL INFORMATION

1.1 INTRODUCTION

This report is the second general assessment prepared by the Consultants for the Wave Energy Steering Committee, the first having been submitted in August 1977. The primary objective of this report is to present to those responsible for directing the U.K. wave energy programme a full assessment of the devices now under development from the point of view of their potential for large scale implementation. The Consultants have attempted to assemble as firm a basis of factual information as is possible at this stage, to guide future decision making.

The report is presented in three volumes, Volume 1 is an Executive Summary and includes the conclusions for the whole report. Volume 2 is the main body of the report and deals with the technical assessment, Volume 3 contains the costing information.

The report assesses devices and not Device Teams. The Consultants have tried to present as fair a picture as possible of the devices as conceived by the Teams, and the Teams are of course the principal source of information. However, the text also refers to work from other sources. Every effort has been made to identify the inherent strengths and weaknesses of the devices independently of the work of the development Teams.

It is recognised that at some stage certain devices will be dropped from the programme to allow concentration of effort on the more promising schemes. The Consultants have therefore given special prominence to those topics which are likely to have most influence on such decisions.

An important limitation imposed on this report is its timing. It finds many Device Teams halfway through planned programmes of work, and in many areas detailed information necessary for a complete assessment is missing. In these areas attention is drawn to those factors which may later modify the stated conclusions on particular aspects of devices. However, the Consultants feel that it is now possible to reach reliable conclusions on many of the broader aspects of device development.

Seven devices are included in the assessment. Some are much further advanced than others, and some are much more complex. The depths of the assessments carried out reflect these factors.

Chapters 4 to 10 of this Volume present for each device a Reference Design which has been used as the basis of assessment. Table 1.1 sets out the key parameters of the Reference Designs. These designs have either been produced in their entirety by the Device Teams, or have been in part worked up by the Consultants in consultation with the Device Teams. The devices are described and assessed technically in terms of their material and workmanship content, and in terms of their annual average power output (termed 'productivity' in this report).

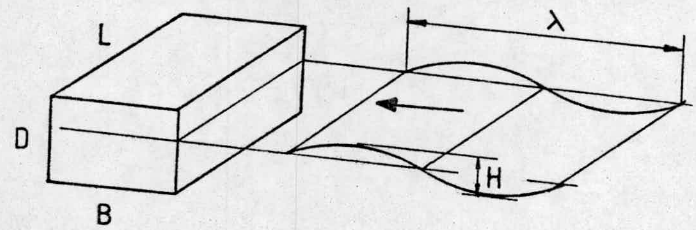
For each device a brief summary of the fundamental mechanism of wave power extraction is given as general information, and for comparison between devices. There is a strong link between these fundamentals and the most important engineering problems, including costs.

Except where it is directly applicable to the assessment, this report does not deal in any detail with the extensive programme of support work which has been initiated in areas of interest to all devices. This work is undertaken by Technical Advisory Groups (TAGS) and their work is documented in numerous separate reports.

	MODE OF INTERACTION WITH WAVES				REACTIVE FRAME				POWER TAKE-OFF	
		MOVING SURFACE MODULATORS		PLAN	FLOATING					
DEVICE	RECTIFIERS	RESONANT	NON-RESONANT	GEOMETRY	INERTIA	SPINE	OTHERS	SEABED	MECHANICAL	FLUID
NEL OSCILLATING WATER COLUMN		Vertical hydraulic interface (OWC)		Line, parallel to crests	Inertia in heave surge and roll					Rectified air turbine
COCKERELL RAFTS		Rigid horizontal surface		'Squat' longer dimension normal to crest	Inertia power from relative motion of articulated structure				Gears? Rams?	
RUSSELL RECTIFIER	Vertical face flap rectifier			Line parallel to crests				Bottom sitting Breaking surface		Rectified water turbine
SALTER DUCKS		Rigid inclined interface (partially immersed)		Line parallel to crests		Spine			Gears? Pumps?	
FRENCH FLEXIBLE BAG			Light flexible vertical interface	Line normal to crests	Inertia, with spine action					Rectified air turbine
WELLS OSCILLATOR		Horizontal hydraulic interface (OWC)		Point absorber	Inertia in heave (and roll)					Non-rectified air turbine
VICKERS OSCILLATOR		Immersed horizontal hydraulic interface (OWC)		Point absorber				Bottom sitting Submerged		Rectified water turbine
OTHER POSSIBILITIES NOT COVERED	Horizontal face rectifier	Immersed rigid interface	Other cross sections, Light rigid interfaces	Ring buoys, Focussing arrays			Propeller devices	Rigid tethered, Active tethered - power taken from moorings	Many others - Friction Direct electrical generation, etc.	Non-rectified water turbines

CLASSIFICATION OF DEVICES

VOLUME 1 - TABLE S.3
VOLUME 2 - TABLE 1.1



λ = wave length
H = wave height
Lc = crest length

	L	B, b	D, d	Mass and Inertia
NEL	Not critical Mooring and power take-off favour long devices Structure and end cell behaviour favour short devices	B governed by inertia and duct design $b \doteqdot \frac{\lambda}{8}$ to tune the column b = column width	D governed by inertia	Adequate to detune response of structure from waves
RAFTS	Not critical Power take-off favour long devices Structure and efficiency favour short devices	$B = 0.7\lambda$ to minimise transmitted energy B = total length of the 3 rafts in a string	Small as convenient structurally and to accommodate power take-off	Some significant mass needed to overcome return power take-off torque but does not govern design
RECTIFIER	Not critical, but 100 m suits construction. May prove beneficial to hydraulically link devices	Best at $\frac{\lambda}{3}$ not much inferior at $\frac{\lambda}{4}$	Operating water depth + H/2	Structural strength requirements govern
DUCKS	Spine:- Several crest lengths needed to achieve force cancellation (approx. $3 \times \lambda$) Duck length not critical	B = D for cylindrical spine	$D \doteqdot \lambda/10$ for highest efficiency	Lowest inertia possible for ducks, but enough ballast for correct trim
AIR BAG	L - small, governed by duct size and bag geometry	$B = \lambda$? for stability in pitch	D governed by structural strength d bag = $\frac{\lambda}{15}$	Ballast needed to overcome buoyancy of bags
WELLS	Circular point absorber L = B	$B = \lambda/4$ for efficiency	Not yet known	Inertia of torus adequate to restrain dome
VICKERS	Not known	Not known	Not known	Mass determined by structure and need to overcome buoyancy of air voids

Note - relations given are only intended to be an approximate guide.

TABLE 1.2 FACTORS WHICH DETERMINE THE CRITICAL DIMENSIONS OF DEVICES

Two important areas remain largely unresearched at the time of reporting. These are manning and maintenance. This is simply because, with the devices themselves at a very early stage of development, work in these areas must still be largely conjectural. Cost estimates for these items have been based on extrapolation from available data in related areas of marine and power technology.

1.2 THE DEVICES

1.2.1 LIST OF DEVICES ASSESSED

	Designation in Text
NEL Oscillating Water Column, developed at the National Engineering Laboratory	NEL Device
Cockerell Rafts, developed by Wavepower Ltd. (WPL)	Rafts
Russell Rectifier, developed by the Hydraulics Research Station (HRS)	Rectifier
Salter Ducks, developed jointly by Edinburgh University and Sea Energy Associates (SEA)	Ducks
French Flexible Bag, developed by Lancaster University	FFB or Air Bag
Wells Oscillator, developed by Queen's University, Belfast	Wells Device
Vickers Oscillating Water Column, developed by Vickers Ltd.	Vickers Device

1.2.2 CLASSIFICATION

1.2.2.1 Objectives of Classification

The devices in this report form a small sample of the possible means of extracting power from the sea, and it is important for these devices to be seen as members of large families of devices. Classification helps to illuminate the fundamental physics of wave power, to highlight the main engineering problems and the possible means of overcoming them. Also, classification will help to ensure that all credible options are given due attention.

Several classification schemes have already been proposed, and the following is based on that suggested to TAG 1 by the Consultants. Firstly, it was decided that a single 'tree' classification was not desirable, as several aspects of devices are best considered independently. For example, whether a device is resonant or non-resonant does not depend on whether it is floating or sitting on the seabed. In practice it is found that there are too many aspects of wave power devices, and too many overlapping and hybrid possibilities for a formal classification to be adequate in all cases. It follows that devices are best classified under several headings, and these have been chosen to meet the objectives for classification set out in the first paragraph. The most obvious features of the devices, such as their position relative to the seabed, do not necessarily form the highest level of classification. Finally it is considered that over-detailed classification will defeat the purpose of the exercise, and would lead ultimately to all devices being put in their own family of one.

1.2.2.2 The Proposed Classification System

A. Classification by Mode of Interaction with the Waves

Devices

Rectifiers
(e.g. Russell Rectifier)

Moving surface modulators
(All other devices in this report)

A.1 Sub-Classification of Moving Surface Modulators by the Nature of the Surface

(Note - the moving surface of water column devices is taken at the inlet to the column)

Moving surface modulator

Resonant

i.e. Surface with mass or added mass using resonance to counteract inertia

Non-Resonant

Light membrane with small added mass (e.g. French)

Rigid surface with mass (e.g. Salter Ducks, Cockerell Rafts)

Flexible or hydraulic interface with large mass and added mass (e.g. NEL, Vickers, Wells)

A.2 Sub-Classification of Moving Surface Modulators by the Position of the Moving Surface

Moving surface modulators

Horizontal

Vertical
(e.g. NEL, French)

Mixed
Salter Ducks
(largely vertical)

Near the surface of the sea (e.g. Cockerell Rafts, Wells)

Immersed (e.g. Vickers)

The importance of this classification is that it determines the relative values of added mass and added damping which apply to the moving surface.

A.3 Sub-Classification by Plan Geometry

Device

Focussing Devices

Line absorber

Axi-symmetric point absorber (eg. Wells, Vickers)

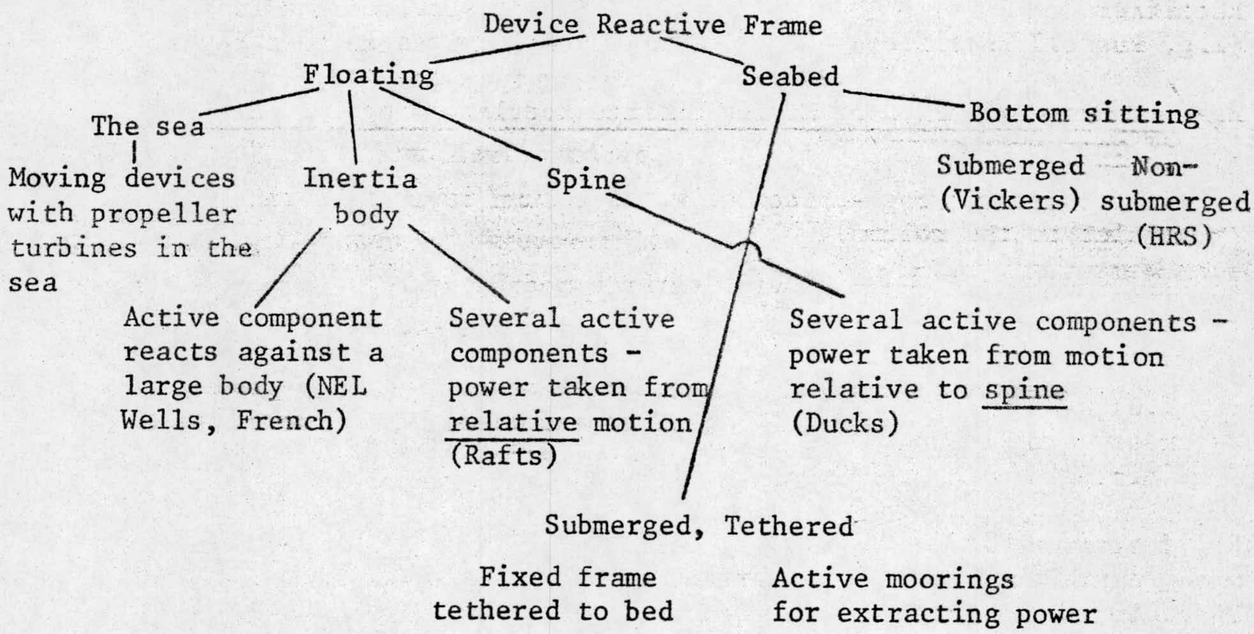
Others, isolated devices, e.g. WPL rafts)

In line with crests (e.g. NEL, Ducks, Rectifier)

Normal to crests, (e.g French)

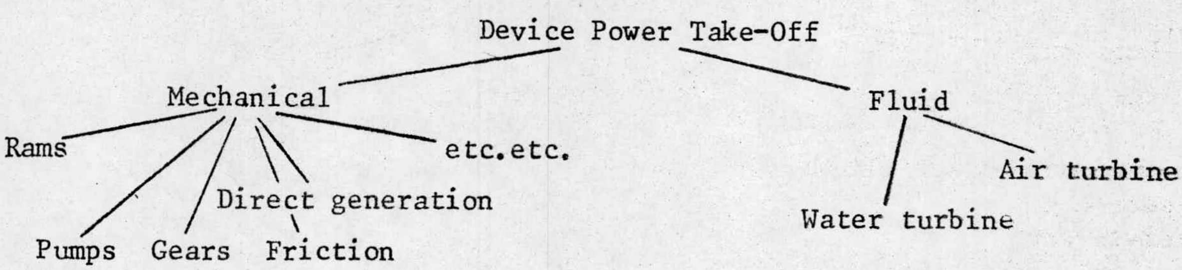
B. Classification by Reactive Frame

This factor determines the nature of the main structure of a device. Structure is known to be the largest cost centre for wave power devices.



C. Classification by Primary Power Take-off

This factor determines many of the engineering problems, such as maintenance and reliability, as well as being an important cost centre.



Fluid power take-off can be further sub-divided into rectified and non-rectified flow. A detailed classification for the power take-off chain has been prepared by TAG 6.

1.2.2.3 Comments on the Classification

Firstly, it can be seen that there are representatives of the majority of sets in the classification scheme. A noteable exception is the submerged tethered device with active moorings. However, taking the sets which are represented, it must be remembered that the number of permutations of the various features are almost limitless, and the best device may yet prove to combine the best features of several devices. As an illustration, the following combinations are all plausible concepts.

a) An NEL device sitting on the seabed.
b) A Duck string aligned normal to the wave crests.
c) An NEL device with non-rectified air flow through the turbines.
d) A Vickers device with a flexible air filled membrane driving an air turbine with non-rectified flow.
e) An Air Bag device sitting on the seabed.

1.3 EXPLANATION OF HYDRODYNAMIC TERMS USED IN THIS REPORT

If, when a body is moving in the sea, all pressures and forces are linearly related to acceleration, velocity and displacement then the system is said to be linear . The majority of hydrodynamic theory is based on this assumption, which has the considerable advantage of allowing superposition of solutions for various assumed conditions. Also solutions can be reversed; for example, the waves radiated by forced oscillation are equal to the waves absorbed by a body under reversed forces. Linear theory holds for small displacements and small wave heights.

As displacements and wave heights increase the hydrodynamic interaction becomes increasingly non-linear due to geometric non-linearities and the influence of drag forces. In general wave power devices behave less efficiently in the non-linear regime.

For the linear regime three concepts aid understanding of the interaction of the device with the sea. For a moving surface, if the variation of the total pressure force in time is known, then the force can be split into components proportional to acceleration, velocity and displacement. For each particular motion the action of the fluid can therefore be thought of as equivalent to added mass, added damping, and added (or buoyancy) stiffness. These parameters are in fact dependent on the period of oscillation in cyclic movement. Thus the principle of a resonant device is that the inertia of the moving surface plus added mass is counteracted at resonant frequency by spring plus added stiffness forces. The geometry and size of a device determines the balance between these forces, and hence the resonant frequency.

Using the principle of reversibility it can be seen that high added damping leads to a large capacity to absorb power. For optimum performance the power take-off damping must be equal to the added, or hydrodynamic, damping. This is analogous to impedance matching in electrical circuits.

Hence it can be seen that, in the linear range at least, the three parameters, added mass, added damping, and added stiffness, determine the optimum size of the device and the required power take-off characteristics.

CHAPTER 2.0 - STATE OF DEVELOPMENT AT JULY 1978

This chapter consists of a summary presentation in tabular form of the current state of development of all devices.

Notes explaining Table 2.1

a) A full description of the stage of development of each device is given in Chapters 4-10. It is not intended that judgement of a particular device should be made on the basis of Table 2.1. The table is intended to act as a reminder of, but not a substitute for, the Consultants' conclusions on the stage of development of each device.

b) Brief notes and a star rating are included in the table. Such a concise means of presentation is necessarily imprecise for particular items, but overall the picture should be correct.

c) The ratings refer to the state of development to a somewhat arbitrary scale intended to represent 'adequacy' relative to the overall level of progress in the wave power programme. The scale therefore includes implicitly the Consultants' judgement of an appropriate balance between aspects such as laboratory testing and design study, etc.

d) It follows from the above that the ratings are not necessarily a measure of the effort that has gone into a particular aspect of a device. A device which is inherently simple will be rated highly (more stars) after relatively little work, because the total effort needed to develop it is likely to be small. Conversely, a complex device which is difficult to engineer may only merit a low rating even after a considerable expenditure of effort. The ratings are inversely related to the work to be done, rather than being a measure of the effort expended to date.

e) A high rating in most areas implies that work is well advanced towards achieving a technically sound system for which a reliable assessment of potential can be made. A high rating is not intended to indicate that the final assessment will be favourable, as the device may prove to be uneconomic, or suffer from other overriding disadvantages.

PLEASE READ PAGE BEFORE READING THIS TA

	NEL	WPL	HRS	SEA/SALTER	FRENCH	WELLS	VICKERS	
1. Theoretical Appreciation	Significant ****	Good ****	Moderate **	Ducks - Very Good. ***** Duck Spine interaction - very early stages *	Moderate **	Moderate **	Very preliminary *	1.
2. Laboratory or Tank Testing of Prime Mover 2D - Monochromatic Waves	Yes ****	Yes ***	Yes **	Technique (but does not test whole system) *****	Narrow tank tests with monochromatic seas only **	Yes **	Only on an exploratory model	2.
3. 2D - Random Waves	Yes ****	Yes ***	No -	As above *****		No -	No -	3.
4. 3D - Wide Tank Models including inclined seas	Yes ***	Yes - but primitive facility only *	No -	Spine only Duck model not yet available *	No random wave test -	No -	No -	4.
5. Larger Scale (Lake or Sea Tests) Prime Mover	No -	Solent Tests in progress **	None -	Loch Ness trials in progress **	None -	Yes (not wholly successful) *	No -	5.
6. Proof of feasibility of structure and ability to survive	Adequate *****	Incomplete **	Adequate *****	Not proven -	Torn air bags are a potential haz to be researched **	Probably feasible ***	Feasible by nature of concept ***	6.
7. Structure Design of a Prototype. Structure for costing	Adequate at this stage ****	Preliminary **	Adequate but loading uncertain ***	Incomplete *	Preliminary **	Exploratory *	No -	7.
8. Moorings	Current proposals unsatisfactory **	Current proposals unsatisfactory *	not applicable *****	No proposals -	Proposals not proven -	No proposals -	Not applicable *****	8.
9. Prototype Power Take-off. Proof of feasibility	Adequate at this stage *****	Incomplete **	Adequate (but note seaweed ingestion problem) *****	Solutions proposed - scarcely satisfactory	Adequate at this stage *****	Partial ***	Adequate *****	9.
10. Prototype Power Take-off. Design for costing	Conceived but only very preliminary design ***	Modest amount done much remains to be done **	Definition of heads and flows lacking **	Components not designed *	Very preliminary *	Exploratory design *	No -	10.
11. Optimisation of Configuration	Several shapes explored and tested but much more to do ***	Possibly half the options now explored ***	Two shapes explored but more to be done **	2D work complete 3D work starting ***	Preliminary only *	Several shapes explored and tested but much more to do *	No -	11.
12. Optimisation of Structure	In hand, but fairly early stages ***	Little done *	Modest **	Not yet relevant -	Preliminary only *	No -	No -	12.
13. Major Unsolved Problem areas Structural	None	Hinge Force Problem	Float out strength	Spine design unsolved	Damage control	Not known	None identified	13.
14. Major Unsolved Problem areas Power Take-off	-Turbine efficiency -Power collection	Cost and reliability	Seaweed ingestion and efficiency	Accessibility, environment, maintenance, reliability	Not known	Power collection	Efficiency	14.
15. Progress in Attacking High Cost Centres	Some progress but no major breakthrough yet ***	Very little *	Some progress, but limited **	Not yet relevant -	Too early -	Too early -	Too early -	15.
16. Potentially Attractive Variants worth exploring	Major variants still unexplored ***	Several, but potential advance fairly limited *	Few *	Not a lot of obvious ground to explore **	A few **	? -	? -	16.
17. Any Special Notes	Comparatively inexpensive to develop if successful -	Will require very extensive development programme. -	Siting requirements severely limit total available energy resource -	An attractive concept but unsolved problems remain formidable -	Offers possibility of breaking through the size barrier -	Turbine design worth investigating independent of device	Comparatively inexpensive to develop if successful -	17.
18. Expenditure to date								

Table 2.1 STATE OF DEVELOPMENT

NOTE - PLEASE READ PAGE BEFORE READING THIS TABLE

CHAPTER 3.0 - BASIS OF THE CONSULTANTS TECHNICAL ASSESSMENT

3.1 CONSULTATION AND INTERACTION

3.1.1 CONSULTATION WITH DEVICE TEAMS

In the case of the first five devices reported on, the Consultants have had more or less regular contact with the Device Teams over a period of about a year. Discussions have taken place on most technical aspects of the devices, and the Consultants have thus been aware of the development of the devices and of the main problem areas. Contact with the remaining Teams have been less, in proportion to their involvement in the WESC programme. Special consultations took place with Teams in May and June of this year, immediately prior to the final phase of the assessment exercise and all aspects of the designs were discussed. The Teams subsequently supplied the Consultants with their latest information on their devices.

3.1.2 WORK WITH TECHNICAL ADVISORY GROUPS (TAGS)

The Consultants attend meetings of all technical advisory groups, participate in the work programmes of the groups, and from time to time prepare formal reports for them. The Consultants are thus fully conversant with this aspect of the wave energy programme and the present report draws on the outcome of the TAGS work as appropriate.

3.2 REFERENCE DESIGNS

The assessment and costing exercise for each device is based on a Reference Design. These designs are as far as possible the designs submitted by the Teams, carried out to general criteria previously agreed with the Consultants. Where any design submitted was incomplete or deficient, the Consultants have attempted to carry out the work necessary to complete the design. The Reference Designs as calculated and drawn are, of course,no better than the information on which they are based, and are to be considered as recording a transitory stage in the development of each device, valuable for carrying out this assessment, and as exercises which highlight problems and crystallise thinking in important areas.

3.3 CONSULTANTS BACK-UP STUDIES

Following their report of 1977 and in anticipation of this report, the Consultants carried out studies in a number of areas to prepare the ground for this report. Generally this work is not separately reported, but an exception is made in Chapters 12 and 13 concerning system simulation, which is a very important input to the assessment of each device.

3.4 PERFORMANCE SPECIFICATION

3.4.1 DEVICE SIZE

The wave period corresponding to peak efficiency, for most devices, is a function of device size. There are several criteria which can be used to decide the 'best' size for a device. The device may be sized for best cost effectiveness irrespective of resource utilisation, or for larger output at lower cost effectiveness to increase the total resource available, or to meet some practical consideration such as the availability of construction docks of a particular draught. In practice. each Team has aimed at its own compromise; resonant devices have generally been designed for maximum efficiency in the range of 8-10 seconds wave period. This was thought to be reasonably close to the best design point for cost effectiveness, whilst making good use of the available resource, but this is an area still requiring investigation.

It should be noted that the criteria for optimising the devices are still the object of speculation. Clearly the final size, and even type, of wave power device will depend on the goals set. In particular the value placed on the product (generated electricity) needs to be known, and the variation in this value depending on its firmness (reliability) and on when it is delivered (i.e. summer or winter). Furthermore, it must be known whether maximum use is to be made of the Nation's total wave power resource, and if so the appropriate criteria for cost effectiveness. These questions, until recently somewhat academic, are becoming pertinent as the time approaches for a selection between devices.

The Consultants have accepted the Device Teams' chosen size, and have then gone on to fix the power take-off plant ratings on the basis set out and discussed in 3.4.2.

3.4.2 POWER GENERATION AND TRANSMISSION - DETERMINATION OF PLANT RATINGS

3.4.2.1 General

This area presented some of the most intractible problems to the Consultants. Firstly, the amount of plant installed for a given size of device must, in most cases, be determined from study of the economics of the system. It is clearly not desirable to install plant which is capable of handling the absolute maximum of power which is available, perhaps for only a few seconds during the year. Secondly, producing specifications for plant is extremely complex for random inputs, requiring as it does, short and long term statistical descriptions of several inter-related parameters such as velocity, pressure, forces and torques. These parameters are all required to determine strengths, fatigue lives, wear rates and efficiencies. Thirdly, the input parameters are directly dependent on the characteristics of the plant, and most importantly the control system, and an iterative approach is needed to specification and design. Fourthly, the Teams have only just begun to study these problems and are not yet able to propose a comprehensive specification for their plant.

Fortunately, it is likely that the technical appraisal and costing of the devices will not be unduly sensitive to errors made at this stage in defining plant specifications. This is because the present technical appraisal largely concerns the feasibility of types of plant, rather than the correctness of a particular size of plant, and secondly because the total costs of devices appear not to be sensitive to errors in estimates for those components whose 'sizes' are most open to doubt. Even so, it is clear that the area of plant specification still requires urgent attention.

In the circumstances, the Consultants have been forced to base the specification for plant largely on power handling capacity, with only approximate estimates of flows, velocities, heads and torques, etc. In this chapter a description is given of the power levels for each stage in the power chain of each device. More detailed analysis including the other parameters are given in Chapters 4 to 12.

It will be seen that determining plant ratings for the first stages of the power take-off chain is particularly difficult, but the later stages, e.g. transmission, are more reliable.

The Consultants have sized the plant on the basis of computer simulation of the influence of plant rating on device productivity, and on the following assessment of mean and peak power at each stage in the power chain.

The output of the plant rating design exercise for four of the devices is given in Tables 3.1 to 3.6.

3.4.2.2 Computer Simulation

Results have been obtained from computer simulation of annual device productivity with various levels of limiting mean power acceptance (i.e. cut-off levels). In their previous report the Consultants indicated that civil costs are likely to be considerably greater than mechanical and electrical costs. Hence for maximum economy, sufficient plant should be installed in a given-size of device to allow for very nearly the full potential to be realised. This gives an approximate guide to the correct level of planting. A fuller account of this study is given in Chapter 13.

3.4.2.3 Factors Affecting the Power Level at Different Stages in a Power Chain

The variation of power level and hence the required plant rating in a matched power chain depend on the following factors:

a) Nature of the incident power in the sea.
b) Hydrodynamic conversion efficiency and self-limiting characteristics of the wave energy convertor.
c) The nature of the mechanical power abstraction, particularly the self-limiting characteristics of the power take-off machine, which in turn depend on the control system.
d) Position down the power chain. The nearer to the device/sea interface the higher the rating needed.
e) Energy storage, smoothing by flywheels, accumulators, reservoirs, etc. (temporal smoothing).
f) Integration of a number of inputs (spatial smoothing).
g) Efficiency of links in the power chain.
h) Special power limiting devices included in the power chain, e.g. by-pass valves, clutches, etc.

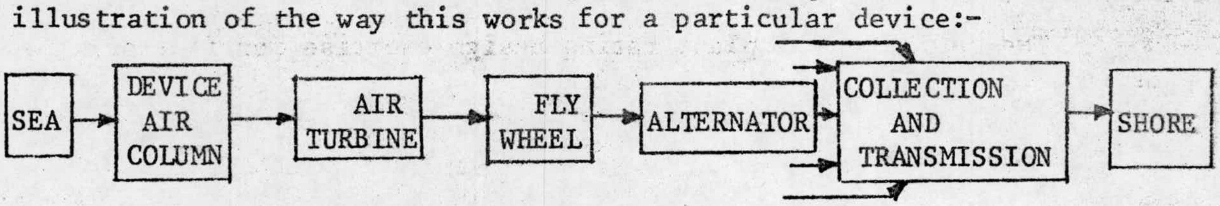
3.4.2.4 Variation in Power Level in the Sea

This heads the list in the previous paragraph, and is at the heart of the power take-off problem. Variations in power levels can be categorised under five headings.

a) Long-term variation due to changes in climate.
b) Annual variation - good and bad years.
c) Seasonal variations - winter, spring, summer, autumn.
d) Variation from hour to hour - changes in sea state. It is commonly assumed that sea states stay nearly constant for 3 hour periods.
e) Variations in power in a given constant sea state.

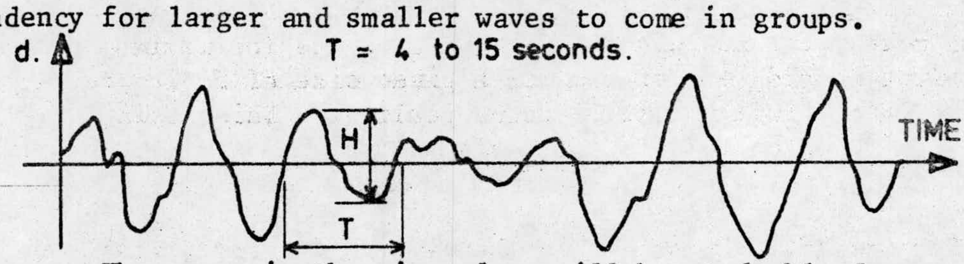
3.4.2.5 Variation in Power seen by Different Components in the Power Chain in a Constant Sea State

The power seen by each component in the power chain depends on both the characteristics of the prevailing sea, and on the factors listed in 3.4.2.3. The Oscillating Water Column provides a convenient illustration of the way this works for a particular device:-

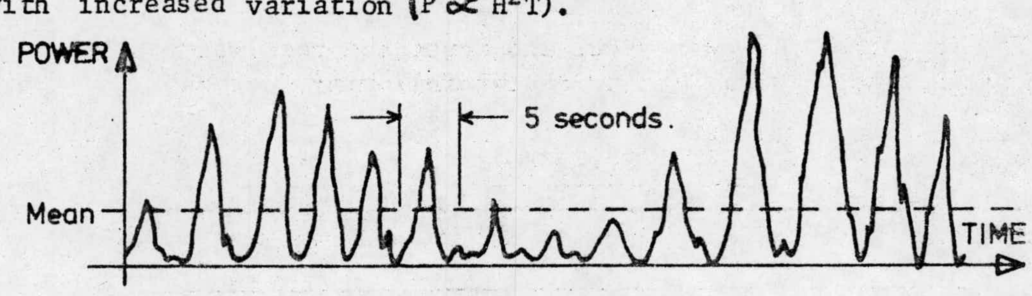


Power Chain for an Oscillating Water Column

The wave height and period will vary from wave to wave, with a tendency for larger and smaller waves to come in groups.



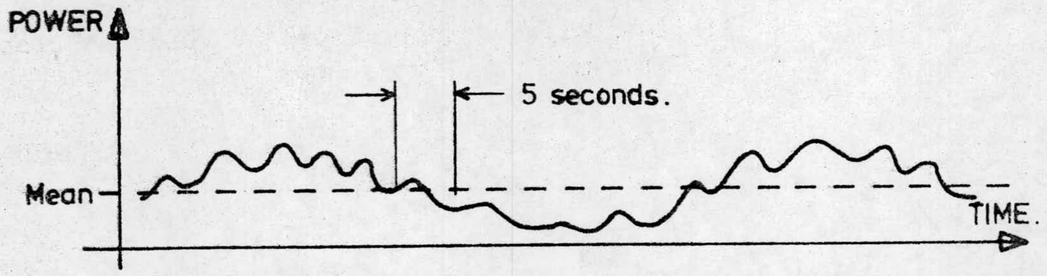
The power in the air column will be at double frequency and with increased variation ($P \propto H^2T$).



This is the power input that has to be accepted by the first stage of the power take-off, in this case the air turbine. Even for a regular sea it can be seen that the peak instantaneous incident power will be twice the mean. In a random sea this will increase to at least 3 or 4 times the mean, with occasional extreme values even higher. It can be seen that the power arrives in pulses at about 5 second intervals.

The turbine has inherent power limiting characteristics which depend on its stall characteristics and on control of guide vanes to throttle air flow, Hence, the most extreme incident power levels will not appear in the output.

The function of the flywheel is principally to smooth the turbine output. (The complex question of the influence of flywheel size on overall turbine efficiency will not be discussed here). It would not be feasible to provide a flywheel large enough to provide constant power over many minutes, but the cyclic variations within a wave can be reasonably averaged out.



Peak power levels here may be perhaps twice the mean level at this stage.

The output of the alternator would show similar characteristics - slow variations from minute to minute with only a small ripple corresponding to twice the wave frequency.

The next stage in the chain is shown as a collection system. The input here would be from many columns and depending on their number, smoothing would be accomplished by virtue of the spatial variation of power along a long line of many devices. If the system collected power from several kilometres of columns (i.e. many wave crest lengths), the output would be nearly constant over a period of several hours. If the system collects over only one or two hundred metres (perhaps one complete device) the output would be much closer to the output of a single alternator, that is, much more variable.

3.4.2.6 A Comparison between Mechanical and Turbine Primary Power Take-off

One important contrast between the alternatives of mechanical and turbine systems (either air or water) for primary power take-off is as follows:-

a) A turbine can easily reject excess power using throttles, by-pass valves, and to a certain extent its own operating characteristics. Hence choice of turbine size can be based on an economic appraisal.

b) In a mechanical device it is difficult to limit the power accepted by the primary power take-off. Non-optimum damping and the non-linear hydrodynamic characteristics of the device can help, but the primary power take-off will be sized on a criterion of survival. Furthermore, it may also be necessary to incorporate some means of power shedding as soon as possible in the power chain to reduce the necessary ratings for the generating and transmission plant.

3.4.2.7 Definition of Power Rating Used in this Chapter

Unless otherwise stated, power rating in this Chapter implies the maximum power level which a piece of plant is capable of delivering (output).

In calm and moderate seas the power level may approach the rating only rarely, if at all, and only for a few seconds at a time.

In stormy seas, the power level may be frequently or even continuously at the rated capacity, depending on the position of the plant in the power chain.

This definition should be distinguished from the "Design Point", the latter being the power level for maximum efficiency.

DERIVATION OF PLANT RATINGS - COCKERELL RAFTS - WPL REFERENCE DESIGN 1978

Sea → Device → Gears → Pistons → Hydraulics & Accumulator → Bypass valve → Turbogenerators → Collection & Transmission → Shore

MODE 1 - Calm to moderate seas - Damping optimised - Bypass valve shut
(MODE 2 - Moderate to storm seas - Damping not optimised - Bypass valve open)

LINK	DERIVATION						Peak Rating (instantaneous per raft)	Max. Hourly mean per raft
	Efficiency at full load	Hourly mean power in kw/m	VARIABILITY FACTOR (MAX/MEAN) within each wave period	x	for longer term (wave group)	Peak Output in kw/m (instantaneous).		
Sea	-	≈ 63 kW/m (64 to 1500 kw/m)						
Device (hinge power)	0.8 max. (0.8 to 0.08 varies)	50 (100 ??)	2.0	X	2.0	200 (400)	10 MW (20 MW)	2.5 MW
Gears	.90	45 (90 ??)	2.0	X	2.0	180 (360 ??)	9 MW (18 ??)*	2.3 MW
Pistons	.98	44 (88 ??)	2.0	X	2.0	176 (352 ??)	8.8 MW (17.6 ??)*	2.2 MW
Hydraulic + accumulators + bypass valves	.97	43 (86 ??)	1.1	X	2.0	95 (190 ??)	4.8 MW (9.6 ??)*	2.2 MW
Bypass valve shut (open)	1,00 (1.0 to 0.0 variable)	43 (43)	1.1	X	2.0	95 (95)	4.8 MW (4.8 MW)	2.2 MW (2.2 MW)
Turbogenerator	.97 (turbine) X 0.97 (generator)	38	1.1	X	1.5	63	3.2 MW (3.2 MW)	1.9 MW (1.9 MW)
Collection & transmission to Skye	.92	35.	1.0	X	1.05	37	1.9 MW (1.9 MW)	1.8 MW (1.8 MW)
Skye to Perth + inversion	.92	32	1.0	X	1.05	34	1.7 MW	1.6 MW

* NOTE - for storm conditions these components are designed on a criteria of speed and volume flow rather than power rating.

TABLE 3.3

Table {3.1 / 3.2} missing!

DERIVATION OF PLANT RATINGS - HRS DEVICE WITH TWO WATER TURBINE GENERATOR SETS

Sea → Device → Reservoir → Turbines → Generators → Power collection conversion transmission

LINK	DERIVATION					Peak Rating per 100m unit MW	Hourly mean rating MW
	Efficiency at full load	Hourly mean power kW/m	VARIABILITY FACTOR MAX/MEAN		Peak Output (instantaneous) kW/m		
			within each wave period	for longer term 'wave groups'			
Sea	-	100	-	-	-	-	-
Device Captured power in device	.40 present conversion efficiency with cut-off in storm seas	40	1.5 say	1.0 limited by o/spilling cut-off	60	6.0	4.0
Power between reservoirs (smoothed)	.96	38.5	*1.2	1.0	46	4.6	3.85
Turbines	.90	34.5	1.2	1.0	41	4.1	3.45
Generators	.95	33	1.2	1.0	39.5	3.95	3.30
Cables to shore located converter station	.92	30	1.05	1.0	31.5	3.15	3.0

* NOTE Variability reduced by reservoir storage (upper and lower) so that factor limited to the inter-reservoir head variation within a wave cycle. Reservoir cannot store for more than one wave cycle.

TABLE 3.4

DERIVATION OF PLANT RATINGS - FRENCH FLEXIBLE BAG REFERENCE DESIGN 1978

SEA → BAG DEVICE → AIR TURBINE → GENERATOR → COLLECTION & TRANSMISSION

LINK	DERIVATION: Efficiency at full load	DERIVATION: VARIABILITY FACTOR: within each wave period	X	for longer term 'wave groups'	Peak Rating (instantaneous)	Max. Hourly mean power
Sea						
Captured Power in ½ of device		1.7 (some smoothing along device)	X	4.0 ?		
Air Turbines in ½ of device	0.80	1.7	X	1.7 after throttling	10.4 MW per ½ device	3.6 MW per ½ device
Generator	0.96	1.05 2 halves joined together + inertia	X	1.8	13.1 MW per device	6.9 MW per device
Collection & transmission to Skye	0.92	1.0	X	1.05	6.7 MW per device	6.4 MW per device
Skye → Perth + inversion	0.92	1.0	X	1.05	6.1 MW	5.8 MW

TABLE 3.6

DERIVATION OF PLANT RATINGS - SALTER DUCKS - SEA REFERENCE DESIGN 1978

Note - the tabulated figures are approximate.

Sea → Device → Gears → Pumps → Hydraulic + Accumulators → Turbo-generator → Collection and Transmission → Shore

MODE 1 - Calm to moderate seas - Torque level optimised

(MODE 2 - Moderate to storm seas - Damping off optimum - Torque level reduced - Some pumps partially or fully short circuited).

Link	Derivation				Peak Rating (instantaneous) per 24m Duck.	MAX. Hourly mean per 24m Duck
	Efficiency at full load	Hourly mean power in KW/m	Variability Factor (MAX/Mean) within each wave period X for longer term 'wave groups'	Peak Output in KW/m (instantaneous)		
Sea		94 kW/m (95 - 1500 kW/m)		can be 10,000 kw/m or more		
Captured Power	up to 0.8 (low)	75 (75) ?	2.0 X 2.0 ?	300. kW/m ? (300)	7.2 MW ?	1.8 MW
Gears*	0.90	67.5 (67.5 average,*)?	2.0 X 2.0 ?	270 W/m (270)?	6.5 MW ?	1.6MW
Pumps*	0.90	60.8 (60.8 average*)?	2.0 X 2.0 ?	243 kw/m ? (243)	5.8 MW ?	1.5 MW
Hydraulic circuit & accumulators	0.95	57.7 (57.7)	1.0 X 1.4 (smoothing & many outputs coalesced).	81 kw/m (81)	1.9 MW	1.4 MW
Turbogenerator	0.92 X 0.96 (turbine) (generator)	51.0 (51.0)	1.0 X 1.4	71 (71)	1.7 MW	1.2 MW
Collection & transmission to Skye	0.92	46.9 (46.9)	1.0 X 1.05	49kw/m (49)	1.2 MW	1.13 MW
Transmission & inversion at Perth	0.92	43	1.0 X 1.05	45	1.1 MW	1.0 MW

* NOTE - in MODE 2 some pumps will be partially or fully short circuited to decrease the duck torque.

TABLE 3.5

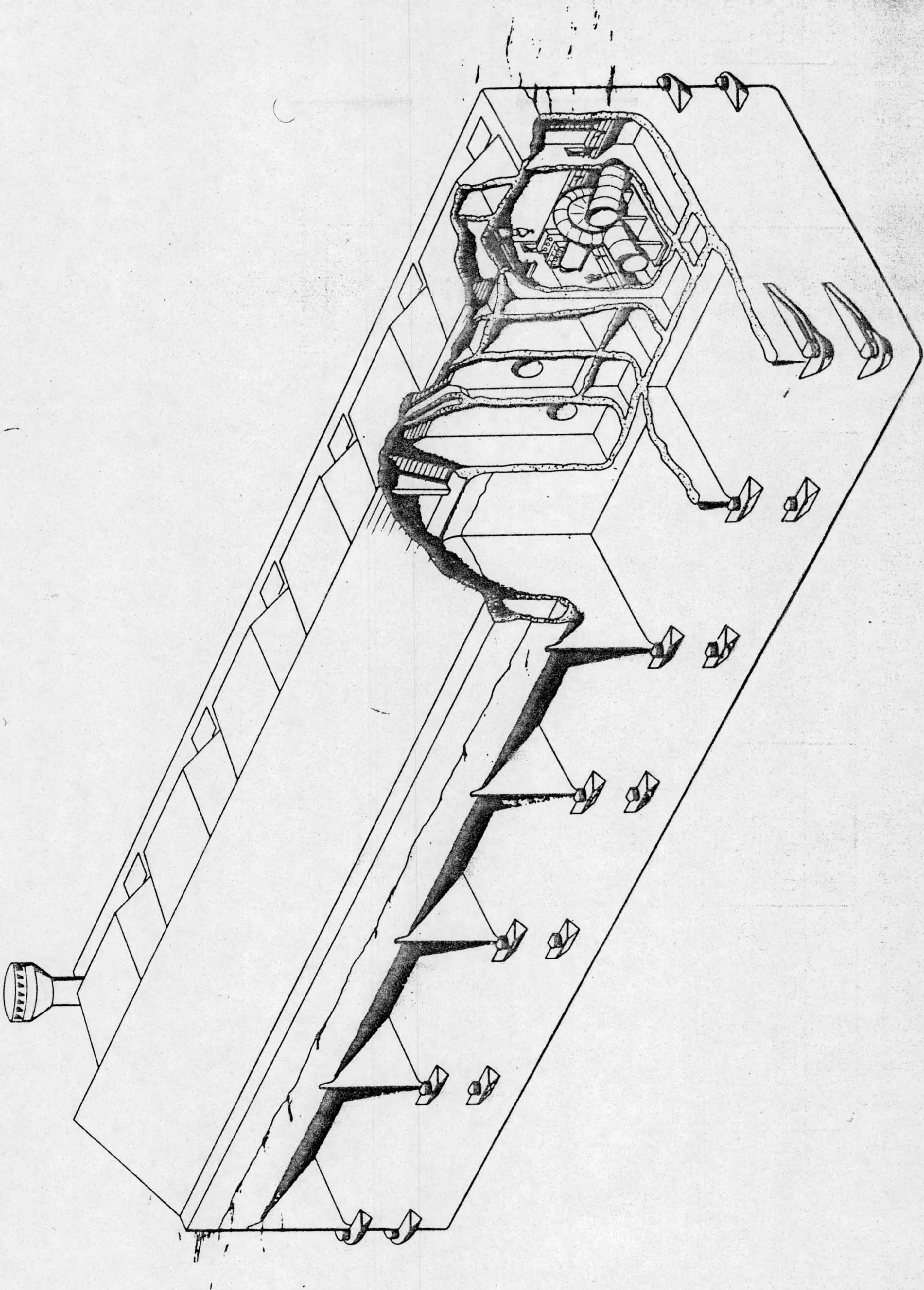


FIG 4·1.

CHAPTER 4.0 NEL OSCILLATING WATER COLUMN

4.1 GENERAL DESCRIPTION OF THE DEVICE

4.1.1 OSCILLATING AIR COLUMN

If a partially submerged vessel is open to the sea at the bottom or the side, the column of water trapped by the vessel will oscillate in response to passing or incident waves. This oscillating column of water may be made to do useful work by using it as a piston to pump air through an air turbine. A wide variety of different configurations offer themselves as practical devices for accomplishing this, for whilst this type of device is simple in concept, the combinations of parameters that determine overall efficiency are particularly complex. Chapter 2 includes a description of different types of reference frame for wave power devices. In principle all are possible for oscillating water columns, and devices based on three different reference frames are currently under consideration by the NEL Team. Only one however, has thus far been studied in depth under a WESC development contract, and this is the Reference Design assessed in this report.

4.1.2 NEL DEVICE

This device uses the inertia of a large body to react the air column, and also to reflect part of the incident waves. The NEL device is a massive floating structure approximately equal in depth and width, and about three times as long as wide, see Fig. 4.1. The front face contains a row of working elements in which oscillating water columns continuously pump air through massive ducts to drive low pressure air turbines linked directly to generators. The system of ducts, turbines and generators is located behind the front chambers inside the main body of the device. A large two way rectifier system is included in the ducts to the turbine to provide continuous undirectional air flow. This somewhat unusual cross-section of the device has been arrived at as a result of a thorough development programme on this specific type of air column device.

4.1.3 PRINCIPLES OF OPERATION

4.1.3.1 General

The action of the water column can be compared to that of a classical mass/spring/damping system, where the parameters of the system are as follows:-

a) Mass - the resistance of the water column to acceleration due to its own inertia and that of the associated mass of water at the mouth of the column.

b) Spring - the resistance of the column to displacement upwards or downwards. This depends on the weight of the water lifted for a given displacement, and hence the area of the water column at its top surface. The air normally acts almost incompressibly and contributes little to the system stiffness for practical damping values.

c) Damping - the velocity proportional component of resistance to motion, supplied by the air turbines, and to a lesser extent hydrodynamic losses in the water column.

The water column is therefore to be expected to exhibit many of the response characteristics of a simple dynamic system, but the analogy with a one degree of freedom system is only approximate. However, it does exhibit a form of resonance, and must be tuned to an appropriate wave period for maximum efficiency in typical conditions.

Oscillation of a water column itself generates (radiates) waves. This is radiation damping. If the waves are radiated in all directions, they represent a major loss of energy. To maximise energy trapped, the NEL device is made directional by introducing a reflecting surface which directs the radiated waves back into the incident wave train, and also reflects the incident wave. For maximum efficiency the radiated and reflected components, which are in antiphase, cancel. To achieve this the damping applied by the air turbine must equal the hydrodynamic damping. This reflecting surface is the fundamental distinctive feature of the NEL Reference Design, and increases the maximum possible efficiency from 50% for a symmetric device, to 100%.

Overall, the characteristics of the floating device also parallel a mass/spring/damping system, but with three degrees of freedom; pitch, heave and surge. Hence the key parameters are:-

a) Mass - the mass and rotational inertia of the body plus the associated added mass of water which tends to move with the devices. This added mass is not the same for the three degrees of freedom.

b) Stiffness - buoyancy stiffness (note that this is zero for surge).

c) Damping - largely caused by radiated waves. For large bodies losses due to drag tend to be very small.

The main structure provides the inertial reactive frame for the system and must therefore be proportioned such that the motion of the device will be detuned from the operating waves.

4.1.3.2 OPERATING CYLE (See Fig. 4.3)

Separate inlet and outlet ducts lead to and from the turbine. Each of these is provided with two rectifying valves, one connecting to the air space over the water column and the other to atmosphere. The water column oscillations causes the pressure of the enclosed air to alternate above and below atmospheric. The induced air flow from above the water column to atmosphere and vice versa is controlled by the rectifier in such a way that it passes through the turbine in a constant direction.

4.2 STATE OF DEVELOPMENT

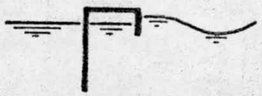
4.2.1 GENERAL

As a result their review of the then existing wave power devices, carried out in 1974, NEL decided that a device based on the oscillating water column offered good prospects and in 1976 they received a contract from WESC to undertake preliminary researches. A number of possible shapes and arrangements were studied and tested in a narrow tank with the object of finding a configuration that offered good efficiency over a reasonable band width. Figure 4.2 on page illustrates some of

①

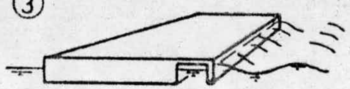
Basic concept - poor efficiency

②



Back reflector - good efficiency

③



Horizontal reflector - poor efficiency

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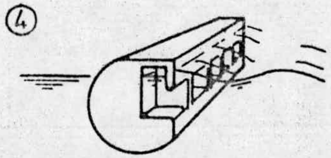
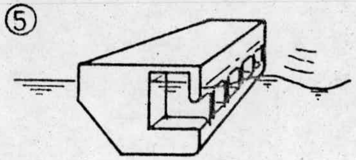


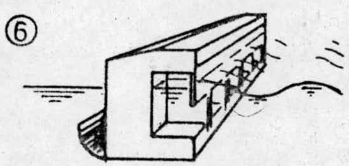
Cylindrical model - poor efficiency

⑤



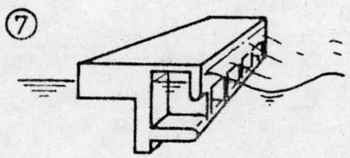
Best model performance at June 1977 - basis of first Reference Design

⑥



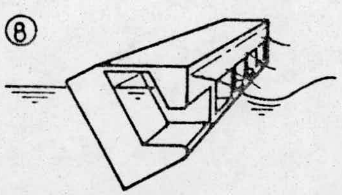
First Reference Design 1977 derived from best model

⑦



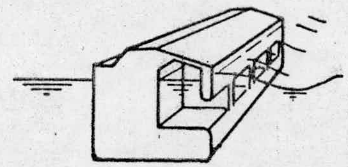
Minimum displacement model Performance good but hard to construct

⑧



PGA 16 Model suggested by RDP Lower concrete weight but performance unsatisfactory

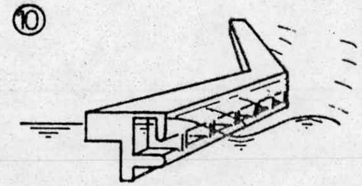
⑨



1978 Reference Design - good performance allied to practical construction shape

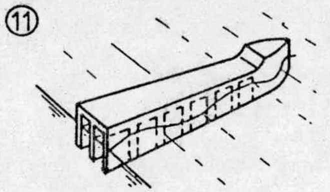
Variants Proposed for Examination in the Ongoing Work

⑩



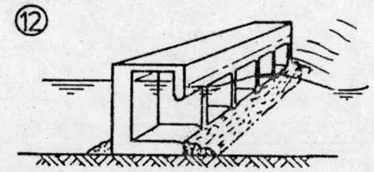
Taking advantage of chevron shape for easier stability

⑪



Line ahead device

⑫



Bottom mounted - design study only model under investigation now by R.D. and Partners

⑬

PROGRESS OF DEVELOPMENT

FIGURE 4.2

the options that were examined. The Team rejected the symmetrical option (1) at an early stage, as having an inherently low efficiency in extracting power across an incident wave front. The alternative arrangement (2) using a deep back face appeared to be much more attractive as a power extractor. However, it clearly presented engineering problems at the prototype scale, and an alternative arrangement (3) was investigated in which the deep back face was replaced by a horizontal surface which would be more practical to construct. The efficiency of this arrangement was found to be poor and this ideas was not pursued as a first choice, and the programme was directed to identifying a suitable shape for a floating inertial solution. Many shapes (including the cylindrical arrangement of (4) were tested. Since the only test facility at this time was a narrow tank, the shape had to meet all the performance requirements in a two dimensional solution. Arrangement (5) was the best performing model at June 1977 and was the basis of the first reference design of August 1977, shown in (6).

Up to the end of 1977 the main development work was confined to narrow tank testing in monochromatic and random waves. The availability of the Salter wide tank early in 1978 allowed a certain amount of controlled development testing of genuine three dimensional models to begin. At about the same time Consulting Engineers Roxborough Dinardo were brought into the Device Team to provide the civil engineering support. The six months to June 1978 has been spent primarily in attempting to improve the cost effectiveness of the basic 2D device. Starting from the criteria of cheap construction cost and reduced mass, with minimum loss in performance, shapes (7), (8), and (9) were designed and tested, with the varying success noted in the table.

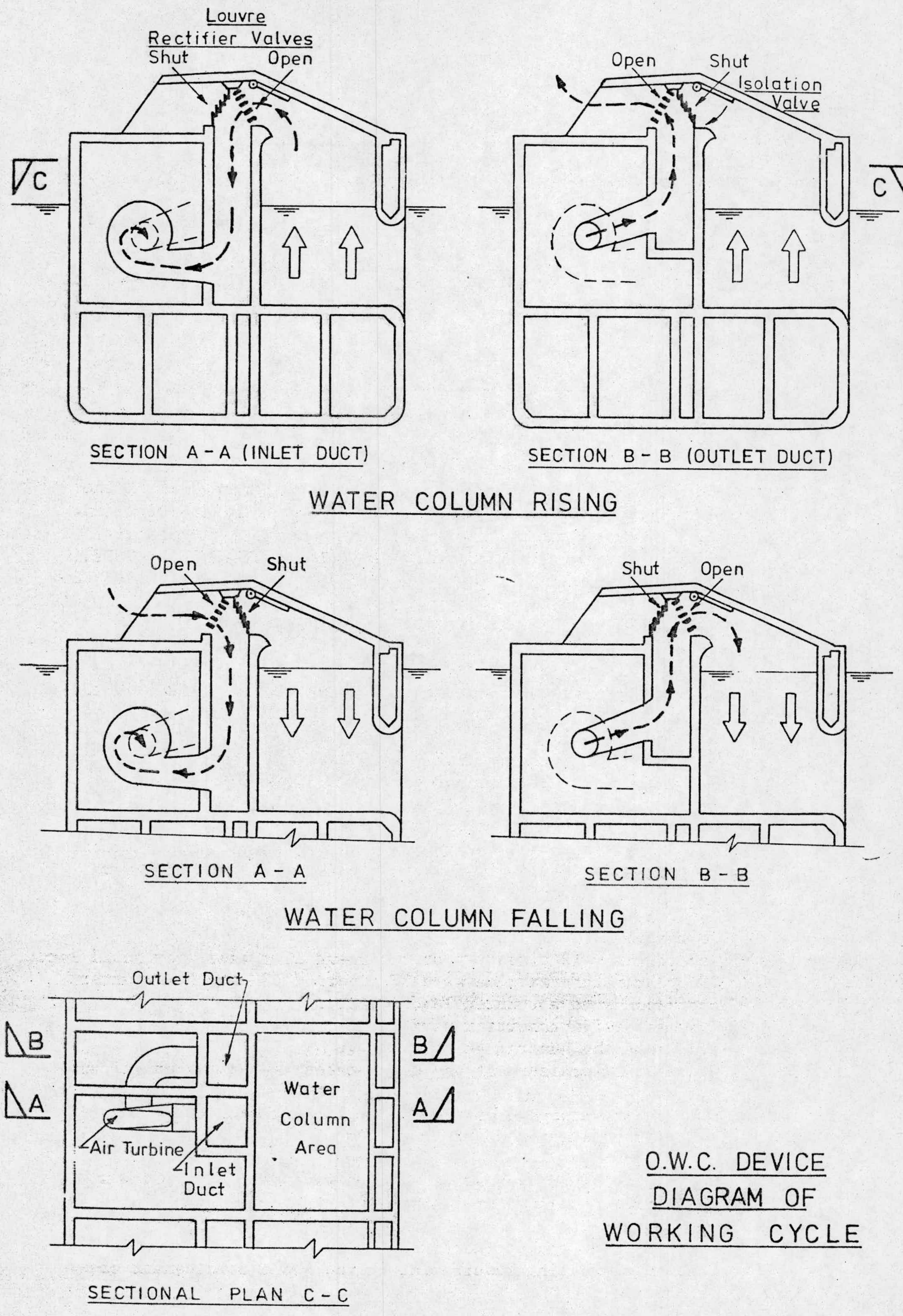
The most striking feature of the last year of development work is that there has been no major breakthrough. The Reference Design of 1978 is essentially similar to that of 1977, albeit better researched, designed, and developed.

4.2.2 TEST DATA

Designs have been tested in both monochromatic and random waves and in the model scale real sea in the Salter tank. There is adequate efficiency data for this stage of the programme. The wide tank tests were used to examine the effects of varying lengths of whole floating units, different responses in end cells, directional effects, mooring forces, water ingress and extreme sea state behaviour. These were carried out on 1/100 and 1/150 scale models.

4.2.3 THEORETICAL STUDIES

The Device Team had been backed by idealised mathematical modelling of the OWC from an early stage. The behaviour of the actual floating device is extremely complex, and the understanding of it is based on a mixture of mathematics, parametric studies, observation, and intuition. As testing moves further into three dimensions, the interactive action of the sea on the device and on the water column will become even more complex and development will probably depend primarily on parametric studies.



O.W.C. DEVICE
DIAGRAM OF
WORKING CYCLE

FIGURE 4.3

4.2.4 ENGINEERING DEVELOPMENT

In the 1978 Reference Design, research testing and engineering development are in satisfactory balance. It is anticipated that this balance will be maintained in an ongoing programme.

4.2.5 UNEXPLORED OPTIONS

Because so many different types and variants are available for the reference frame, and because most of the work done thus far has been confined to two dimensions, a great deal of ground remains unexplored. Options (11) and (12) of Fig. 4.2 are two that will be examined in the near future. Option (1) in its point absorber form is the subject of a separate programme at Belfast.

DRAFT

4.3 DEVELOPMENT OF THE REFERENCE DESIGN

4.3.1 GENERAL

The recent development of the OWC device has included considerable input on the structural and constructional aspects and the Reference Design was decided upon at an early enough stage to allow tank testing to prove its performance and efficiency. Much of the material that follows is based on Report No. PR2: Y5/DEY/2 - "WAVE ENERGY STUDY, NEL OSCILLATING WATER COLUMN, 100MW POWER STATION, 2ND INTERIM REFERENCE DESIGN" which the Device Team and their Consultants have produced.

4.3.2 SPECIFICATION

- Location - Initially west of the Western Isles in 80 m depth of water.
- Rating of power take-off - see Chapter 3.
- Moorings - Compliant moorings and fixed orientation to suit predominant sea direction (see section 4.3.4.3).
- Cross section of the device - As derived from model tests and construction considerations.
- Size of device - The water columns are 12 m wide x 18 m long. Six of these are combined to form an overall unit of 116 m length.
- Materials - Reinforced and prestressed concrete.
- Design codes - As set down in Chapter 3.
- Sea states - 12 times a year recurrent 20 m wave, period 11 sec. 100 year return extreme wave 37 m period 18 sec. (the latter corresponds to a reduced load factor).
- Extreme design condition wave taken as 15 m trough to crest, i.e. half the hundred year 30 m wave.
- Hydrostatic pressure at any depth taken as the sum of (a) and (b) below:
  a) Still water hydrostatic (max. 25 m) x 1.15 (to allow for undetermined structural loadings, e.g. longitudinal bending).
  b) Increase in hydrostatic due to wave height of 7.5 m x 1.50 (to allow for dynamic effects of device motion and wave action).
- Provision for ease of construction and maintenance - this has been a controlling requirement in the whole development of the design.

4.3.3 KEY DIMENSIONS AND STRUCTURE OF THE REFERENCE DESIGN

4.3.3.1 Cross Section of the Device

The oscillation of the water column, and the heave, pitch and surge response of the overall Device interact in a complex way. These motions have been investigated theoretically and experimentally. Satisfactory performance derives from a correct matching of device size to the most useful wave length of the incident sea. This determines the overall shape and rather massive dimensions.

The finally adopted cross-section as shown on Drawings WP/78/OWC/4 and 5, is a rectangular concrete box, divided by solid internal walls into several compartments which are utilised as ballast tanks, machine rooms and air flow ducts. The mass distribution and geometric properties of the Reference Design cross-section have been kept as close as possible to those of the earlier models and feedback has allowed the testing of models which are based on the Reference Design. Streamlining behind the device, which was thought to be a vital element of the cross-section, has proved to be of relatively little importance. The critical dimension of the cross-section are governed by performance requirements as follows:

a) Width of water column - This is a critical parameter governing the tuning of the device to wave period and the hydrodynamic damping. For peak acceptance of wave energy, the width needs to be about one tenth of the wave length for which the device is optimised.

b) Width of whole device - The overall width of the device is 35 m. This is governed by the need to get a cut water area that provides the correct buoyancy stiffness both in heave and pitch. A model was tested with the width reduced by 15% and theoretical optimum was markedly reduced).

c) Overall draught - This is governed by the need to provide a deep reflecting face to minimise the energy which passes underneath the device. At 25 m draught, which is the minimum acceptable for satisfactory performance, the Reference Design is saddled with major problems during construction.

d) Depth of front face - The function of this face is purely to keep the air cell trapped even at the trough of the wave. Hence this dimension is a function of the minimum wave height for which the device is required to function efficiently. Increasing the depth beyond the minimum required increases the wave energy lost by reflection from the face of the device.

e) Depth of the face openings, function of toe - The toe increases the radiation damping of the water column by causing the water flow to and from the cell to interact with the incident wave close to the surface of the sea (where the wave induced water particle motions are greatest). It has been found that this enhanced directional radiation damping broadens the frequency response bandwidth of the OWC. Optimising for a best combination of efficiency and bandwidth leads to a front face opening of 8 m.

f) Height of the air column - The air acts approximately incompressibly, hence the height of the air column is determined by the clearance needed to prevent water reaching the top of the column in extreme conditions.

4.3.3.2 Length of the Device

The overall length of the Reference Design was chosen by the Device Team to be 116 m (6 No. water columns at 18 m each, plus diaphragms) partly from structural, constructional and mooring considerations, but mainly from tank tests on various lengths of models. The proportions of the design are such that it is a compact and rigid structure, suitable for construction in several existing construction yards. The length is such that, in extreme conditions, it will be able to ride the waves rather than have its end columns forced under water with the attendant risk of turbine flooding and excess hydrostatic loading.

It should be noted however, that the shorter device length is less effective than a larger device in tapping the total energy resource, and the Consultants consider that the device length is likely to increase if a requirement to maximise output is imposed on the designers.

4.3.4 STRUCTURE (INCLUDING MOORINGS)

4.3.4.1 Load and Stress Constraints

Structural analysis and design is thus far based on assumed loadings. Analysis based on measured hydrostatic and hydrodynamic loadings in extreme conditions will not be possible until load measurements are taken in the tank. The present approach of assuming a 50% extreme wave response with a factor to allow for dynamic effects is considered reasonable. Limit state design philosophy has been adopted for the design. Concrete thicknesses are all determined by strength requirements - primarily by the need to resist the high hydrostatic heads which are a feature of the device. Concrete thicknesses could be reduced in a number of places if some of the internal cavities of the device were pressurised with air to reduce the inbalance of pressure on the walls. Thus far this option, which might possibly reduce the volume of concrete by 10% to 20%, has not been explored.

4.3.4.2 Float Out and Ballasting Constraints

Since the draught of the working device is 25 metres, no existing construction yards could accommodate the complete structure and the design has been constrained by the need to provide for a two stage construction technique. The first stage takes place in the dock facility and the base of the unit is then floated out at about 8 m draught. A temporary steel bulkhead is positioned at the water column opening and construction then continues on the floating device in a deep water site.

The steel bulkhead ensures that trim can be maintained as construction continues with the minimum of trouble since the floating body will in effect be a rectangular vertical sided caisson. In the working state with the water column open and full of water, the device requires a large amount of ballast positioned at the opposite side to the water column to maintain the correct inclination. Low strength concrete has been proposed for this purpose. The concrete ballast will be placed before final fitting out and the temporary bulkheads will remain in position until the units are finally sited. Provision is made for temporary water ballast to maintain trim until the bulkheads are removed.

4.3.4.3 Moorings

These present major problems. Forces are large, and full compliance is required. Devices must ideally be moored close together to utilise all the energy available, but this leads to difficulty with overlapping mooring systems and increases the problem of preventing accidental collision of adjacent devices. The Device Team have proposed a solution to meet the criteria of simplicity and reliability for widely spaced units in a prototype 100MW installation. A different, more compact solution will be required for a 2GW installation. Chapter 14 deals with the problem of mooring in detail.

4.3.5 POWER OFFTAKE

4.3.5.1 Equipment Housing

The turbines, generators and control systems are housed within the body of the device. This arrangement allows minimum freeboard, thus reducing the area exposed to extreme waves. It also assists in keeping the centre of gravity low and utilises space which would otherwise be empty. The rectifying valves and air ducts are the only components which have to be situated above the water column. The water column cover slabs and machinery space decks are formed from precast concrete plank units. The seaward face is gently sloped to reduce wave slam effects.

4.3.5.2 Power Plant - General

The firmly agreed basis of the power offtake is a separate air turbine for each working air cell, driven by a pulsating rectified air flow. Beyond this, it is still not clear which particular arrangements for power generation will prove to be most effective overall. Two approaches are referred to in this section, the first being at present favoured by NEL and the second involving a number of alternative ideas which the Consultants would like to see investigated further.

4.3.5.3 Power Plant

The NEL design is based on individual air turbine generator units working entirely independently but requiring six sets of interconnecting cables, rectiformer units and the associated automatic controls. The level of power abstraction proposed by NEL as regards the air turbines is only equivalent to 8.9 kW/m. This has been explained as being the rating for the longer duration sea conditions. Turbine speed would be less than 400 rpm. It is understood that the turbine would have a much greater capability in keeping with a maximum continuous rating for the associated generator of 1500 kW at 500 rpm. It is not immediately obvious why such a low nominal rating has been proposed for the turbines which would normally be rated in keeping with the continuous duty specified for the generator. The maximum continuous electrical rating of the NEL device would correspond to 77 kW/m which is appropriate to the proposed Hebridean location.

If the NEL device is compared with others in the present review and due account is taken of the device length and performance characteristic, it would seem appropriate to provide air turbines having an hourly mean rating of about 100 kW but stressed to withstand momentary torque inputs about twice this figure. The medium speed radial inflow air turbine proposed by NEL is probably the most suitable prime mover. The Consultants however, have not made any parallel study of turbine, type, performance and dimensions.

A characteristic of secondary power abstraction using very low presssure air flows is that relative motion of the device and the external waves results in a pulsating air flow which is made unidirectional by the automatic rectification valves. On considering the motion of the liquid piston in air cell it is reasonable to expect, as a very rough approximation, that the rectified air flow to the turbine will persist for only about 70 per cent of the wave period. There will be virtually no flow during the remaining 30 per cent of the time. This will result in a pulsating power input to the alternator shaft. The speed must not be allowed to fall too much during the quiescent period, and this implies sufficient ωr^2 in the rotating masses. As mentioned in the NEL Second Interim Report, and as advised by TAG 6, an inertia constant of 15-16 seconds is necessary.

4.3.5.4 Valves and Ducts

The NEL device includes carefully designed air flow rectification valves. These are of the multiple light-weight louvre type arranged so that when moved to the open position they form part of flow directing guide vanes. Device performance relies on efficient use of the air flows induced by the water column movements. The NEL have now included hydraulically operated servo mechanisms to open or close the louvre valves without delay and in keeping with air flow direction at the cell exit. This is a good feature, ensuring that all air movement is directed correctly and that when valves are closed they form an effective pressure seal. Reduction of leakage losses is important. It is suggested that the rectifier valve mechanism should be initiated by changes in the direction of motion of the water column surface and not by air pressure differential.

Large air flows through the connecting air ducts can result in pneumatic losses. These can be minimised by careful design of the air passages as regards sectional area and changes in flow direction. The latter should be assisted by suitable guide vanes. Aero-dynamic model testing should be included when preparing a prototype design. An essential feature already included, is an arrangement for intercepting and removing quantities of sea water carried over in the air stream during the delivery stroke.

4.3.5.5. Power Generation

The NEL design proposes six separate turbine alternator sets. This entails the provision of six medium voltage flexible submarine cables and six sets of rectiformer units and their controls. The arrangement would be relatively expensive, would reduce the overall reliability owing to the greater risk associated with the cables, and would incur reduced efficiency due to the continuous cylic loading of the individual circuits. It is appreciated that the NEL proposal is associated with sea-bed mounted electrical equipment. In a subsequent section of this Report - Chapter 11 "Generic Topics; Generation, Collection and Transmission" - we have explained our objections to the use of any electrical equipment on the seabed other than straightforward submarine cables, which are unavoidable.

A suggested alternative approach to a power offtake system which the Consultants believe should be further considered, is shown schematically in Fig. 4.4. The air turbines are individual to the six cells, similar to those proposed by NEL but of greater output to suit a more appropriate and representative power conversion level. In place of the alternators, each turbine drives a unidirectional high pressure oil pump. Each set would be provided with a flywheel to compensate for the loss of the rotor momentum. The aim is to achieve a compact and relatively efficient means of summating the six cell outputs. The proposed oil hydraulic system would operate at 1500 p.s.i. (10.34 MNm^{-2}).

Each pump provides a flow of high pressure oil corresponding to the cyclic power input from the air turbine. The six flows are collected in a common pressure main, the output being a relatively steady discharge reflecting input diversity. Further smoothing is achieved by a suitable arrangement of air/hydraulic accumulator, the rise and fall of fluid level being used to control the rate at which the high pressure oil is used by a single power turbine. By-pass filtration and series cooling of the return oil has been included.

The oil hydraulic turbine would operate in the range 1000-1500 rpm and would drive a three phase alternator generating at a suggested 3.3 kV. Conventional winding insulation should then be thoroughly reliable. The alternator would have a rating of between 4.0 and 4.5 MW. The output from the device would be stepped up to 22 kV in an appropriately designed transformer mounted on the device and then transmitted via flexible submarine cable to the nearby platform-mounted diode converter equipment. There would thus be only one set of electrical equipment per device with the attendant advantages of reduced cost, greater reliability and improved efficiency due to the more effective use of material and the steady loadings achieved.

It appears that whether the separate power outputs of each cell are combined electrically (as NEL) or hydraulically (the alternative suggestion) significant penalties are incurred. The alternative possibilities of combining outputs mechanically on a single shaft or of passing the whole air flow through a single turbine both require investigating.

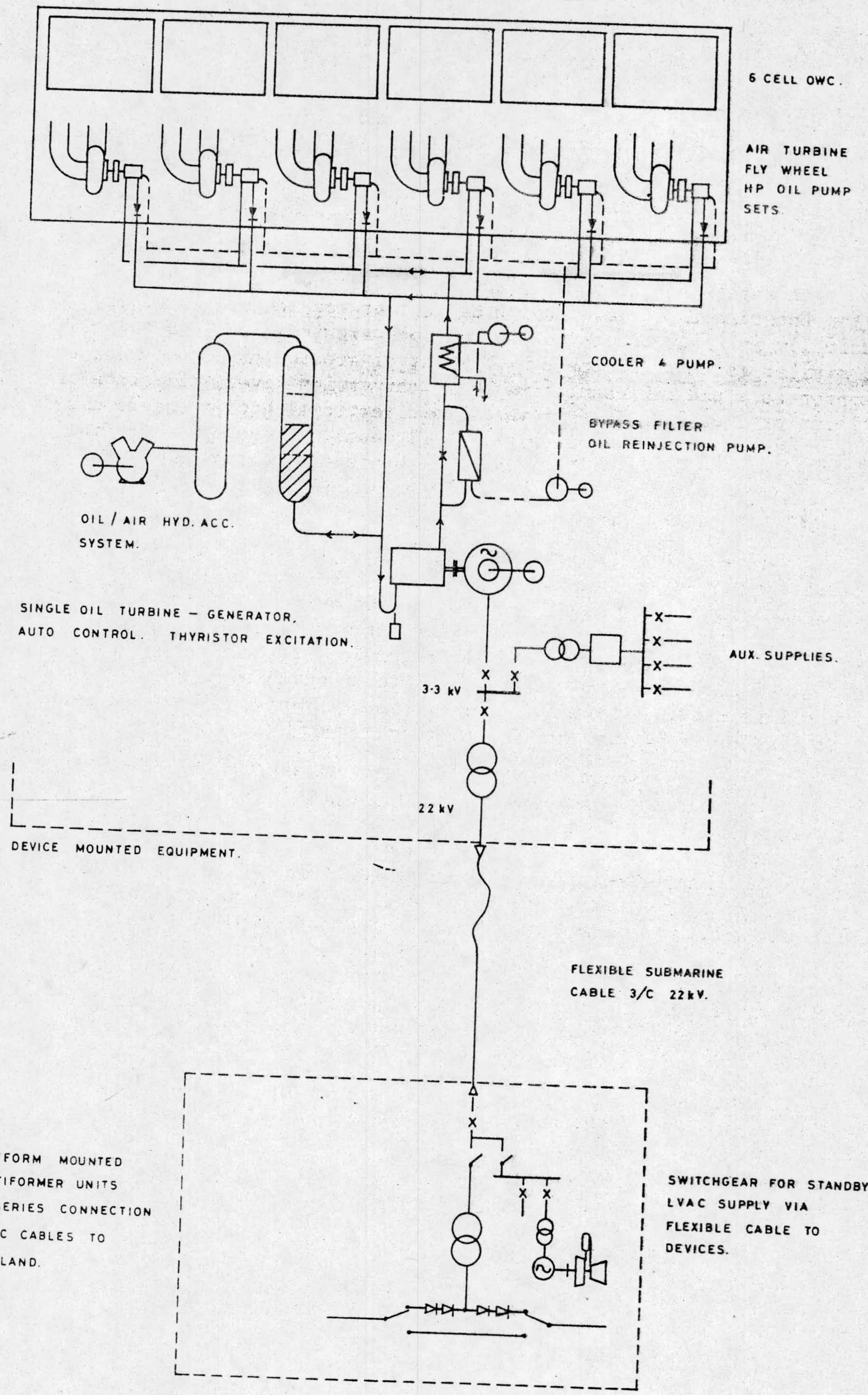
4.4 CONSTRUCTION AND INSTALLATION

4.4.1 SEQUENCE OF CONSTRUCTION (Outlined in Drg.No. WP/78/OWC/6)

The first stage of construction will take place in a large dry dock of the type used for constructing gravity platforms for the N.Sea. Depending on the size of the dock from one to perhaps three of the devices might be constructed simultaneously.

4.4.1.1 Dry Dock Construction - Stage 1

a) Prepare dock floor (area per device 116 m x 35 m) position shuttering and reinforcement and pour base slab and starter walls.



PROPOSED POWER OFF-TAKE SYSTEM USING HYDRAULIC COMBINATION OF CELL POWER OUTPUTS WITH SMOOTHING AND SINGLE TURBINE/ALTERNATOR FEEDING PLATFORM DIODE RECTIFIER UNIT.

NEL – OWC DEVICE

FIGURE 4·4

b) Shutter, reinforce and place concrete to a height of about three metres on all full-height walls.

c) Erect slipform shuttering on these walls and slip to 15 metres high. Erect conventional shuttering on all other walls and pour concrete to underside of intermediate slab.

d) Erect shuttering and pour intermediate slab.

e) Remove construction equipment from dock, ballast down units to ensure no premature float-off, flood dock. De-ballast and float out units in sequence (draught 8.5 m under keel clearance 1.0 m minimum). Tow to sheltered deep water construction site (minimum depth 25 m).

4.4.1.2 Deep Water Construction - Stage II

a) Moor the unit and establish communications and access with shore-based facilities.

b) Erect temporary steel bulkheads to water column face, extending these as construction progresses. Continue slip forming of main walls, and carry out in-situ pours for intermediate walls, slabs, etc. to full height (approximately 29 m).

c) Place precast nose units, stress together and grout.

d) Place precast roof beams and planks and concrete in position.

e) Place concrete ballast and trimming water ballast.

f) Install machinery and equipment and place precast plank deck over.

g) Construct "Conning Tower", valves, ducts, access and escape hatches and finishing screeds to roofs and decks.

h) Add trimming water ballast and tow-out to operating site, remove steel bulkheads and finally trim with water ballast on site after location and mooring.

Floating cranes with capacity up to 100T will be required during Stage II.

4.4.2 DEVICE SITING

The Device Team recommend a water depth of 80 m for the siting of the device. This is based on the requirements of (a) their proposed mooring system, where additional depth will help to provide extra compliance, (b) the anchoring system where the Device Team consider that a sand bottom is desirable and unlikely to be found at less than 80 m. This increased depth (compared with the depth of 60 m previously favoured) incurs increased costs for transmission and for the bottom standing collector/transformer platforms. Drg.No. WP 78/OWC/2 illustrates the proposal. The Consultants believe that an alternative mooring and anchoring system may well result in a return to rather shallower water, with cost saving.

4.4.3 INSTALLATION OF REFERENCE DESIGN - DEVICE TEAM PROPOSAL

The mooring system proposed by the Device Team spaces the units 420 m apart. This corresponds approximately to 1 unit to 4½ unit lengths which reduces significantly the potential total power absorption from any site. An array (two or more lines) of devices is considered to be economically unattractive.

4.4.3.1 Site Preparation

The proposed site will require to be comprehensively surveyed to check on the suitability for anchorage. Marker buoys would then be placed with the possible addition of acoustic transponders to assist in the accurate positioning and relocation of anchorage points.

4.4.3.2 Anchor Installation

A two-part anchor system is proposed with a drag anchor to resist horizontal forces and a concrete deadweight for the vertical forces. The drag anchors will be laid in the approximate pre-determined position and 'dug-in'. The dead weights will then be lowered into position and divers will lock these on to the anchor chains.

4.4.3.3 Mooring Line Installation

The two lower sections of the mooring lines will be attached to the deadweight anchors and the top end of the nylon rope supported by a buoy.

4.4.3.4 Mooring Hook-Up

All the mooring points on the device will have been fitted with a top steel section of the mooring line, the free end being made fast on deck. The device will then be towed into position and barge mounted winches used to connect the wire rope from the device to the buoy-mounted end of the nylon rope.

The water will be gradually admitted to behind the temporary steel bulkheads until the pressures are equalised and the bulkheads can be removed and returned to the construction site for re-use. The water ballast will then be adjusted to put the other device into its operating aspect.

4.4.3.5 Electrical Connections

The Device Team have proposed a novel system for leading the power and control cables from the device to the seabed mounted transformer rectifier module. The flexible cables from all of the generators in a unit would be gathered into a single GRP sheath about half a metre in diameter which would be formed into a several metre diameter helix lightly sprung in the closed position. This would be positioned on the seabed module with a line to a buoy on the surface. When the OWC has been moored in position the line would be taken from the buoy and winched into the device extending the helix until the upper end could be anchored inside the device and the electrical connections made. The helical form would accommodate the maximum excursions of the device.

4.5 OPERATION AND MAINTENANCE

This paragraph simply lists the factors considered by the Team in developing their design. Space does not permit reproduction of their ideas in detail, but it may be noted that they are practical and fairly comprehensive:

- Structures are unmanned in operation.
- Provision is made for routine on-station maintenance.
- Provision is also made for on-station replacement of major components by module replacement using floating cranes.
- Major repairs or refurbishment require tow to sheltered water site.
- Mooring forces and performance are monitored to allow early detection of deterioration and immediate replacement.
- Drag anchors and chain to be recovered every five years for inspection.

4.6 ALTERNATIVE REFERENCE DESIGN

The Device Team have engaged the services of Scott Lithgows (British Shipbuilders) to carry out a pilot study into the building of OWC units in steel. This is completely practical, and the resulting structure is similar in size, similar in performance, but of slightly different (and preferred) shape. Each structure would contain about 12,000 tons of steel.

The initial estimate of the basic structure cost is such that, even allowing for large savings from possible refinement of designs, etc. the steel version cannot compete on cost with the concrete one. This is commented on further in the costing annexe.

4.7 CRITICAL ASSESSMENT OF TECHNICAL FEASIBILITY

4.7.1 GENERAL

This device could be designed and built in the fairly near future, with a high level of confidence that the prototype would work. The structure is monolithic and the power offtake reasonably conventional. This can not at present be said of any other device in the programme. The structure is however, massive. Its appearance has been described as improbable, and in spite of the irrefutable logic behind its development, it does not have the right "feel". It is inflexible in its requirement for special construction sites, and for deep water bases for off-station maintenance. The weight of the Reference Design structure has changed little over the last year, and it seems certain that this particular configuration of the OWC is now close to its optimum form. The case must be strong for following some of the other water column options referred to in 4.2 to see if they lead to more economic alternatives.

4.7.2 DEVICE CONCEPT

This is technically sound but has led to a structure with a draught which locks it into a very special system of construction facilities. Reduced option of construction location always increases the price. The concept of the OWC as such is very attractive, for reasons already indicated, and the absence of any sort of articulation removes many of the problem areas which bedevil some other devices.

4.7.3 THE STRUCTURE

With the reservations already stated in respect of size, the structure is otherwise straightforward to make. It has no serious stress hot spots, with the possible exception of the mooring attachment points. Difficult concrete pours have been largely eliminated, and slip forming techniques can be used. The problem of corrosion of exposed steel embedded in concrete is shared with most devices, and the judicious use of other materials should reduce this problem to manageable proportions (see also 4.7.4. following).

The major question must be whether the total volume of structural concrete can be reduced. The option of providing some internal pressurising to reduce differential hydrostatic head remains to be explored, as does the option of using curved shell elements to resist pressure. The total cost saving on the structure from these two sources might be up to 15% but this is simply conjectural at this stage.

4.7.4 WATER INGRESS

Entry of water through the valves has been a cause of concern to the Team. The Device has been shortened to reduce the submergence of the ends due to longitudinal pitching, and the working chamber has been shaped to throw any water slopping up the sides away from the duct inlets.

The Device Team have conducted a series of tank tests to investigate the problem and the results have been encouraging. As long as the chamber is dynamically damped the only water to hit the roof comes from isolated points at the edges of the cell. Prevention of water entry will depend on maintaining damping, and/or on providing emergency shut-off at the inlet valves. The prototype must incorporate fail-safe systems to ensure this, but there seems to be no reason to doubt that it can be done.

Ingress of water from the rear also gives cause for concern. As shown on the Reference Design Drawings the ducts from the outside atmosphere into or out of the valves face horizontally to the rear of the device. A slight downward slope of the bottom of these ducts is designed to help water to drain out but more development work will be required to ensure that water will not enter the Device from this area.

Overall it is assessed that water ingress is a problem area, but that solutions are likely to be found.

4.7.5 MOORINGS

See Chapter 14. Moorings for this Device will be easier than for the rafts, but harder than for most other devices. The shape, alignment and motion of the Device all combine to make the problem particularly severe for this Device both in terms of force and excursion. The short length and close spacing between devices, which is necessary for an effective power station, create problems in preventing collisions, and these are magnified by the need for highly compliant moorings. The mooring problem is soluble at a price, but further work might well show that the price is too high. In particular, mooring requirements are tending to govern the spacing between devices, and hence the total resource available. In many ways this may be seen to be unacceptable.

4.7.6 POWER OFF-TAKE

This is a positive asset of the Device in the sense that no one questions that it can be made to work, and it is reasonably within present technology. There is no firm evidence yet on which to base predictions of efficiency of the turbine in the unusual condition of pulsating air flow.

4.7.7 FUTURE EXPERIMENTAL PROGRAMME

In view of the comment made in 4.7.1 it will be seen that the primary need is to give priority to investigating alternative configurations of OWC. All the evidence, in wave power as in other areas, is that the major breakthroughs on cost come through new layouts and new concepts. It is now essential that the various avenues for investigation of OWCs are clearly identified and researched in a single co-ordinated programme.

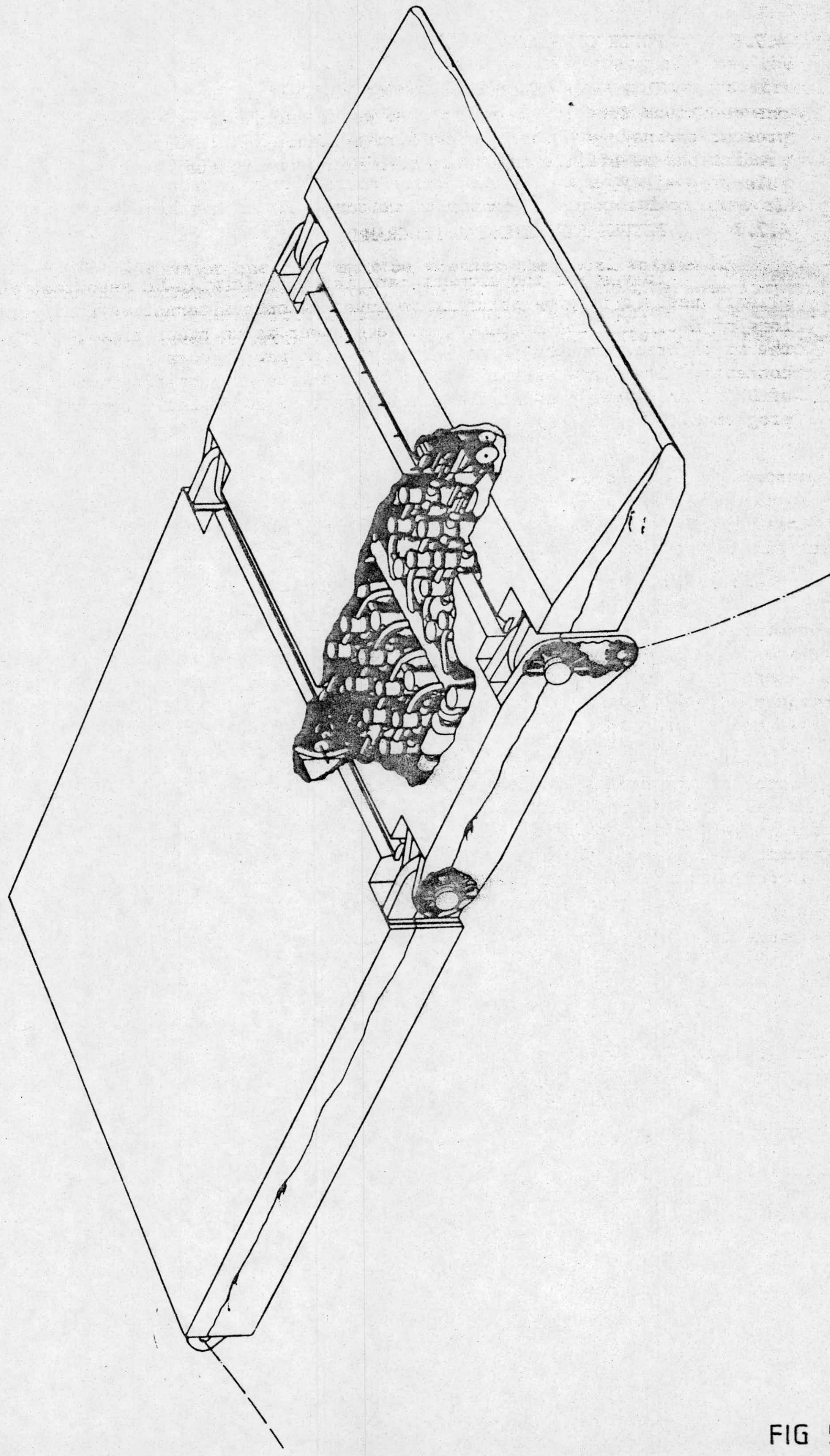


FIG 5·1

CHAPTER 5 - COCKERELL RAFT (DEVICE TEAM - WAVEPOWER LTD.)

5.1 GENERAL DESCRIPTION OF THE DEVICE

5.1.1 CONCEPT AND LAYOUT

The basic concept remains the same as that described in the Consultants' 1977 Report. Three rectangular, shallow-draught, flat-bottomed pontoons are hinged together along two parallel lines to articulate in one plane as the waves pass under them. The relative angular motion of the pontoons are used to drive a power take-off system. The rafts (three connected pontoons are termed one raft) are moored to the sea bed in discrete lines abreast. Power-collection cables from the single rafts lead along the sea bed to a converter station on a fixed platform from which a single high voltage cable will run to shore.

The pontoons are of constant depth except for the tapered nose of the front pontoon and local tapering to the underside edge of the central pontoon. The depth of the front and rear sections is proposed by the Device Team as 7 m and the depth of the centre section is increased to 8.6 m to accommodate plant. The front and rear pontoons are stiffened internally by a rectangular grid of diaphragms at approximately 5 m centres. The structural material has not been finally determined. The present thinking of the Device Team is that the first and third pontoons will be of reinforced or prestressed concrete and the centre pontoon of structural steelwork. The reasons for this are discussed in paragraph 5.3.3.1.

In the Reference Design, each hinge line has two self-lubricating spherical bearings each on a hollow forged trunion shaft, some 2 m in diameter. The bearings are housed in substantial steel castings which are themselves welded to a further steel fabrication which is spliced to the pontoon structure. By providing only two bearings to each hinge line the serious difficulty of accurately aligning the axis of more than two bearings is avoided. The use of the spherical type of bearing permits deformation of the pontoons along the hinge line without undue wear (or even seizure) of the bearings. Their location, at the extreme ends of the hinge line leaves the interior part of the hinge line entirely clear for provision of power take-off equipment.

It is proposed that the rafts be moored in 50-60 m depth, this being the Device Team's intuitive estimate of the least depth required to keep outside the area of breaking waves.

5.1.2 PRINCIPLES OF OPERATION

5.1.2.1 General

The relative angular motions of the pontoons about the hinges are used to pressurise water in a hydraulic main which leads to a turbine driving an alternator. Several different mechanical systems for producing pressurised fluid have been considered by the Device Team and these are briefly summarised in paragraph 5.3.3.11. At the time of writing, the most promising system is that adopted for the Reference Design. Toothed racks are fixed in the first and third pontoons. Gear pinions supported on cantilevering arms from the centre pontoons engage in these racks. After further gearing up, eccentric arms drive pistons in double-acting cylinders, Sea water is drawn into the cylinders through non-return valves and pumped into the pressure main.

5.1.2.2 Geometric Parameters Related to Device Performance

The performance is dependent on the following geometric parameters:-

a) overall plan dimensions of device,

b) lengths of individual pontoons,

c) spacing between devices in a string,

d) the self weight of the device (but this is found not to have a critical effect as long as it is above a minimum value required to prevent resonating wave slam),

e) the depth of water required to be sufficient to prevent waves breaking,

f) orientation - there will be a directional effect in oblique waves,

g) mooring attachment points, mooring configuration and stiffness.

5.2 STATE OF DEVELOPMENT

5.2.1 GENERAL

This device has been tank tested to a point at which the Team have a reasonably good idea of the power potential and some very preliminary idea of the forces to which key components might be subjected. Power offtake has not yet passed beyond the desk study stage. Because the pontoon structure has to house the power offtake, carry the hinges and support the mooring forces, none of which are yet defined, no well-founded structural analysis and design has yet been possible. Thus, work related to the costing of the Device is still at a very early stage. The following paragraphs note some of the testing work that has been done and identify the state of knowledge in some key areas.

5.2.2 TANK TESTING

5.2.2.1 Performance Testing

A comprehensive series of wave-tank tests has been carried out on behalf of the Device Team by British Hovercraft Corporation Ltd.

The quantitive testing was undertaken with both monochromatic and random waves in two medium width tanks of breadths 1.55 and 1.875 x the model width, which thus tended to simulate devices at scaled spacing of 77.5 m and 94 m, head on to the waves. Variables investigated included:

a) The number of hinges in a raft - Tests covered one, two and three hinges. The original concept was for three hinge lines per raft, but this series of tests showed that the power produced at the third hinge was very small compared with that at the other two.

b) The amount and type of damping - Two types were employed, torque held at a constant value and torque proportional to angular velocity. These would correspond to different possible types of power take-off systems.

c) The bow size and shape - In order to reduce the extreme loading in heavy seas by encouraging large waves to pass over the structure an extended bow was fitted to the front pontoon for some tests. Thus far however, little has been done to study the effect of profiling the front pontoon and the shape used in the Reference Design is largely intuitive.

d) Flume width - The flume width was reduced to only 4 cms more than the width of the model to compare the power output with the normal test cases with the flume at 3.66 m wide.

For tests using regular waves the wave height and wave length were varied. Routine measurements taken during testing were:-

Power output at each operative hinge.
Efficiency (defined as power produced across the hinges divided by power contained in the waves)
Torque.
Angular rotation.

From these tests conclusions have been drawn on the basic behaviour of the device. Some important variables could not be checked in the test facility, the most notable one being directionality.

5.2.2.2 Structure and Mooring Forces

A series of two dimensional tank tests was carried out to measure forces on the hinges and mooring lines. Realising that these would depend on both mooring line characteristics, damping characteristics and perhaps the point of attachment of the mooring line at the hinges, a fairly wide combination of conditions were tested. These included:-

a) hinges undamped,

b) hinges damped (by double acting pumps pumping air with restrictors across their inlets and outlets),

c) restrictors removed and water pumped to a head of 0.4 m,

d) hinges undamped and vane attached to the underside of forward pontoon,

e) as configuration a) but with highly compliant mooring lines,

f) model ballasted to represent 40% increase in displacement.
The tests were performed in monochromatic waves of three different wave lengths and varying heights, and also in random waves. Peak values of forces were measured, and in the case of hinge forces, both horizontal and vertical components. Angular displacements at the hinges were also measured.

5.2.3 SEA TRIALS

A 1/10 scale model was moored in the Solent in March 1978. This was to meet the dual objectives of gaining sea experience and gaining further data in real sea conditions. In particular, mooring problems were of major interest. The model is instrumented with load cells on all six mooring attachment points and also at the sea bed end of the forward mooring line. An accelerometer is fitted on the front pontoon to record slamming effects and the hinges are strain-gauged to record horizontal, vertical and axial components. The power generation system is also fully instrumented, and has been designed so that different types of damping can be simulated. Windspeed, wind direction, tidal current speed, water depth and wave data (via an NMI wave-rider buoy) are all being simultaneously monitored. Significant effort has gone into developing anchors for the model, but this experience will not necessarily be relevant to full scale devices.

5.2.4 THEORETICAL WORK AND INTERPRETATION OF RESULTS

The Team has been well supported in their theoretical understanding by analytical work carried out by CEGB, Marchwood, and this work has been used to guide the experimental programme. Many features, however, can only be investigated and developed by tank testing.

The two dimensional tank tests have indicated firstly that the Device has a good hydrodynamic response over a reasonably wide band of wave periods which, for a 100 m long raft, peaks at about 6.25 seconds zero crossing period, and secondly, that it is not necessary to accurately tune the raft by accurate ballasting or by varying the pontoon lengths or shape.

Information on hinge and mooring forces is still very uncertain. Forces recorded show a wide scatter depending on the test conditions applicable (ref. para 5.2.2.2). The interpretation of the test results is made difficult by the interdependence of hinge forces, damping, and mooring compliance, and a satisfactory correlated set of data covering the areas of interest is not yet available.

The Device Team has idealised the several very different relationships obtained between hinge forces and wave heights and has (optimistically) assumed that with further optimisation of mooring line elasticity and hinge damping, the forces can in future be made much lower than recorded in tests so far. Scaling up the idealised hinge forces to prototype size is done by multiplying by the cube of the linear geometric scale factor, which, for a 100 m long prototype is 580,000. It is therefore very important that the correct peak model force is used. It is apparent that tests to date have not simulated the sea states represented on the Station India scatter diagram by waves of up to 28.4 m high with an energy period of 11.75 secs.

The damping characteristics will be dependent on the power take-off system adopted. It is clear that the optimisation of the hinge forces must proceed in conjunction with the design of power take-off systems.

5.2.5 POWER TAKE-OFF SYSTEMS - DESK STUDIES

The Device Team have considered desk designs of six possible power take-off mechanical systems in some detail, i.e.

a) plunger pump,
b) plunger pump with gear reduction,
c) double-acting vane pump,
d) single-acting vane pump (two types)
e) bellows pump.

Engineering problems have been identified in all systems. The system currently favoured by the Team is the plunger pump with gear reduction, which has the following advantages.

a) small angular excursions of the hinges can be geared to provide full strokes of the pump,

b) some tolerance in the linear dimensions of the adjacent pontoons, including elastic deflections, can be accommodated by the gear teeth and by adjustable rack mountings.

The system however involves a lot of components and will never be cheap. Enough work has been done on some of the other systems to identify major difficulties.

Vane pumps have major sealing problems, related to the relative structural and tolerance deflections of the adjacent rafts. Particular difficulties are:

a) sealing the gap between the end of the vane and the wall of the pressure chamber. These components are on adjoining pontoons between which there will be relative structural movement,

b) the alignment of the centre of rotation of the vanes and the centre of curvature of the pressure chamber with the axis of the structural hinges of the raft. When this is achieved statically it will be upset ~~by~~ structural deflections of the rafts in motion, giving rise to fouling of the vane with the chamber wall,

c) the angular travel of a vane must have a definite limit to avoid damage, but it must adopt the relative angular travel of the pontoons. The angular travels in the majority of waves will be small and hence a high pressure would be required to generate the required power. This would conflict with the sealing difficulties mentioned in (b).

The bellows pump has the merit of having no alignment problem. However, it suffers from having a limit imposed on its range of movement and from having a low travel in normal operation, thereby requiring an unacceptably high pumping pressure to produce the necessary power.

In every system the proposed pumping medium is sea water since an oil system would require a heat exchanger with consequent loss of efficiency.

Thus far there can be no satisfaction that a good and economical power take-off system has been identified, in spite of considerable thought over the last two years.

5.2.6 ALTERNATIVE CONFIGURATIONS

A series of tests was carried out on a variant concept consisting of articulated pontoons submerged beneath the surface, flotation being provided by buoyancy tanks fixed above the pontoons by a lattice structure. This investigation was prompted by an idea that performance might improve with a submerged raft absorbing more energy than a buoyant one by virtue of its being enveloped by the water. Results of these tests were disappointing. Peak efficiency was only 48% compared with 78% for the floating pontoon, and the band width was much reduced. Angles of the underwater pontoons were varied by adjusting the attachments to the supporting structure but no further improvement in efficiency was revealed. In view of these findings and the quantity of the outstanding work still to be conducted on the floating raft, the Device Team has decided not to continue investigations into the submerged plate concept at present.

Other configurations which might yet be investigated include a two pontoon system, and a concept involving a number of front pontoons side by side, hinged onto a common, much wider, back raft.

5.3 THE REFERENCE DESIGN

5.3.1 INTRODUCTION

The Device Team are continuing their work on the basis of a raft plan size of 100 m x 50 m although they recognise the advantages of going smaller. Smaller rafts would incur forces reduced by the square of the reduction factor, and the reduction which would follow in the cost of large manufactured components of these were reduced to more conventional size could more than offset the diminished power output. The final choice of device size will depend on an economic analysis and this can not be done until some value criteria are established for the output power.

5.3.2 GENERAL SPECIFICATION

Location	- West of the Western Isles
Water Depth	- 50 - 60 m depending on nature of seabed for anchoring
Rating of Power Take-off	- 1.75 MW per device.
Moorings	- Compliant horizontally, but relatively less compliant transversely. Maximum travel from mean position ± 5 m.
Arrangement of Devices	- In straight lines abreast parallel to prevailing wave front, at 100 m c/c with intermittent gaps of 150 m.
Dimensions	- Front pontoon 25 m x 50 m x 7 m Centre pontoon 25 m x 50 m x 8.6 m Rear pontoon 50 m x 50 m x 7 m
Material of Structure	- Front pontoon Reinforced and prestressed concrete Centre pontoon Structural steelwork Grade 43A Rear pontoon Reinforced and prestressed concrete
Characteristic strength of concrete	- 40 N/mm^2 at 28 days
Structural steelwork	- Mild steel - all welded construction
Hinges	- 2 hinge lines, each containing 2 hinges with self-lubricating spherical bearings. Radius of bearing surfaces 1.25 m Length of bearing 1 m

Hinges (contd).	Diameter of trunnion shaft 2 m Wall thickness of trunnion shaft 100 m m.
Peak hinge design forces	- 1900 tonnes (horizontal) per bearing 950 " (vertical) " "
Peak angular travel	- at hinges $\pm 25^{\circ}$
Peak design mooring force	- 1500 tonnes per rope
Estimated lightweights	- Front pontoon 3500 tonnes Centre pontoon 700 tonnes Rear pontoon 7000 tonnes
Estimated machinery weight	- 1000 tonnes
Ballast	- 4300 tonnes
Displacement at Design Draught	- approx. 16,500 tonnes
Power take-off equipment (present proposals)	- Radius of rack 6000 mm Width of rack 350 mm Gear ratio 32:1 Pump pressure 5.5 bar No.of pumps per raft 16 Pump diameter 750 mm Pump stroke 1500 mm

5.3.3 DEVELOPMENT OF THE REFERENCE DESIGN

5.3.3.1 Choice of Materials

The Device Team has an open mind on which constructional material will be used. Reinforced concrete has been found to be reliable material for sea structures, and many ships, floating docks, barges, pontoons, as well as fixed structures have been constructed in R.C. and in prestressed concrete. Economically there is probably little difference in first costs. Moreover, maintenance costs of the hull through its lifetime are virtually eliminated with concrete. Traditional shipbuilding uses steelwork for the reasons of advantageous operating costs (more engine power is expended in propelling the hull of a concrete vessel) and the intricate internal fitting out of a vessel does not lend itself to the thicker concrete sections.

In the case of the Cockerell Raft the end pontoons are of simple form and self propulsion is not a factor. It would therefore seem that concrete is the better choice of material for these. The centre pontoon however is a floating engine room and steel construction has many advantages. In particular it affords more internal space, it is better able to accommodate access holes, and provides a convenient strong machine bed. The Reference Design therefore consists of concrete outer pontoons and a steel central pontoon.

5.3.3.2 Structural Design

The following are the main considerations which have been taken into account in the structural design.

a) overall bending, shear and torsion in the pontoons,

b) local panel bending and shear from normal forces,

c) locally applied forces (particularly at the hinges) and their dispersion into the structure,

d) effects of repeated loading,

e) constructional procedures,

f) buoyancy.

Thus far no detailed design has been carried out to determine the constraints on access panels in the main structural deck.

5.3.3.3 Overall Bending

A preliminary assessment of the overall bending strength of the rear pontoon has been carried out, based on Lloyds' Rules for the design of tanker sections. This exercise considered the third pontoon (being the longest) as an isolated pontoon of 65m effective length, the increase over its real length (50m) being the estimated allowance for the effect of the torque at the hinge. Thereafter the rear pontoon was considered as an independent structure. Transverse moments and shears produced by the internal balancing of forces from the hinges and the power take-off have not yet been designed.

5.3.3.4 Local Panel Bending

Local panel bending is a function of the diaphragm spacing and normal water pressure. Hydrostatic panel pressures are easily calculable but slamming forces, mainly of the front pontoon have not yet been accounted for in detail. They are related to the degree of hinge damping and will be reduced if the Team's efforts in profiling the front pontoon are successful in reducing response in extreme waves.

5.3.3.5 Hinges, and Mooring Forces

The importance of accurate assessment of the hinge and mooring forces has already been stated. The magnitude of these forces means that their distribution into the structure requires very substantial members. The Device Team's present assessment of the extreme hinge line load (horizontal component) is 3750 tons. Reaction to this force in the concrete structures has been provided in the Reference Design by prestressing tendons in the webs in line with the bearings. The tendons are anchored at regular intervals along the length of the web since the centrifugal force of the pontoon varies linearly with distance of elements from the hinge. The distribution of this force from the main webs into the raft is by in-plane shear in the top and bottom slabs, for which significant transverse reinforcement will be required.

The Device Team has not rigidly specified the exact position and structural detail of mooring attachment points. Since the forces are of an order comparable with the hinge forces and could be applied at one attachment point, it appears that the most feasible way of transferring these to the structure is by a similar system of steel casting, steelwork fabrication and splicing. The Device Team's most recent indications to the Consultants were that the hinges themselves were the favoured attachment points, and the Consultants have therefore indicated a further casting for the mooring attachment housed in the same fabrication as the hinge trunnion bearing (see Drawing No. WP78/RAFT/4). The mechanical connection of the mooring line to this trunnion has not been designed. It is not possible to ascertain from the test results so far obtained whether, with this mooring attachment point, the reaction across the splice face would increase, or indeed whether the present test results would be applicable.

5.3.3.6 Fatigue in Concrete Reinforcement

A method for assessing the number of loading cycles on the third pontoon due to wave bending based on the Station India scatter diagram has been indicated by the Device Team. It has been applied to a reinforced concrete design and the Miner's sum calculated for the reinforcement. In principle, on the basis of current knowledge of fatigue in offshore reinforced concrete structures this would appear to be a sound approach. No fatigue check has been carried out for the specimen structural steel design. It is true that Lloyd's Rules for ship design have a fatigue consideration built into them, but this is considering 25%, not 100%, of the vessel's life at sea to be spent in the N.W. Atlantic, and only half that in the loaded condition.

Fatigue could well prove to govern the design, but little can be done to check this until a structure loading spectrum is defined.

5.3.3.7 Concrete Thicknesses

The Device Team's design of the rear pontoon in reinforced concrete shows slabs generally 300 mm thick. Their calculations require reinforcement in the slabs almost above the practical construction limit. Since there are loading effects not yet allowed for, it seems likely that the slabs will in future need to be thickened or the raft deepened, or both.

At present no criterion has been identified for the web thicknesses except for those carrying the hinge reactions, which will be of the order of 2 m thick at the splices to accommodate the tendons.

It is appreciated that the weight of the two prestressed diaphragms is considerable and will produce a disparity in the draughts of the steel and concrete sections. There is however scope for reducing the thickness along the length. A reduction in the width of the spherical bearing and hence in the width of fabrication base plate would give the most immediate reduction in web thickness but this is dependent on a detailed design of bearing which has not been done.

5.3.3.8 Constructional Steelwork Design

Early in May 1978, a contract was made by the Device Team and British Shipbuilders (Swan Hunter Shipbuilders Ltd.) for the latter to carry out a design of a complete raft structure in steel, together with a cost estimate and corrosion protection recommendations. This design showed simple repeated welded box sections with plate or framed diaphagms and bulb stiffeners, very suitable for series production. The bottom plates are designed for a normal hydrostatic head of 7 metres for the aft and centre pontoons and 9 m for the forward pontoon. The top plates are designed for 1.75 m head for the aft pontoon, 2.45 m head for the centre pontoon and 5 m head for the forward pontoon. The overall bending is stated to be according to Lloyd's Rules for tanker sections but it is not stated on what spans it is based. The design specifically excludes consideration of the hinge and primary mooring forces and warns that the open arrangement of the centre pontoon structure to accommodate machinery is subject to a detailed design check on torsional behaviour.

The present Reference Design utilises the Swan-Hunter design for the centre pontoon, adding bearings and bearing castings as described in 5.3.3.9 Consideration of the radical departure of the pontoon from the simple design concept given to Swan Hunter suggests that their design is now unrealistic.

5.3.3.9 Hinge Assemblies

The integrity of the device is dependent on the design of these components but it is not possible at the present stage to define with confidence the parameters of the hinge design, as discussed in paragraph 5.3.3.5. The Team's present proposal is for a peak horizontal hinge line force of 3,800 tonnes (1900 tonnes per bearing) and a peak vertical hinge line force of 1900 tonnes (950 tonnes per bearing). These are perpendicular to the line of the hinges. It is not known whether these occur simultaneously but for the purposes of the present Reference Design it will be assumed they do.

Spherical bearings have been approximately designed for these forces allowing for wear. A diameter of the order of 2.5 m and a length of the order of 1 m are required. The bearings fit over a forged hollow shaft. To spread the loads into the fabrication, massive steel castings house the bearings and the outer ends of the trunnion shafts. The castings are welded into steel fabrications which splice onto the raft. The Device Team favoured a tapered trunnion with the bearing at one end in the belief that this would make extraction of the trunnion and replacement of the bearings easier, but it has the disadvantage of magnifying the hinge force at the adjacent trunnion bearings. The Reference Design therefore shows the bearing central between the two fixed trunnion bearing assemblies. The latter share the hinge forces equally.

Additional loading will occur on the bearings and fabrications from lateral wave components. Magnitudes of these will be determined in future three-dimensional tests, and no calculations have been made for them at present. The bearings, castings and fabrications are massive components at the very top end of current manufacturing capacity. They will be very expensive.

5.3.3.10 Moorings

These are shown on Drawing No. (WP78/RAFT/3) and reference should also be made to Chapter 14. The Team are proposing a linked system of lines by which devices are held in position with respect to the bed and to one another. It seems most unlikely that the proposed system will be retained by the Team as their preferred option, since it has too many snags.

5.3.3.11 Power Take-off General

It has been necessary to carry out the design specification of the turbines, generator, transformer, and associated equipment at a late stage in the report programme, and this equipment is not at present shown on the Reference Drawing (August 1978).

The Reference Design uses four pairs of double acting piston pumps for each hinge line so that the centre pontoon contains sixteen pump sets, feeding pressurised sea water through a pressure and flow smoothing system to the generator turbine. The piston rod of each pump is connected to a crosshead driven by a connecting rod and crankshaft. A pair of pumps is driven by one crankshaft having two crank throws at 90^{o} to each other, and a train of gears drives the crankshaft from a pinion meshing with an internally toothed arc of rack which is attached to the outer pontoons and has its centre of radius on the centre line of the raft hinges. The pump crosshead, crankshaft and gear train is supported on a structure attached to the centre pontoon but passing through the raft hinge line to the outer pontoon rack, with a seal between the pontoons which encloses the machinery and allows it to operate in dry conditions. A bilge pump system pumps away any leakage of sea water past the seal. The maximum amplitude of pontoon angular rotation allowed for is at present $\pm 25^{o}$, but further raft tests may indicate that a larger angular amplitude should be allowed for.

The pump inlets are positioned at the underside of the centre pontoon using low pressure differential-suction valves, with singular non-return valves at the pump outlets for the feed of pressurised sea water to pressure smoothing air to water accumulation and to the generator turbines. Piston and piston rod bearing and seal assemblies are attached externally to the cylinder which therefore leaves the shape and design of the cylinder free of accurate machining and alignment other than the faces which support the bearing and seal assemblies. Access to the seals is therefore kept simple and from the outside of the pumps, and a temporary seal would be required to enable the main seals to be adequately maintained whilst in service. The type of cylinder shown would lend itself to being incorporated into the pontoon structure without the need for separate cylinder assemblies.

For servicing the cylinder seals it should be possible to temporarily open the pump inlet valves so that the pumps would not operate under pressure. These valves could incorporate a remotely operated lifting feature and a pressure limiting relief valve for power shedding purposes.

To ensure the maximum life and reliability of the piston and piston rod and the bearings and seals it must be possible for them to be aligned correctly to the same axis, otherwise excessive wear rates will result.

5.3.3.12 Pump Gear Drive

The Reference Design proposes a high speed increase gear ratio of the order of 32:1 and a pumping pressure of approximately 350 lbf/in^2 ($2.4\ MNm^{-2}$). This will allow the full stroke of the pump pistons to be to be utilised even for small angular rotations of the raft pontoons, but the flow rate required to utilise the full power generating potential of the raft will be correspondingly high resulting in large diameter and bulky piping between the pumps and the turbine.

5.3.3.13 Accumulator

The flow of water from the pumps would be directly proportional to the angular displacement of the rafts but would vary from zero at the trough and crest of a wave to a rate about 57 per cent greater than the average of the cycle whilst the raft passes through the mean sea level position.

To reduce the constant cyclic variation in power available and the associated wear and tear of the control mechanism of the power unit, it is desirable to achieve a reasonable measure of flow smoothing. This requires relatively large accumulator capacity. Air in the accumulator has to be at the same pressure but should not be in contact with the sea water. An elastic air bag within the accumulator might be a practicable method of segregation, particularly in view of the motion to which the accumulator will be subjected as part of the equipment in the centre raft.

Some work has been done in an attempt to estimate the accumulator volume necessary to smooth out the pulsations from random wave groups. It has been found that it would be possible to produce a smoothed hydraulic discharge for a period of between 30 and 60 seconds and with pressure variation within the limits ± 12.5 per cent. Even this modest smoothing would be of benefit to the power turbine and its control system. It would involve the provision of four large storage vessels, each 2 m diameter x 8 m long internally, two air/seawater and two air only.

5.3.3.14 Water Turbine - Type and Rating

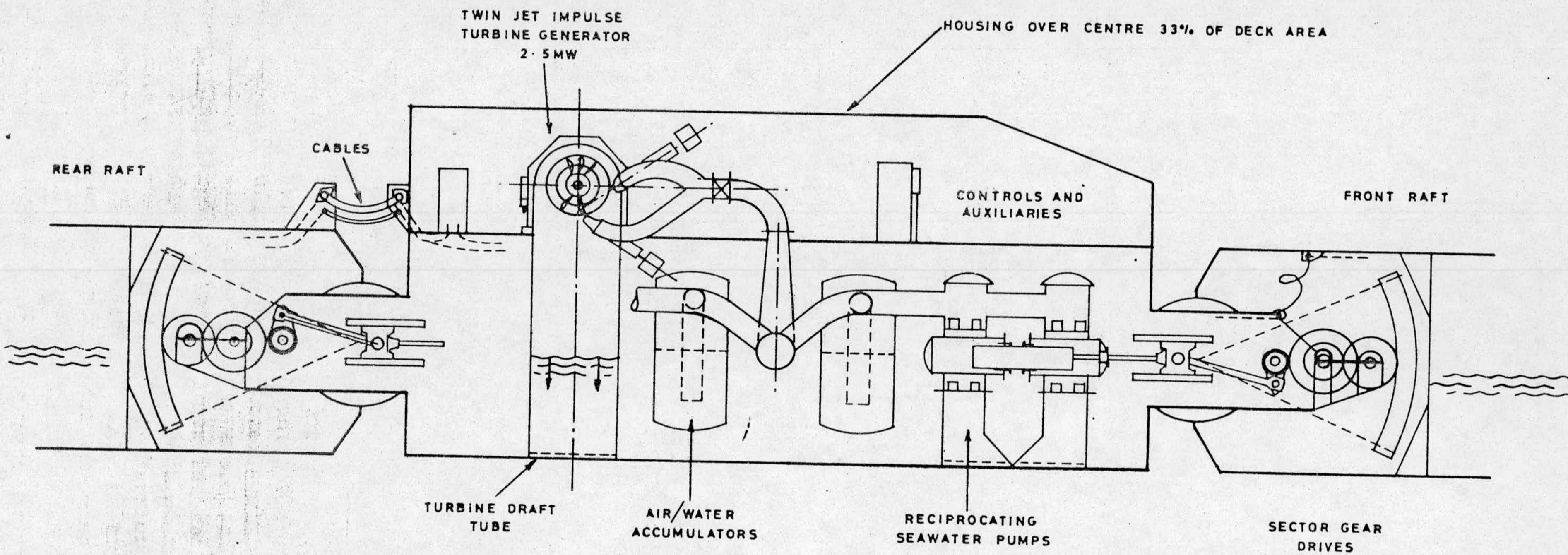
The pressurised water is used to drive a conventional impulse type water turbine rated for a maximum continuous output of 3500 hp. The turbine might be of the Pelton or Turgo type. Much of the energy abstraction will be at low part load, and consideration has been given to a layout using two 1750 hp turbines in a double overhung arrangement, the intention being that the part load efficiency would be improved by using only one machine below 50 per cent load. Further consideration has led to the adoption of a single 3500 hp turbine of the straightforward Pelton design and conventional bucket wheel but having two individually controlled jets. This retains the improved part load efficiency characteristic, requires less space, and would be cheaper. This arrangement of power plant is illustrated in Figure 5.2.

5.3.3.15 Turbine Generator Unit

Special Note - This unit is not shown on the Reference Design Drawings.

An essential requirement of an impulse hydraulic turbine is that it must have free water discharge beneath the runner. For this reason the turbine generator unit must be installed on the upper deck of the centre raft, there being a vertical water shaft, rectangular in section directly below the turbine and open to the sea at the bottom. A sudden load rejection relief valve could also be arranged to discharge into this duct.

It is proposed that the turbine generator unit (3500 hp.2.5 MW) should be enclosed in a watertight deck house on the centre raft. This would also contain the alternator excitation control equipment, main and auxiliary switchgear, 3.3 kV power and auxiliary cabling, and remote control communications equipment. Hopefully, space would be found in the

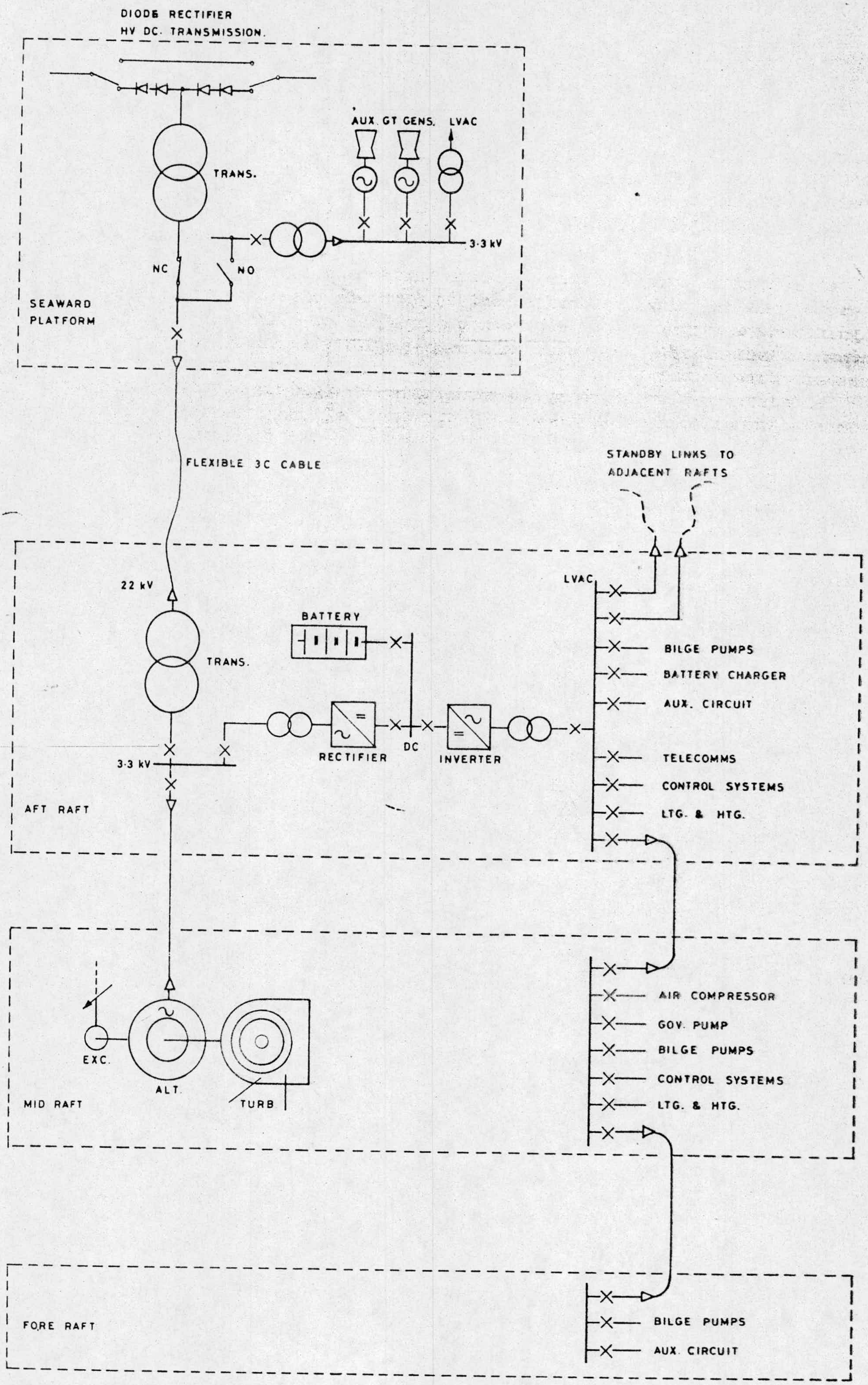


SCALE 1:200 (5mm to 1m)

ARRANGEMENT OF POWER PLANT IN CENTRE RAFT

WPL DEVICE

FIG 5.2



SCHEMATIC DIAGRAM OF RAFT POWER SYSTEM

WPL DEVICE

FIGURE 5·3

hydraulic pump room in the lower part of the centre raft for the hydraulic/air accumulators. If there are difficulties in doing so, then the accumulators would have to move into the forward part of the upper machinery space.

The turbine generator unit will be subject to the motion of the raft.. It would be advisable to install the machine on an axis parallel to the second hinge. This would equalise the loads imposed on the machine bearings resulting from the acceleration and deceleration of the rotating masses.

5.3.3.16 Transmission

The output of the 2.5 MW alternator, probably produced at 3.3 kV when at full load, would require to be raised in voltage to about 22 kV for efficient power transmission via submarine cable to the adjacent converter platform. The transformer required to do this should be installed in the rear raft where conditions are more stable and there is plenty of space. Consideration should be given to a transformer of the type already proving highly reliable as used in 25 kV a.c. locomotives.

An indication of the extent of the electrical equipment required on a three raft assembly is given in Figure 5.3 which illustrates the way in which the electrical output would be conveyed into the high voltage d.c. transmission loop, and also the method by which essential auxiliaries would be supplied in each raft. The reason for the intervening direct current stage in the auxiliary power supply is to overcome the problem resulting from variable speed and voltage generation. The auxiliaries rely on a steady a.c. voltage and frequency.

5.4. CONSTRUCTION AND INSTALLATION

5.4.1 CONSTRUCTION

The concrete pontoon shells would present a straightforward construction operation, either in dry dock or on slipway. The steel pontoon shell was reported by Swan Hunter to be ideally suited for construction within shipbuilding facilities where a major feature would be panel-line techniques. (It seems likely however that they have not yet appreciated the complexity which is now starting to appear). The proposed power take-off system could be handled within the engineering division of British Shipbuilders. Swan Hunter warn of the possibility of a bottleneck in production caused by the amount of mechanical engineering work. The large bearings and bearing castings would have to be manufactured as individual components. The time to produce a small number of comparable components on the Thames Barrier Project was 9 months for a bearing and 9 months for a hinge casting. A completely new U.K. works would be needed to make these components in any quantity.

5.4.2 PAINTING OF STEEL PONTOON

It is proposed that in a series-built system units are pre-painted in a painting hall with a controlled environment prior to final erection, thus ensuring ideal conditions. Only the jointed zones would require in-situ painting.

A painting specification has been prepared by Sigma Coatings Ltd. which is confidently expected to provide a minimum maintenance interval of ten years, including submerged areas, when used in conjunction with a cathodic protection system. Sigma Coatings suggest coating and immersing trial panels now to gain field results on the life of coating before the commencement of prototype building.

5.4.3 LINKING OF PONTOONS

This involves the insertion of the trunnion shaft into the hinge assemblies and also the alignment of the gear wheels in the racks and the assembly of the hinge seals. This operation is probably the most easily carried out in a fitting out dock where the pontoons can be easily manoeuvred.

5.4.4 INSTALLATION

This should present relatively few problems, once the mooring system is installed. The rafts are highly seaworthy and should have a relatively wide weather window.

5.4.5 TIMESCALE

The estimated number of rafts required to produce a peak mean output of 2 GW is 1143. If this scheme were constructed over say a 20-year period an annual average production of 58 devices would be required.

The constraints would be in the making of the large special components - castings, hinges and gears. These would require the installation of completely new manufacturing capacity in the U.K. to meet the specified programme requirement.

5.5. ASSESSMENT

5.5.1 GENERAL

At the time of the last report a number of areas were identified as potential problem centres requiring careful examination. Over the past year testing and desk studies have confirmed that most of them are indeed very real problems, and the solutions proposed, while individually feasible, have collectively added considerably to the cost and complexity of the device. In two problem areas, mooring and storm survival, feasibility is not yet confirmed.

5.5.2 STRUCTURE

Moving the power take-off inside the rafts, whilst beneficial to power take-off itself, has added considerably to the complexity of the raft structure. The centre pontoon is now a giant machine room, with machinery access holes in the top and also resists reciprocating machine loads. Design of the hinge components to meet the loads now predicted shows them to be massive and expensive, and difficult to splice into the pontoon structure. The front and back pontoons, in concrete, also require access holes for maintenance or replacement of the gears. There will be problems in fixing the hinge bearings, and some very heavy reinforcement will be required.

There is no doubt that the pontoons can be made. The attraction of being able to make them on slipways or in shallow basins remains, but they are no longer quite as simple as they were, or as cheap. Provision of a third hinge, which has been suggested, would add at least as many problems as it might solve.

5.5.3 HINGES

This is given a separate heading, although it is clearly part of the structure. It seems likely that the hinges can be made and lubricated to meet the performance requirement, although the requirement extrapolation beyond current experience.

5.5.4 POWER TAKE-OFF

This is now required to meet more adverse performance requirements than a year ago. The extreme angular excursions of the pontoons are increased and the angular movement at mean take-off power is somewhat reduced. The solution proposed seems feasible but expensive, with an excessively long power chain thus:

linear (pontoon) ⟶ rotation (gears) ⟶ linear (piston) ⟶ rotation (turbine) ⟶ rotation (generator)

At each of the first three stages the forces on the mechanical and structural components are very large. Forces from the first stage in the chain impose large transverse moments on all three pontoons. Access for installation, maintenance, and replacement of machine components is bad, and is likely to appear worse when the secondary power conversion module is placed on top of the centre raft.

The air environment for the gears depends on the integrity of a 'wiping' seal 50 m long, between a rotating cylinder and the pontoon housing. Seals of this type have been viewed with disfavour in other devices. Marine growth can build up on the extremities of the seal travel during calm weather and will then prevent proper operation in subsequent storm conditions.

Tolerance between the gears and the rack is dependent on the stiffness of a beam structure spanning almost 50 m. This is not ideal for a gear system.

No doubt some of the problems mentioned will find better solutions than in the admittedly preliminary design the Team have offered, but the big problem that will not go away is the one set by the nature of the Device itself.

5.5.5 MOORING

The mooring problem cannot yet be described as solved. A compliant four corner mooring system implies no redundancy, and a single failed mooring line releases the raft to rotate and move a long way. A linked moored array based on this principle is vulnerable to a single failure producing a domino effect, and could have catastrophic consequences.

5.5.6 SURVIVAL

The question of whether the raft can survive the worst conceivable wave is not yet resolved. The power take-off system presents the pontoons with a hard 'stop', and if they run up against the stop the massive inertia forces could well destroy the system. This is clearly a question that can be checked out and answered one way or another.

5.5.7 PERFORMANCE

The rafts appear unlikely to perform well in angled seas, but there is no quantitative data on this.

5.5.8 MAINTENANCE

As indicated, the main problem for maintaining the machinery will be to provide access through the structure. Routine greasing in fair weather, inside the structure, should be practicable, given safety guards to keep staff clear of the moving parts. Major maintenance and repair would have to be off station. Every effort will have to be made to avoid the need to split the rafts to gain access to the gears, and until a proper structural analysis is carried out it will not be known if large enough parts can be provided in the deck.

Unfortunately, the primary power offtake units cannot be modularised for this Device, which is clearly a disadvantage. The secondary power conversion plant is modularised.

Maintenance of the steel hull will not be a problem, but simply an additional and continuing expense.

5.5.9 CONCLUSION

The overiding impression of the Raft Concept is that it is progressively changing from a simple conceptual design into a complex and expensive piece of machinery. This trend is paralleled by other articulated devices in the programme, and it seems likely that the cost and complexity is associated with the principle of extracting power by means of large forces which have to pass through moving, oscillating, linkages.

In considering the direction of future work by this Device Team it seems clear that desk studies of a number of the important cost centres must have a high priority. These centres include elements of power take-off, hinges and structure, and mooring. To get the results of these studies on a properly quantified basis, more input data of forces and movements is required. Performance in directional seas, and survival are also key areas.

It is thus apparent that the effort needed to improve the Device must be applied in all areas.

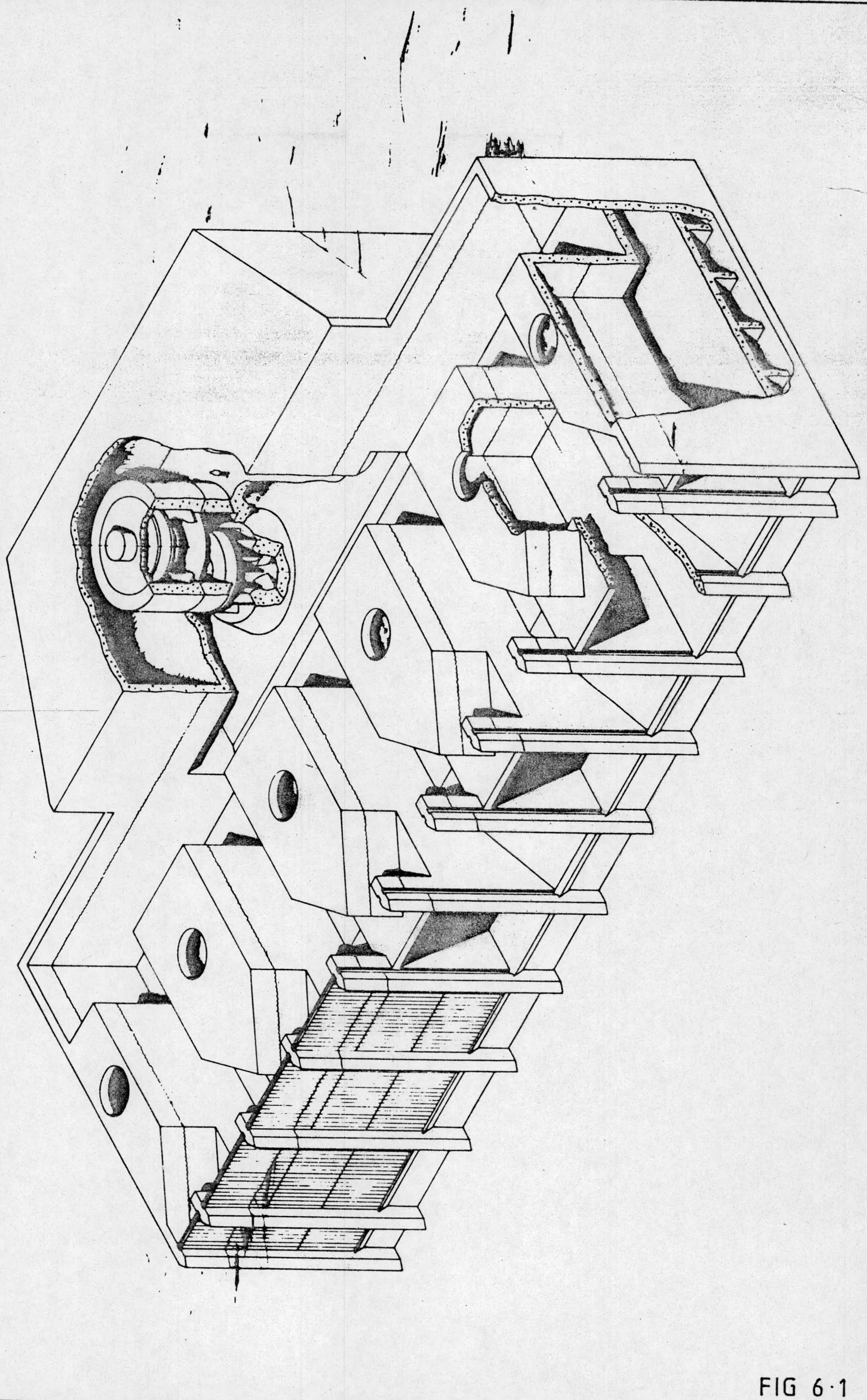


FIG 6·1

CHAPTER 6 - HRS RECTIFIER (DEVICE TEAM - HYDRAULICS RESEARCH STATION)

6.1 GENERAL DESCRIPTION OF THE CONCEPT

6.1.1 DESCRIPTION OF THE DEVICE

A long line of caissons resting on the sea bed in approximately 15 m depth of water is aligned parallel to the general shore line. The caissons project above the water level by some 5 m and will be sited between 1 km and 5 km from the shore depending on the gradient of the sea bed. Viewed from the shore the devices will thus present an unobtrusive low horizontal profile similar to a distant breakwater wall. Discrete lines of devices are separated by gaps of the order of 1-2 km, depending on contours and the requirements of navigation.

A single device is a large, rectangular, hollow caisson with a system of internal reservoirs and an integral module containing the power plant. The vertical seaward face of the device is provided throughout with an array of panels of one-way flap valves, their hinges aligned vertically, and arranged alternatively to allow water to flow in or out.

The device is divided internally into two reservoirs by a slab which extends horizontally for the full length of the device. The reservoir above the slab has a free water surface open to the sky. The outlet reservoir below the slab is provided with a free surface by chambers which project upwards at intervals through the dividing slab and which have a vented roof level with the top of the outer walls of the device. Two large low-head kaplan turbines are placed in the flow path between the reservoirs. The generators, driven directly through vertical shafts, are in a machine house above the turbines.

6.1.2 PRINCIPLES OF OPERATION

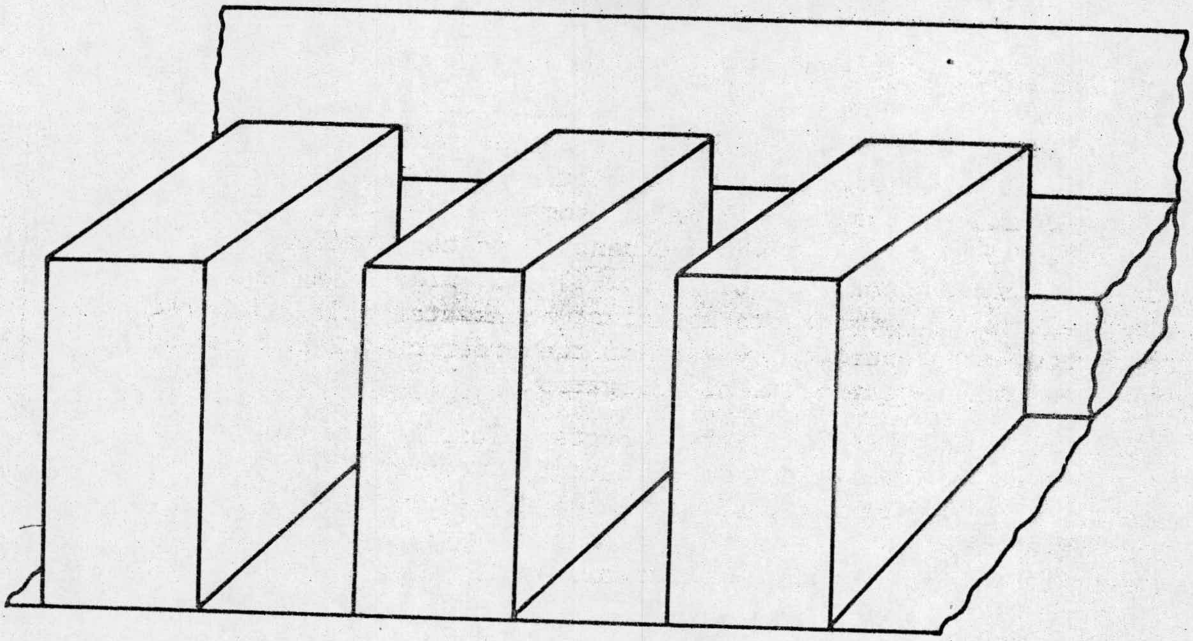
A head difference is maintained between the water in the reservoirs by the inlet flap valves to the upper reservoir collecting water during impinging wave crests and the outlet flap valves to the lower reservoir discharging water during wave troughs. Flow between the reservoirs drives the turbines.

6.1.2.2 Geometric Parameters Related to Device Performance

The device concept allows a very wide freedom in the organisation of the system of valves, reservoirs, and structures.

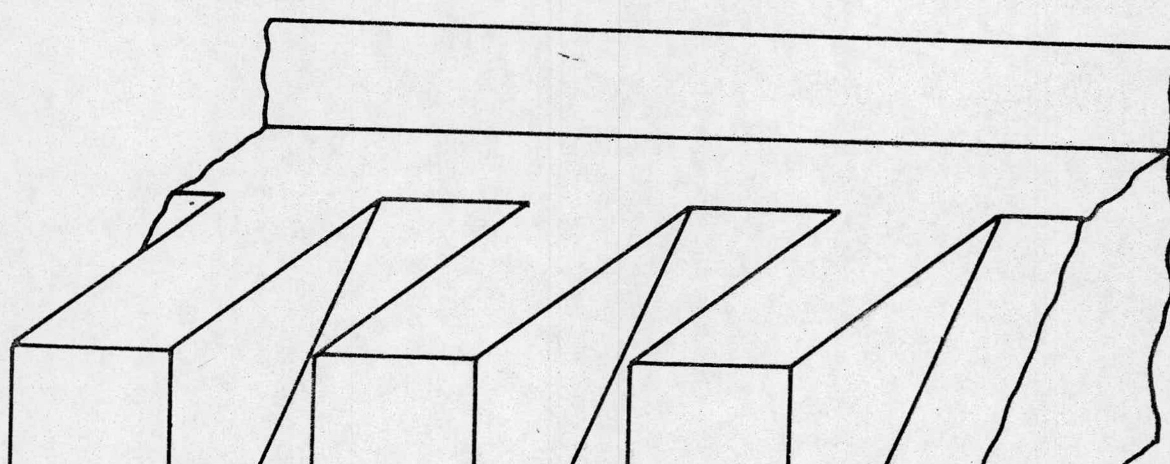
A number of significantly different options are available for the basic layout, and the problem has been to identify the layout which leads to the most cost effective solution. The aim has been to reduce structure cost whilst maintaining productivity.

Overall it will be seen that the basic plan area of the device is governed by the need to capture in a high level reservoir as much as possible of the water delivered through the inlet valves. The usable reservoir volume is a simple function of the width of the device, and it is this dimension which has changed most significantly over the last year, as a result of development work. The height of the device is simply from the sea bed to some optimum level high enough above the high tide level to ensure that a suitably high proportion of the energy available in the working sea can be retained. The device is thus inherently large.



SERIES K CONFIGURATION

FIGURE 6·2



SERIES L CONFIGURATION

FIGURE 6·3

6.2 STATE OF DEVELOPMENT OF DEVICE

6.2.1 GENERAL

The test programme has been running at the HRS for about eighteen months. Thus far work has been confined to 1/30 scale two dimensional models tested in an 8 ft. wide flume in monochromatic waves. Two different layouts have been tested each with three different structure widths. Captured power has been defined as the product of the flow between the reservoirs and the pressure head maintained between them, under steady state conditions. In the model, all head losses incurred in flow between the two reservoirs was physically compensated for by a by-pass pump, so that in calculating the power available from the device, allowance has to be made for all internal pressure losses. Efficiencies are defined as water power through the device divided by power in the impinging waves.

Work at HRS has been concentrated on overcoming modelling problems associated with the flap valves, which are difficult to model at small scale, and then on experimenting with different layouts in an attempt to maintain or improve hydraulic performance whilst reducing structure cost.

Structural design and costing based on the latest caisson layout has been carried out subsequent to the tank testing. Design work has also been done on the turbines and on the flap valves.

6.2.2 THEORETICAL STUDIES

A preliminary attempt has been made to optimise the height of the structure. This has been done by placing it in the shallowest water possible and reducing its vertical projection above water level. The limitation on minimum water depth will be the physical space required to accommodate the turbines and draught tubes, whilst the projection above water level affects the percentage of available energy which is captured by the device. Both these considerations are in a preliminary stage and the presently adopted values of 15 m water depth and 5 m projection at mean low tide level cannot be considered as finalised.

6.2.3 GEOMETRIC CONFIGURATION

The device configuration described in the Consultants' 1977 Preliminary Report has been the subject of significant development over the last year.

A series of tests was concluded on the type 'K' configuration which was the basis of the 1977 Reference Design. (see Figure 6.2). In these tests, front-to-back scaled device widths of 25 m, 37.5 m and 50 m were tested in scaled waves of 4.4 m and 2.3 m heights and 12 sec. period. Efficiencies of 65% maximum were reached in the 50 m wide model with little variation in efficiency between the two wave heights. In the 37.5 m wide model maximum efficiencies of 63% and 69% had been achieved for wave heights 2.5 m and 4 m respectively (both of 12 second periods) whilst for the 25 m wide model efficiencies had fallen to 55% maximum. The Team therefore decided that 35 m would be an optimum width to reduce the structure size as much as possible whilst maintaining efficiency as near as possible to the maximum.

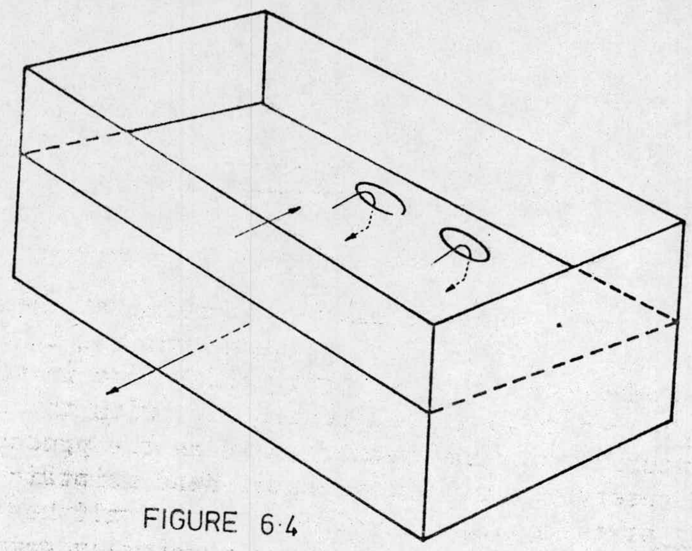


FIGURE 6·4

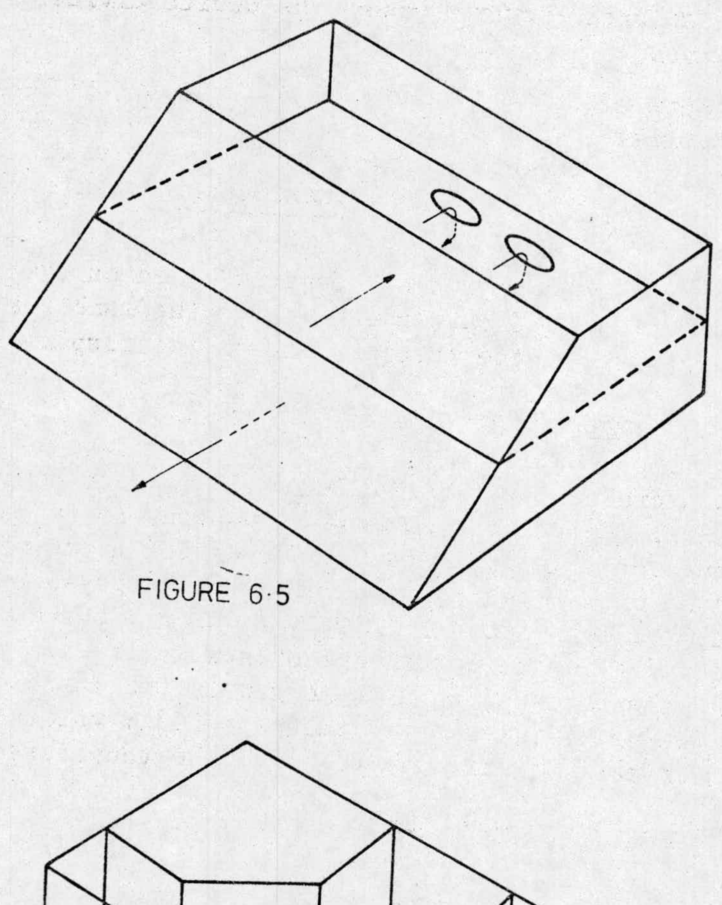


FIGURE 6·5

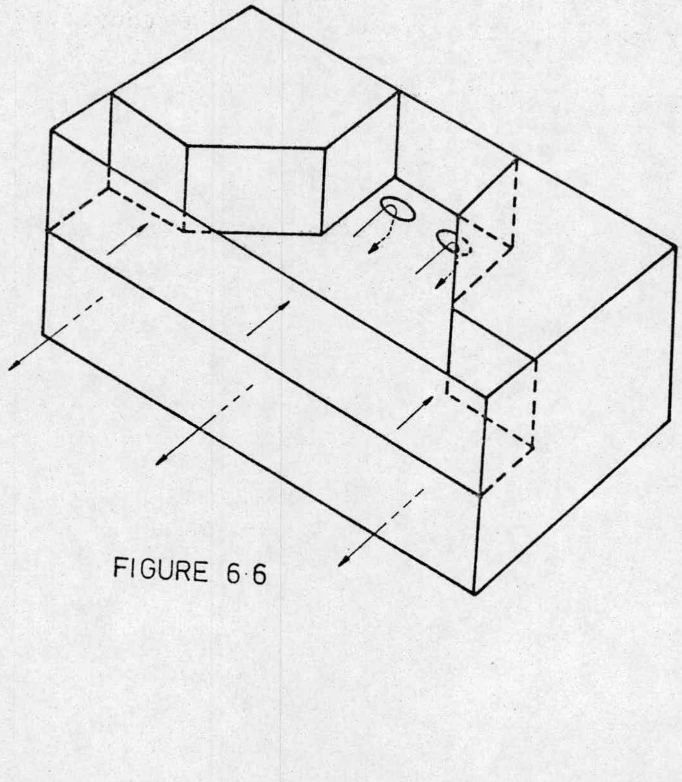


FIGURE 6·6

A significant part of the structure cost in the 'K' configuration is incurred by the numerous full-height internal vertical dividing walls. The Team's efforts have been directed towards deriving a structural layout which will minimise structural costs whilst improving efficiency. Consideration has been given to layouts in which the inlet and outlet reservoirs are divided only by a horizontal slab, all inlet valves being above the slab and outlet valves below (Figure 6.4). This is believed to be less efficient due to the amount of energy at lower depths of water which is fully reflected by the outlet gates and also because the outlet reservoir has no free water surface. The Team considered reducing the first objection by installing the inlet and outlet gates on a shallow ramp so that energy from lower levels could run up the ramp and enter through the valves. (Figure 6.5). The difficulty with this arrangement would be in designing inlet gates to close against gravity with very slight back pressure. Also the outlet reservour would still lack a free surface. The Team has made a model of this configuration and visually it appears to work, but no quantitative tests have yet been carried out on it.

Yet a further layout was considered in which the outlet reservoir is provided with large free surfaces in upstand chambers which are concentrated at the two ends of the structure the remainder of the dividing slab being horizontal (Figure 6.6). The Team considered this also would be less efficient due to the requirement with this layout to move large volumes of water longitudinally within the structure during its cycle. Losses would also occur at the gates due to inlets and outlets being too widely spaced to allow wave refraction to operate.

The Team has therefore developed the layout to the series 'L' configuration as shown in Figure 6.3. In this the outlet reservoir side walls do not extend fully to the base but are supported on columns. The inlet gates are in relatively narrow widths which extend to the bottom of the structure and are thus able to collect energy from all depths and by refraction. The outlet reservoir has a large free surface area and is spread along the length of the structure. Water collected from lower levels is led up a ramped slab above the outlet reservoir.

This configuration also increases the inlet gate area. There is now about 2½ times more inlet area than outlet area. The Team believe this is necessary since the time during which a trough is impinging is much longer than that during a crest, real wave profiles being trochoidal.

It should be noted that having made the above modifications the areas of structural walls and slabs for a unit length of device of 20 m (i.e. one inlet and one outlet path), calculating as though the 1977 Reference Design was also 20 m high and 35 m wide, are virtually the same as for the series 'K' layout; that is, the actual structural advantage appears to be negligible.

Initial tests on this configuration showed efficiencies of about half those in series 'K' until it was realised that the air trapped above the outlet reservoir was pressurised and therefore reducing the head drop across the device. Vent tubes to atmosphere were introduced in the roof slabs and efficiencies were thereby improved.

The length of the device has not featured in the experimental work, which is essentially two dimensional. Length is determined almost entirely by practical design and construction problems. It is discussed in 6.3.4.1 and in 6.4.5.

6.2.4 EXPERIMENTAL RESULTS

The efficiency curves for the series 'L' experiments show peak efficiency of 50% for 12 second waves and 43% for 10 second waves, considerably less than the Team reported for the series 'K' experiments. The Team however is hopeful that further refinements in the configuration will raise the efficiency to its series 'K' level, or above.

6.2.5 FLAP VALVES

The Device Team have now developed a tapered rubber flap which works very well in the model. The thin section at the attachment point provides minimal rotational stiffness as resistance to opening but enough stiffness to return the flap to the closed position when back flow commences, whilst the thicker section over the remaining width provides midspan bending strength against back-pressure. The flaps are fixed to the vertical posts by lines of small rivets.

Whilst this arrangement works well at small scale there will be complications at full size. The rubber flaps will have to be reinforced with metal plate in order to carry bending moments from the back pressures. For this reinforcement to obtain support from the side mullions, the width of the mullion containing the hinge, will have to be wide to accommodate a further length of unreinforced flap in which flexure can occur. The concentration of repeated flexure in a local zone will then impose a fatigue condition on the rubber which will limit its life. A further problem may be warping as the flaps hang from their flexible hinge, subjected to gravitational load.

6.2.6 WEED FOULING AND SILTATION

A start has been made on the question of marine growth on the hinges, and a trial hinge is soon to be placed in the sea. Nothing has yet been done to investigate the potential problem of choking the device with drifting kelp, and stone that may come in with it. Likewise the potential problem of silting up with sand has not been investigated.

6.2.7 SITES

A preliminary study has been made of potential sites. This is reported in 6.6.

6.2.8 STATE OF DEVELOPMENT OVERALL

Less work has been done on this device than on the other three leading devices, and a great deal of work would need to be done on the subjects of valves, fouling and silting before a decision could be made to proceed to large scale. On the other hand, further major reductions in cost are unlikely to be forthcoming unless a radically different and simplified structural configuration can be developed, and a significantly higher efficiency is correspondingly achieved.

The structural thicknesses calculated for the Reference Design are subject to refinement and accurate determination of wave loading. Also the use of struts, closer column spacings, different diaphragm spacings etc. would marginally reduce the volume of concrete (but not necessarily the cost). Overall, the Consultants believe that the scope for significant reduction in the present configuration is small.

The performances already measured in monochromatic two dimensional waves are hardly likely to improve in three dimensional, more realistic conditions.

6.3 THE REFERENCE DESIGN ADOPTED FOR THE STUDY

6.3.1 INTRODUCTION

The Consultants' brief to report in August 1978 found the Device Team in the middle of a test programme investigating alternative configurations. The Consultants emphasise that the thinking underlying the present Reference Design corresponds to a stage in that programme and is not a completely researched conclusion. This design is based on the Team's series 'L' configuration, but with the level of the outlet reservoir roof raised to the level of the inlet reservoir walls to provide a larger free surface. It is necessary to provide a large free surface to both inlet and outlet reservoirs in order to smooth the cyclic variation in head difference between the reservoirs, which is the parameter governing the device hydraulic efficiency and to which the turbine design is related.

6.3.2 GENERAL SPECIFICATION

- Location - West of the Outer Hebrides.
- Water depth - 15 m at mean low water.
- Arrangement of units - Sunk on prepared gravel bed laid along a sea bed contour. Adjacent units are to be placed 2-5 metres apart. A series of units form a string of 3-6 km length, and adjacent strings are separated by a 1-2 km gap to allow ships to pass. (See Drawing WP78/RECT/2)
- Dimensions of unit - Length - 100 m
Height - 20 m reservoir, 25 m turbine housing
Width - varying from 36 m to 59 m at the turbine housing.
- Material of structure - Reinforced concrete, Grade 40, for main structure. Structural steelwork, grade 43D, with rubber/steel flaps for gate units.
- Float-out loading - 3 m design wave height, 14.5 m draught.
- Operational loading - Differential head of 16.5 m between inlet and outlet reservoirs.
- Estimated weights -

Concrete	55,600T
Steelwork in gates	1,307T
Ballast	16,100T
Machinery	1,250T

- Turbines per unit - 1 No. Kaplan turbine rated at 2.5 MW for a 3 m head.
1 No. Kaplan turbine rated at 1.0 MW for a 2 m head.

6.3.3 DEVELOPMENT OF THE REFERENCE DESIGN STRUCTURE

6.3.3.1 Caisson Structure

The overall longitudinal strength of the caisson to resist float out and installation loads is provided by the cellular base. This is the only part of the structure which extends uninterrupted from end to end. The practical limitations on the strength that can be built into the base are in turn factors which limit device length. The base cells also supply facility during tow-out for partial ballasting and trimming as required. The strength and thickness of external walls and internal partitions are all governed by hydraulic pressures.

The back shape of the caisson has been determined by the need to accommodate the turbines and generators, which cannot be fitted into a simple 35 m wide rectangular caisson.

6.3.3.2 Caisson Stability

The device is required to remain stable on a prepared gravel bed under all possible wave loadings. This is accomplished by increasing the weight of the caisson after installation by pumping sand into the base cells.

6.3.3.3 Draught Tube

The shapes of the draught tubes are very important in allowing the turbines to realise their maximum potential efficiency. The shallow depth of water in which the device is being placed requires that the draught tubes must encroach into the cellular base under the turbines before turning a right angle and widening out. The presence of the draught tube breaks up the continuity of the cell base structure, and will weaken it to some extent. A torsion path is provided round the back of the tubes to improve structural performance, but the precise influence of the large 'hole' in the base on overall structural strength has not yet been checked.

6.3.3.4 Flap Valves

Flap valves are mounted on removable steel grillage panels which can be lifted in and out of slots provided in the concrete dividing walls which support them. During tow-out the panels are replaced by blanks to seal the structure for flotation. The valves themselves are based on the Device Team's ideas referred to in 6.2.5.

6.3.3.5 Weed Screens

No weed screens are shown on the Reference Design. If weed is proved to be a problem, then coping with it will be a major undertaking. The Consultants have no firm suggestions to make as to how the design might be modified, or where screens might be placed. The problem of weed is further discussed in the overall assessment (6.7).

6.3.4. DEVELOPMENT OF REFERENCE DESIGN POWER PLANT

6.3.4.1 Turbines

The results of experimental work indicate that energy conversion will be optimised if the water flow through the power plant is adjusted so as to maintain a head ratio h/H within the bracket 0.6-0.9 where h is the level difference between the reservoirs and H is the wave height. This is consistent with the peak efficiencies mentioned in State of Development (6.2). Development work so far has confirmed a mean efficiency 0.40 within these limits, but it is expected that refinement of hydraulic design will result in improved efficiency.

A particularly difficult feature of the hydraulic energy conversion is the exceptionally low net head available to drive the water turbines. The National Engineering Laboratory undertook a review of alternative turbine types for low head operation. It is generally agreed that the Kaplan turbine is likely to be the most suitable and most efficient design, and a normal vertical axis machine of this type has therefore been retained for the 1978 design.

An alternative design with a horizontal axis but still using the Kaplan principle, and producing the so-called 'bulb' turbine, has been examined. The arrangement is, however, unsuitable owing to dimensional difficulties and the need for access to the electrical equipment from above.

Water capture through the intake gates into the upper reservoir - and correspondingly water discharge from the lower reservoir - depend on the amplitude of the wave oscillation at the seaward face of the device and also the period of oscillation. Both quantities vary with wave climate and it is now clear that it is not appropriate to provide a single machine to abstract energy from the inter-reservoir water flow. The present Reference Design employs two Kaplan turbine generator units, the first rated 1.0 MW when h = 2.0 m and the second 2.5 MW when h = 3.0 m. The two machine arrangement should lead to an improved energy conversion efficiency over a more practical range of operating conditions. The installed capacity is based on an assessment of the water productivity of the upper reservoir and it appears that this might vary from 15 cumecs with h = 1 m up to 165 cumecs when h increased to 3 m. It should be noted that even the smaller turbine will not produce any output when the head drops to about 1 m, there being then just sufficient power to sustain the turbine unit at speed on no load.

The 1 MW unit would be used in relatively calm seas with wave amplitudes between 1.5 m and 3 m; the 2.5 MW unit when the head improves and is within the range 2.5 m - 5.0 m. If the water productivity of the upper reservoir can sustain more than 2.5 MW, then the smaller machine can be brought into service again since it would still operate at 0.75 - 0.80 efficiency with the head at 3.0 m. A maximum output between 4 MW and 5 MW should be achieved under appropriate sea conditions and with efficient inlet and discharge valves.

Operation at exceptionally low head calls for a Kaplan turbine design of high specific speed - N_s greater than 950 (metric). The runner of the 1 MW set would be about 4 m diameter and that of the 2.5 m set about 5 m diameter. In both cases the speed would be 60 r.p.m. Under design conditions each turbine may be expected to operate at an efficiency of .90. This, however, would be confirmed by hydraulic model testing during the plant design phase.

The foregoing turbine parameters have been confirmed by Boving and Company Limited, London, who are the recognised U.K. water turbine design engineers.

The proposed layout of the power plant in the rear section of the HRS device is shown in Figures 6.7 and 6.8 . The prominence of the turbine draught tubes is noticeable. At these very low heads it is even more important to recover energy due to the velocity of the very considerable water flows in the draught tube which decelerates the turbine discharge. An attempt has been made to achieve an efficient arrangement. The exit velocity into the lower reservoir would be quite low but in keeping with the general flow rate towards the outlet gates. In practice it would be essential to undertake a series of hydraulic model tests in order to confirm the most efficient draught tube arrangement within the confines of the device dimensions.

It should be added that efficient turbine performance under varying head and flow conditions would be achieved by conventional combinator gear. This involves co-ordinated adjustment of the turbine blade angle with that of the inlet guide vanes. It is achieved by suitably designed cams and a high pressure oil servo system.

6.3.4.2 Generators

Reference to Figures 6.7 and 6.8 shows an unusual feature of the two machines. Owing to the low turbine speed, it has been necessary to include a 3:1 double helical gearbox, so that the alternator speed is nearer to 180 r.p.m. This gives a more conventional electrical machine which can be accommodated within the limited space available.

In this design the combinator equipment passes through the turbine shaft only. Separate thrust bearings have been provided for both turbine and alternator. It would be necessary to arrange that the vertical combinator rod can be installed or withdrawn between the spokes of the alternator rotor.

The waterwheel generators would be of the three phase type wound for 3.3 kV and with superior insulation for continuous operation in a damp atmosphere. Nevertheless, closed circuit air cooling would be used with external heat exchangers cooled by sea water circulation.

Similar excitation arrangements would be used as for all other wave energy power converters, i.e.brushless thyristor type rotary exciters with a wide excitation ratio of about 4 p.u. and short response time. This would be necessary for operation into a series diode converter chain which would be arranged somewhat differently on the HRS device compared with all others. The device being sea bed mounted and static, and furthermore relatively close inshore, the oil immersed isolation transformer/diode units would be mounted in the machinery space at the back of the concrete caisson, together with their control switchgear and cooling arrangements. This is not at present shown on the layout drawing, Figure 6.7.

There would be two sets of 42 HRS units in each 250 MW group. It is proposed that positive, negative and mid-point cables be taken ashore from the centre of the device array. Each generator would there-

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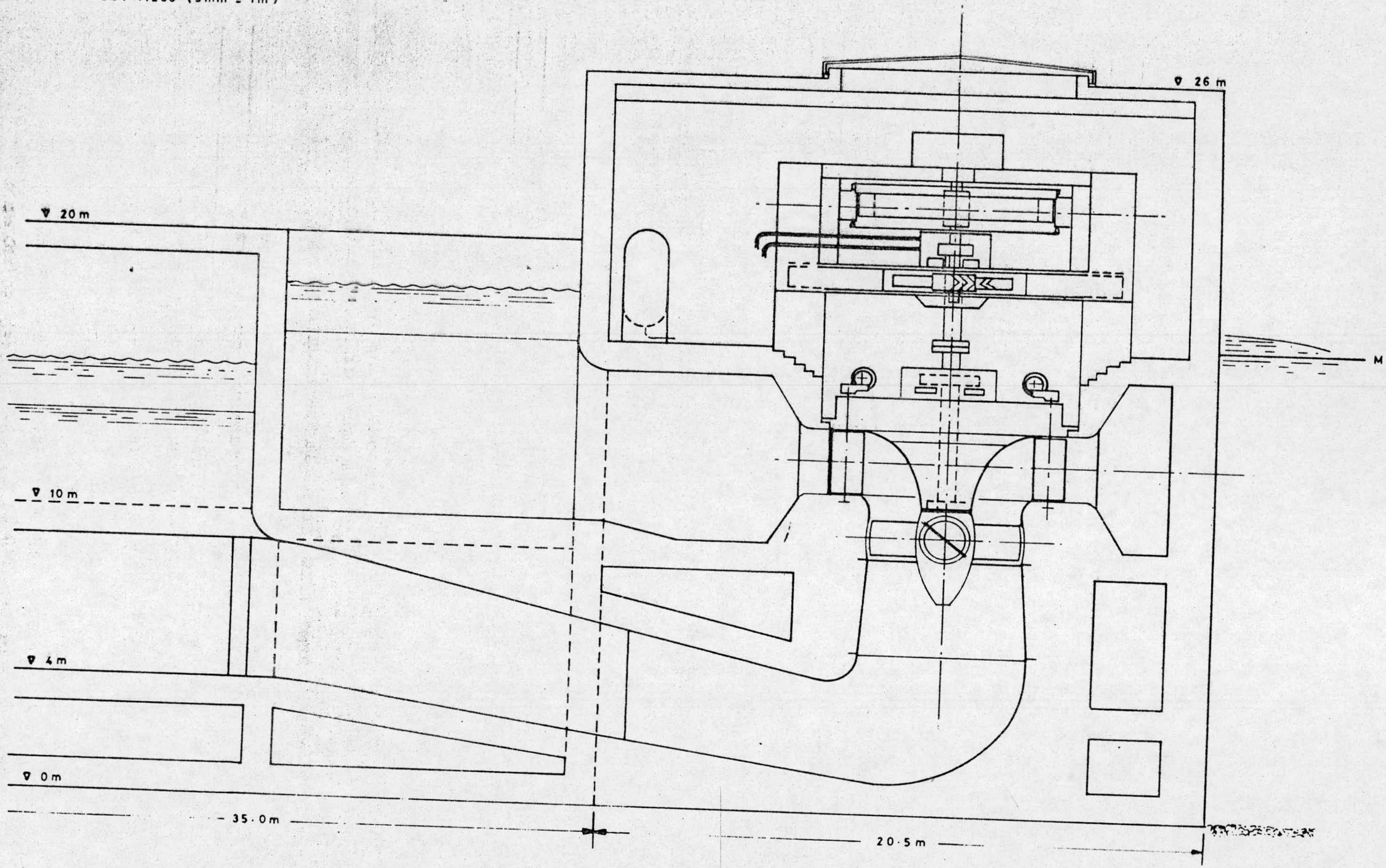
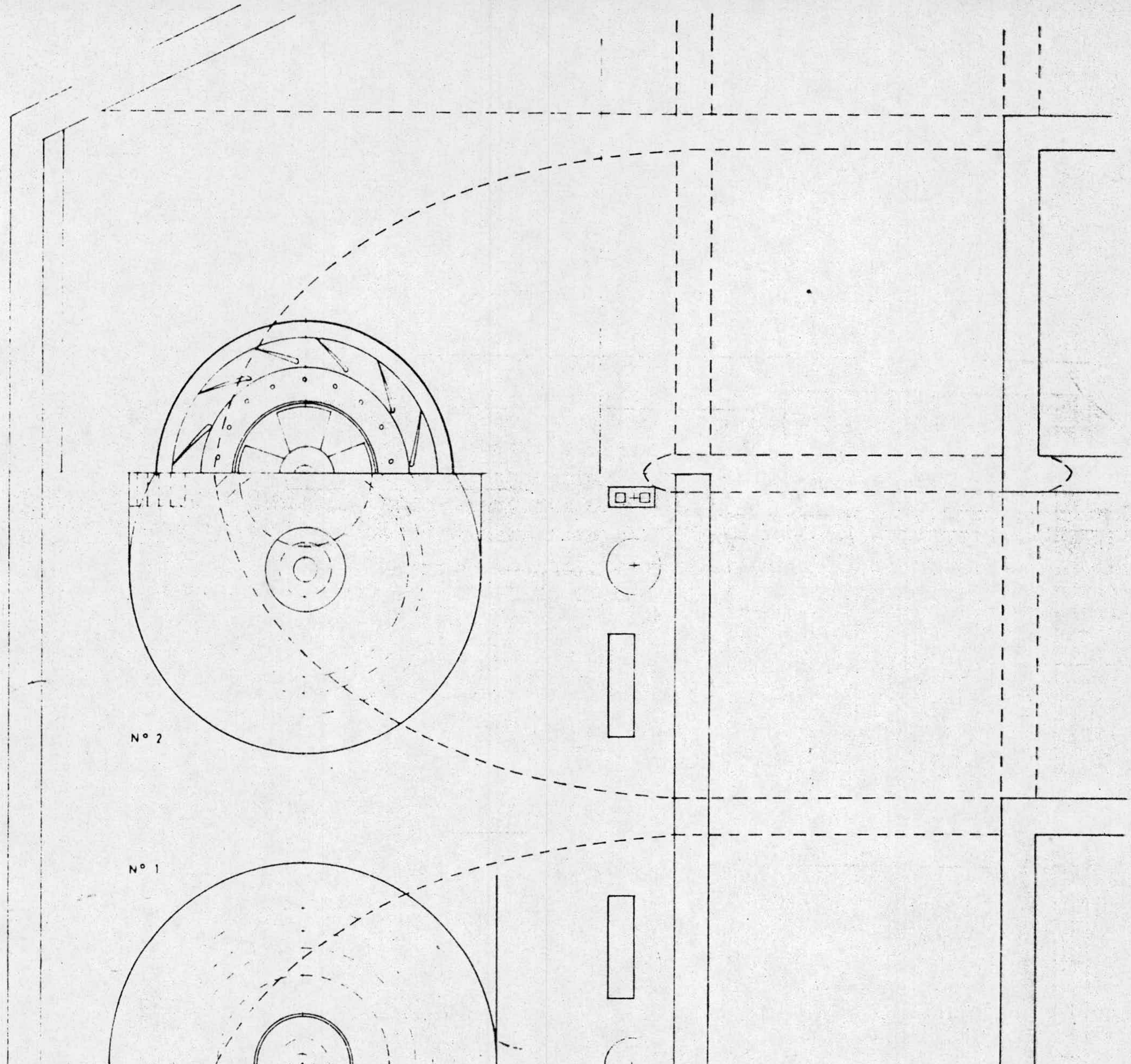


TRANSVERSE SECTION AA THROUGH No.1 2500 kw KAPLAN TURBINE/ALTERNATOR SET. GEARED 60/180 rpm.

H R S DEVICE.



PLAN OF POWER PLANT SECTION OF DEVICE.

UNIT 1. 2500 kW 3m HEAD
UNIT 2. 1000 kW 2m HEAD } BOTH UNITS GEARED 60/180 r.p.m.

HRS DEVICE

FIGURE 6·8

fore be connected to its own small section of the d.c. series loop, the HVDC cables passing directly from one device to another. A particular virtue of the arrangement is that transformation to 22 kV and the use of flexible submarine cables are eliminated. This would result in a useful reduction in cost of plant and power collection.

Reference has earlier been made to the need to maximise the 'live' storage volumes of the upper and lower reservoirs. The device faces the problem of utilising waves of random height by means of reservoirs which change level according to the input sea and the turbine control. This points to the necessity for automatic means of reservoir level optimisation. Water flow through the turbines should be adjusted by guide vane control so that energy abstraction is at a maximum and to this end the differential head between reservoirs should be kept as near as possible to 0.9 H.

A final observation is that the slow speed water turbine generator units for the HRS device will be relatively robust and reliable pieces of equipment able to survive in a sea water environment provided suitable metal alloys are used in their construction. Arrangements for electrical power collection are simplified and the device would be less susceptible to loss of auxiliary supplies.

6.4 CONSTRUCTION AND INSTALLATION

6.4.1 INTRODUCTION

The study and development of the device in terms of its performance insitu must be coupled with appraisal of the practical problems of construction and installation. In this device theoretical requirements and practical limitations are frequently in conflict. Thus:-

a) the alternating pattern of ramps and outlet reservoir prevents a simple structural layout with simple detailing and construction procedure;

b) the open side containing the flap valve will cause complications in making the structure buoyant for tow out, and strong enough for installation;

c) the draught and area of the structure will limit the number of suitable dry docks;

d) the limited freeboard on tow out will be a hazard to flotation in rough water;

e) the uncompartmental layout will tend to lead to instability in the floating condition if water is shipped. Temporary positions should therefore be added for the float out and sinking operations;

f) the site exposure which the working device requires will produce severe weather problems for caisson placing.

In this section these and other problems are considered.

6.4.2 CONSTRUCTION

The Reference Design is calculated to have a floating draught (ballasted to trim) of 14.5 m. This could probably be reduced to about

13 m when more refined structural analysis is carried out, but even so the only existing docks which could provide this draught without modification are at Portavadie (14.32 m), Nigg Bag (13.7 m) and Hunterston (14.0 m). The docks at Kishorn and Ardyne Point would require deepening by some 2 m.

Construction of the concrete structure is not expected to present any major problems beyond the logistics caused by sheer size. The power module would be commenced first since the fabrication work in building in the draught tubes, turbines and electrical plant will place this work on the critical path. Due to their size it is not possible for major components of the turbines to be installed after the concrete shell is completed.

At the request of the Device Team consideration has been given to the possibility of constructing the device in sections and splicing them together on the site but the practical difficulties involved coupled with the irregular nature of the configuration do not make this a feasible proposition.

6.4.3 FLOAT-OUT

The relatively low freeboard of 5.5-7 m and open top would render the structure liable to overtopping by wave crests during tow out. Depending on the weather windows accepted for tow out, it may be necessary to take measures to prevent the shipping of water. An alternative would be to tow out the whole structure supported between massive pontoons which would be flooded to place the device at site and then re-used. With this system some economies could be made over the present Reference Design in the design of the permanent cellular base, which would have a less onerous duty. The economics of this approach have not been investigated.

6.4.4 BED PREPARATION

The uneven sea bed would need to be provided with a level bed of crushed rock under the device. The estimated volume of 10 million cubic metres for a complete Hebridean scheme would have to be imported from quarries yet to be identified, probably by barges. The bed preparation is a very important element of this scheme. The siting and orientation of devices will in part be governed by the need to minimise bed preparation. Techniques currently exist for levelling rock fill underwater, but if this scheme went ahead it would almost certainly stimulate new developments in this area.

6.4.5. SINKING

The draught of the floating structure is very little less than the depth of water in which it is proposed to sink it. The cellular base will provide sufficient ballasting capacity for the sinking operation. Controlling position during sinking will be difficult when the structure is broadside on to the waves. Its final position will depend on which side of the base touches bottom first and this is to some degree uncontrollable in waves. For close spacing, fenders would need to be provided between devices. Even so the enormous forces which could be involved in an impact could cause severe structural damage. The question of linking adjacent devices is discussed in the next paragraph.

6.4.6 HYDRAULIC LINKING

The Device Team has expressed the wish to hydraulically link caissons in a long line in order to improve hydraulic smoothing of the reservoir levels and so obtain improved efficiency. Last year the Consultants accepted this idea and there is no doubt that it can be done, but at additional cost. The 1978 Reference Design does not allow for linking. The following practical problems of linking should be noted.

The positioning of the large caisson on the sea bed will be a difficult operation to control, especially considering the asymmetry of the structure, the large structural mass and the Atlantic sea conditions. The Consultants are very doubtful whether it would be possible or desirable to control sinking so that the caissons are placed within 1-2 m of each other at the closest. Hydraulic linking would then have to be provided over a gap of this order, with a plus minus tolerance of the same again. The hydraulic linking structure must then meet very severe design requirements; these include:-

a) differential settlements in service would occur and would have to be accommodated at the linkages;

b) the joint structure would have to withstand the maximum hydrostatic head that could occur between the upper and lower reservoirs. This occurs when the caisson is being drowned by an extreme overtopping wave and may be of the order of 15 m head of water. The joint structure would have to be capable of accommodating the irregularities in the alignment of the caissons, in respect of six potential degrees of movement, and also of accommodating deliberate 'steps' in the line as particular caissons were adjusted in line or level to allow for changing bed level or curvature of the system in plan;

c) the caissons themselves become more complex with the provision of large ports in the end walls, all needing to be sealed off during replacement.

Solutions variously using cast or in-situ concrete, and/or made to measure sections have been discussed and considered, but unless the rewards in terms of performance can be shown to be really worthwhile, the ideas are unattractive.

The Consultants question whether longitudinal hydraulic smoothing over hundreds of metres would improve the hydraulic efficiency. It us true that a minimum storage volume is required to maintain a permanent flow through the turbines when the inlet flaps are closed, but if this is to be achieved by the transference of water in the reservoirs over a long length longitudinally from a section where the flaps are open, it could be expected that the overall efficiency of the device would fall. This is a question which the Device Team need to investigate numerically, and perhaps in three-dimensional tank tests.

Scour

The presence of gaps between the caissons would tend to promote unacceptable scour of the bed. The crushed rock fill will be protected against scour between caissons, and on front and back faces by properly designed rock scour protection.

6.4.7 TIMESCALE

If the complete Hebridean scheme of 750 to 1150 devices were built and installed continuously, with an average rate of construction of only one year per device, and all the existing large docks were used for the purpose (assuming they were deepened as necessary) so that the following numbers of devices were under construction simultaneously:-

Hunterston	3
Ardyne Point	2
Kishorn	1
Nigg Bay	9
Portavadie	4
	19

then the time to construct the scheme would be 40-60 years.

6.5 MAINTENANCE

Maintenance would be required on the flap valves and on the power plant. Flap valves would be made in panels and lifted in and out of the slotted columns by a floating crane. It may be possible to lift only when the flaps are open (i.e. no reversed hydrostatic pressure causing friction forces between the panel and the column slots) though this should not prove difficult. However, the provision of a crane for this purpose is a problem, since the valves are on the unsheltered sea face, and a long reach floating crane stationed behind the devices will be required.

The turbine runners and draught tubes would need to be dewatered at regular intervals for cleaning and any necessary repair. Provision for this is made by a series of gates which can be lowered through the power house floor across the inlet port to the turbine, and by a further series of gates in the draught tube.

The concrete structure itself should not require any maintenance.

6.6. SURVEY OF POTENTIAL SITES FOR HRS DEVICE OFF THE OUTER HEBRIDES

6.6.1 GENERAL

Two site studies have been carried out in the past year which have an important bearing on this particular device. They are -

"Western Isles Marine Fouling and Ecosystem Survey Report" by the Scottish Marine Biological Association, and "Consultants Report on Sites for Seabed Mounted Devices in Outer Hebrides" by the Consultants, for the Wave Energy Steering Committee.

The surveys have provided valuable information on two aspects of the bottom mounted device. Firstly they have focussed on the likely influence of marine growth on the viability of the HRS device, and secondly they have identified the potential sites for devices, which depend on a satisfactory combination of S.W. aspect, contours, and local geomorphology. Some of the results of the survey are set down in the following paragraphs. Specific information on the bottom derives from inspection of the bottom by divers at 57° 9.8' N and 7° 29.2' W off the S.W. Coast of Uist.

a) The bed rock of Lewisian gneiss was uneven and irregular with undulations of up to 1.5 m in height (elsewhere echo-sounding shows much greater irregularities). This indicates that a considerable amount of levelling by crushed rock bedding will be necessary.

b) At 10 to 15 m depth of water the kelp forest (Laminaria Hyperborea) is at its maximum ecological performance in terms of growth rates and standing crop, with an estimated density of 20 to 25 tons per hectare. As depth increases growth reduces considerably due to less light penetration, and at 25 m depth the crop is down to 3 tons/hectare.

Underwater photographs illustrate the dense and sturdy nature of the growths.

The implication is that the kelp could be a serious problem in obstructing flow through the flap valves, preventing their efficient operation and obstructing flow through the turbines. Measures would have to be taken to control it, by physical removal before it enters the flaps or turbines. Whilst the weed itself may not damage a turbine, stones which are frequently attached to the fronds could do so.

c) The construction of an entirely new habitat in the form of the HRS device would provide ideal surfaces for a wide range of marine fouling organisms not encountered in this sea bed survey, such as barnacles, mussels, hydroids and sea-anemones, together with the shallow water and inter-tidal organisms. Sea squirts, whose life cycle requires no light would be able to colonise devices. Seaweed growth would occur on the devices themselves.

The growth of organisms within the device's chambers may not interfere seriously with the performance since water flows are too slow for fluid friction losses to be important. Growth on the flaps and seals and on turbine runners and draught tubes would be more serious. Shut-down and removal of flap gates and runners for cleaning at regular intervals will be necessary and very expensive. Chlorine has been suggested as a possible means of killing off growths, but the Consultants experience of a similar proposal for another job indicates that this idea is impractical. Quantities required would be excessive, and side effects unacceptable.

d) Sedimentation is not expected to impose operational difficulties to devices mounted off the Uists as silts and sands do not occur in significant quantities in regions of heavy seaweed growth. Further investigation is required for the sites off Lewis.

The sea bed from the coastline out to the 70 m depth contour is essentially rock.

Off S.Uist there is a relatively uniform sea bed fall to 20 m depth in about 4 km followed by a steeper slope to 27 m in the next 0.8 km. Off N.Uist, Harris and Lewis, the sea bed is more irregular and this limits the suitable sites.

e) There is an adequate number of sheltered beaches on which transmission cables can be conveniently landed.

f) The range of spring tides at the coast varies from 3.6 to 4.0 metres along the coast.

g) Present shipping lanes are to the West of St.Kilda or to the East of the Hebrides and therefore will not interfere with the devices in the considered sites.

6.6.2 AVAILABLE RESOURCES

In assessing the number and location of available sites the Consultants were guided primarily by the known directional distribution of incident energy, the directional aspect of the coast, and the presence of shelter from particular directions, which could seriously cut down the resource.

All suitable sites have been identified and classified according to their potential available wave power which varies because of the factors mentioned above.

Class I sites have a potential wave power of 24 MW/kM of device
" II " " " " " " 20 MW/kM " "
" III " " " " " " 16 MW/kM " "
(See also Chapter 13.)

With the devices located in 15 m depth of water the total available suitable sites off the Western Isles have been identified as:

Class I	15.7	kM
Class II	20.6	kM
Class III	38.9	kM
Total	75.2	kM

If the devices were located in 25 m depth of water the total available suitable sites would be:

Class I	24.6	kM
Class II	31.1	kM
Class III	59.1	kM
Total	114.8	kM

6.7 ASSESSMENT

6.7.1 GENERAL

Over the past year development work on the device has been relatively limited, but there has been clarification and new information in a number of areas and the Consultants believe they are now getting a clearer and more reliable picture of the likely potential and limitations of this particular device. One major area of uncertainty remains, which is the potential influence of weed, which could in itself be an insuperable problem.

6.7.2 COST EFFECTIVENESS

In the 1977 assessment it was noted that this device was exceptionally massive and that a number of options should be examined to explore the potential for cost saving on structure. A start has been made on this and the 1978 design is marginally smaller and lighter, but the indications are that there are going to be no very large reductions in structure costs. Options involving significantly reduced internal partitioning have not proved very effective, and a year's work leaves the current Reference Design looking very similar to last year's. It is concluded that the device remains, and is likely to continue to remain, very large in relation to power produced.

6.7.3 TECHNICAL FEASIBILITY

6.7.3.1 Structure

The structure is a very large but otherwise fairly conventional caisson, and will be constructable without significant problems. Placement offshore will be less conventional in the sense that there may be problems in getting enough weather window. Success would depend on developing techniques for doing a lot of work in a short time, but this must mean low utilisation of expensive construction plant. Design of the structure to resist tow out and placement forces adds significantly to the weight of an already large device.

6.7.3.2 Flap Valves

Marine fouling will be a problem and at this stage the feasibility of this component is not known.

6.7.3.3 Power Offtake

Given that the turbines are not asked to accept quantities of weed and stone, the power offtake is completely practical, and can be designed on the basis of current knowledge. It is accessible and maintainable.

6.7.4 WEED, FLOTSAM, STONE, SILT

If the large quantities of loose weed which regularly appear on the shores of the Hebrides are in part generated in water beyond the device line, then there is a major problem as regards weed. Weed screens which are a feature of cooling water intakes for thermal power stations, are notoriously difficult to engineer and clean.

Flotsam could be troublesome if damaging the flap valves. Whilst a protective screen can be provided, the problem must be to prevent it clogging up with otherwise harmless weed and small rubbish.

Small stone and silt appears perhaps the least worrying of the ingestion problems, but again, no work has been done.

Overall ingestion of solids stands out as the major question mark over the feasibility of the Rectifier.

6.7.5 RESOURCE

The survey of potential sites off the Hebrides has been a valuable exercise in illuminating the value of this fixed bottom device. Potential sites are limited by the alignment of contours with respect to the direction of the most energetic seas and with respect to shadow from headlands.

The annual productivity of the device is considered in Chapter 13.

Peak plant capacity of 3 MW per 100 m device is provided. Hence the peak productivity of a total HRS Hebridian Scheme would be between 2.25 GW and 3.45 GW depending on depth of water sited in. This in itself may be considered to be a severe limitation on the potential of the device.

6.7.6 ENVIRONMENT

This is considered briefly in Chapter 17. Overall the device will be visually unobtrusive, and whilst the effect on marine life needs careful researching, there is no reason to think that there will be any serious adverse effect.

6.8 FUTURE RESEARCH

The following areas need clarification:

a) The problem of solids ingestion. No one yet knows how much free floating weed passes the 20 m contour.

b) The likely effect of growing weed on the operation of the flap valves.

c) There is probably some modest improvement still to be gained in the cost effectiveness of the device structural configuration, and if this is to be realised, further testing of alternative configurations will be required.

d) There is as yet no information on performance in random seas or in inclined seas. This information would come from wide tank testing.

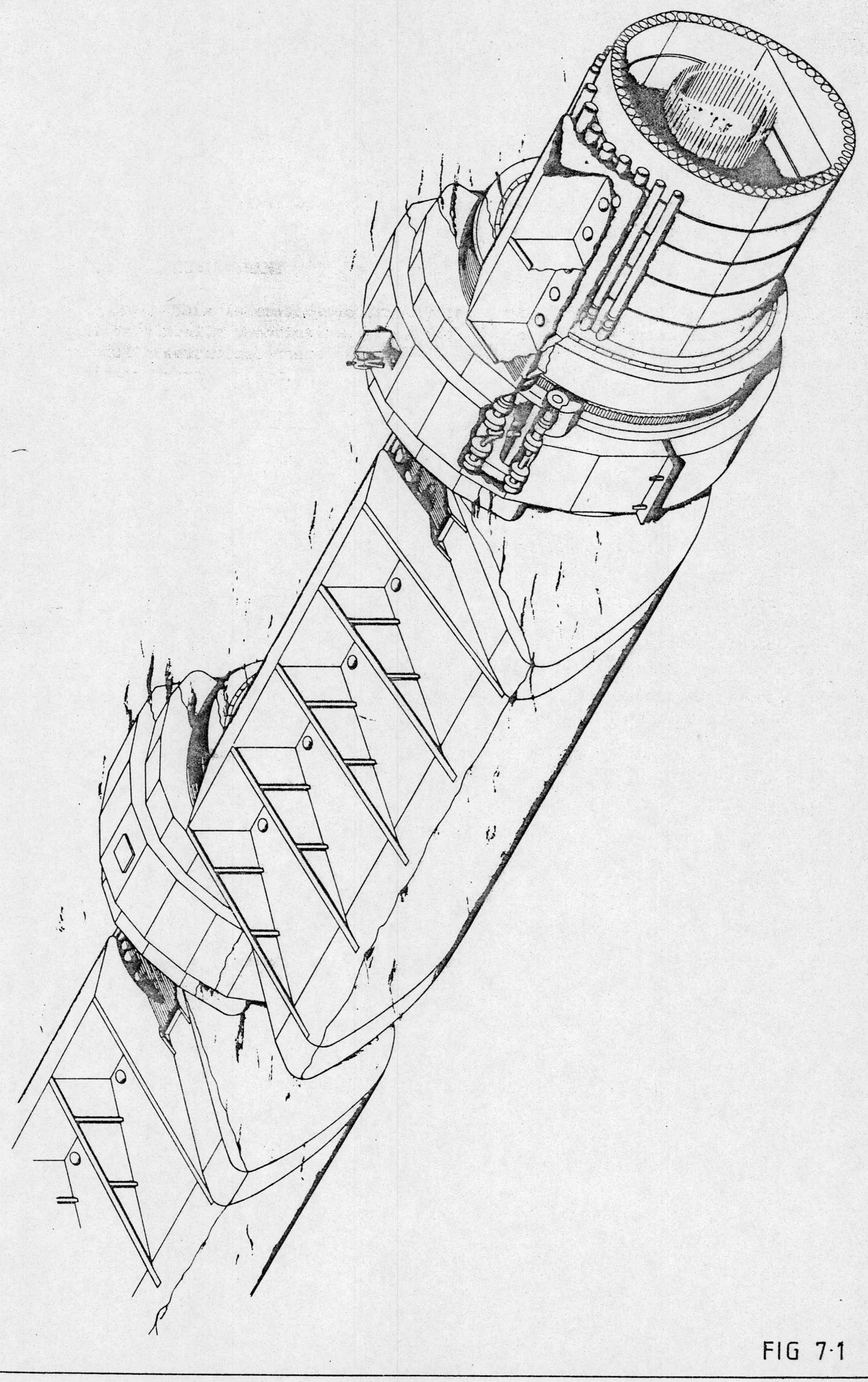


FIG 7·1

CHAPTER 7.0 - SALTER DUCK (DEVICE TEAMS - SEA ENERGY ASSOCIATES AND EDINBURGH UNIVERSITY)

7.1 DESCRIPTION OF THE CONCEPT

7.1.1 THE ORIGINAL IDEA

The essentials of the Device are a very long floating cylindrical spine and a cam, or duck, which is located on, and rotates around, the spine. Power is generated from the relative movement of the duck and spine. The profile of the front face of the duck is carefully chosen such that for small rotations of the Device the displacements match the water particle displacements in a given wave. The back face is circular and therefore does not displace any water behind the duck. The spine provides a fixed reference frame. Theoretically, if the restraining torques applied to the ducks are correctly chosen, the incident wave is unable to distinguish between duck and adjacent water, and transfers all of its energy to the duck. No energy is transmitted from the circular back face and in this case the duck is a perfect absorber.

7.1.2 CURRENT FORMS OF THE DEVICE

The Teams developing this concept have not made a final decision on several important features of their device. Indeed both Teams' ideas are very fluid and several alternatives are being considered simultaneously. For this reason much of this chapter concerns the range of options available, and the general problems involved, rather than specific engineering appraisal of one system. The two main areas for study are the spine, and the power take-off and duck location system.

7.1.2.1 Spines

The original concept for the spine was a continuous backbone which would obtain its stability from the self-cancelling of wave effects over several wave crest lengths. A spine has the additional advantages of allowing the power from many ducks to be collected together into a convenient level for the generation of electricity, and of avoiding the problems of close mooring that are associated with alternative systems with many small devices. For efficient operation the duck clearly requires a stable frame of reference and the movement of the spine should be minimised. SEA have studied the problem of the stresses in long spines of various diameters and have come to the following conclusions (reference - SEA Second Interim Report).

a) Very large diameter spines (probably more than 30m) are required to provide a suitable frame of reference which is both strong enough and sufficiently rigid.

b) Very small diameter spines (certainly less than 2m) would be sufficiently compliant to survive, but these would not provide a suitable frame of reference for the ducks.

c) Spines of intermediate diameters are unacceptable both from the point of view of rigidity and strength. No structural material can accommodate the required strains.

The SEA Team are therefore investigating two alternatives. SEA would prefer to be able to use short spines perhaps 250m long but this option is not favoured by the Edinburgh Team. This spine concept relies for success on the correlation between wave height, wave length and crest length. Small waves tend to have short wave lengths and short crest lengths. Hence in calm and moderate seas the imposed spine forces would cancel out over a comparatively short length, hopefully within the spine length, the spine would remain relatively stationary and the stresses in the spine would be acceptable. High waves in storm seas would be associated with much longer wave and crest lengths. The imposed forces would not cancel and the spine would move with the waves, thus keeping the stresses at an acceptable level. However the correlation between wave height and crest length is very weak, and the Team recognise that this concept is very speculative. A further refinement proposed is a curved spine which would increase the apparent compliance of the spine to the most damaging crest lengths and wave heights. By this means the SEA Team hope to be able to increase the maximum spine length. However, the increase in movement may automatically imply a degradation of device performance in moderate seas, and the rolling of the spine will increase the peak angular excursion to be accommodated by the power take-off and moorings.

Both Teams are considering a second alternative, articulated spines with discrete joints or hinges. For maximum device efficiency these need to be very stiff in calm and moderate seas, but flexible in storm seas for survival.

The various spine options are set out in Figure 7.2.

7.1.2.2 The Power Take-off and Duck Location System

The detailed engineering considerations regarding the alternative systems are discussed later.

Conceptually the system is simple. A bearing is required to allow the duck to rotate on the spine, and a resisting torque has to be applied for power extraction. From the many alternatives possible (see reference TAG 6, SEA reports and the Consultants Report 1977) the Teams have narrowed down their favoured options to the following.

<u>Bearings</u>

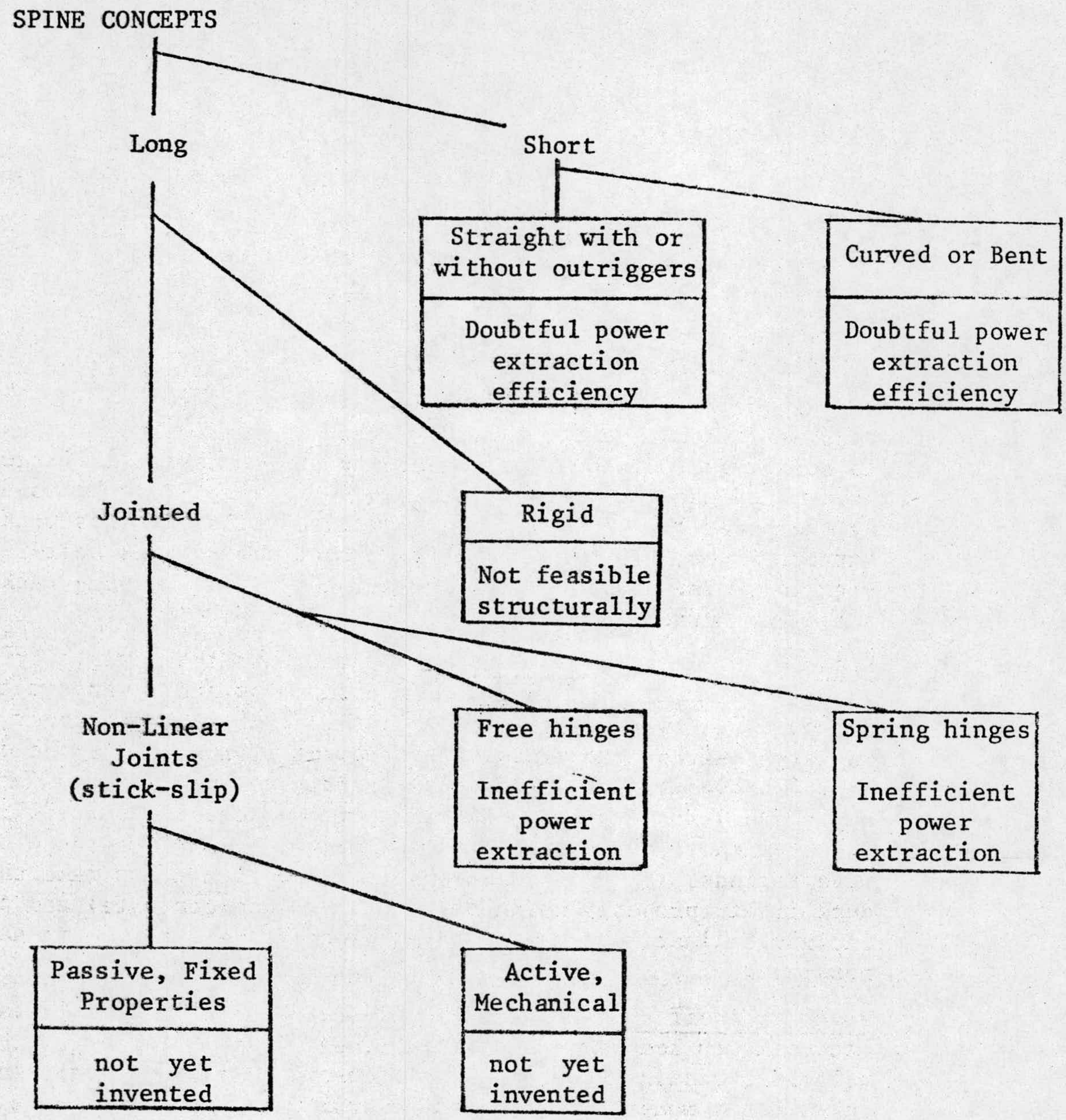
a) Hydraulic
b) Low friction surfaces (e.g. PTFE)
c) Solid Rubber Wheels
d) Rubber tyres

<u>Power Take-off</u>

a) Belts driving hydraulic pumps
b) Gears driving hydraulic pumps
c) Spline pumps

The nature of the bearing has no effect on the behaviour of the duck, but the characteristics of the power take-off are fairly critical. These characteristics which are determined both by the characteristics of the plant, and those of the control system, may in general be either,

a) Linear
b) Stepped linear
c) Torque limited (linear, then at constant torque above a certain speed)
d) Power limited
e) Other non-linear characteristics.



SPINE CONCEPTS

Figure 7.2

In principle, the properties may also be held constant, or varied to suit the prevailing sea.

7.1.3 BEHAVIOUR OF THE DEVICE

Firstly it must be stated that no tests of a complete duck on spine system have yet been completed, with the exception of the Draycot reservoir experiments. Hence the following discussion is based on results inferred from narrow tank testing at Edinburgh.

Duck behaviour depends on the following:

a) Duck shape - In fact the efficiency of the device has been found not to be overly sensitive to the profile of the front of the duck. More importantly, the profile of the top of the duck affects the performance in large waves. This can be used to advantage to reduce the efficiency of the device in large waves, and hence reduce the limit of power needed to be handled by the power take-off. As the duck also rolls onto its back in extreme waves, it is also important that the duck is self-righting. These characteristics are achieved by giving ducks a humped or curved top surface.

b) Rotational inertia of the duck - To maintain the correct trim in calm water the duck needs sufficient mass and lever arm to overcome its own buoyancy. Within this limitation, however, it is very desirable to minimise the moment of inertia of the duck. This improves the ratio of total inertia to hydrodynamic damping, and thereby improves the bandwidth of the device; that is, it improves the responsiveness of the device to a wider range of wave periods. For these reasons it is desirable to make the duck light (probably using steel rather than concrete) and to include ballast in a concentrated block as near to the spine as possible.

c) Power take-off characteristics - Ideally the torque would vary according to some pre-determined linear function of angular velocity, acceleration and displacement for maximum efficiency in calm and moderate seas, and according to some other function when required to shed excessive power in storm seas.

For the first of these modes of operation the Edinburgh Team have shown that it is only necessary to make the torque proportional to velocity for the results to be very near the optimum, as long as the actual duck inertia is kept small (see b above). For the more extreme seas the Team have investigated the effect of imposing a limit on the maximum torque. For this case it is found that as the torque limit is brought down, the maximum power absorbed by the duck decreases, and the maximum angular velocity increases. There are many possibilities apart from the above which could also be investigated. In particular, no real system will have truly linear characteristics even in fairly calm seas, and it will in future be necessary to relate the model power take-off characteristics to those which can be achieved in practice.

d) Spine compliance - It is arguable that it is more correct to estimate the influence of spine movement on duck performance from narrow tank tests than it is to estimate design parameters for the spine itself. Spine compliance, as used here, is a measure of the resistance of the spine itself, to movement in either surge or heave. It seems that optimum duck performance is not achieved from fully rigid restraint. Two nearly equal optima appear, one with no heave compliance and a very small surge compliance, and the second with infinite heave compliance and the same small surge compliance. For intermediate values of heave compliance the efficiencies are low, or zero.

Hence, the spine can either be made stiff for bending in both directions, or can be stiff for bending in a horizontal plane with hinges to allow vertical bending.

In the second case the spine moves considerably, and the hydrodynamic interaction with the waves is rather different to that suggested for the initial concept. This suggests that revised duck geometries may be found most suitable in this case.

7.2 REFERENCE DESIGN - SEA TEAM

7.2.1 GENERAL

The Reference Design is based on drawings prepared for the SEA Second Interim report. Calculations relating specifically to this design are not available, and the Consultants have not been able to make any more than the most elementary checks on the sizes and strengths of components. Where SEA are still investigating more than one option the Consultants have taken the currently most favoured option. A modified spine joint has been included to bring the design up to date with the latest ideas of the Team.

7.2.2 SPINE

The spine is constructed of 2m long precast concrete rings separated by special rubber pads, and prestressed together with Parafil cables. The rubber pads are ommitted between those rings which support the power take-off system for each duck.

The concrete is reinforced circumferentially but has only nominal reinforcement longitudinally. The rubber pads are of novel design. Each contains a flat air filled space, the two sides of which are brought into contact by the compression from the Parafil cables. In calm and moderate seas the bending moment in the spine is insufficient to overcome this pre-compression, and the pad acts as a very stiff interface between adjacent concrete rings. For more severe conditions the moment overcomes the prestress and the pad opens by up to one inch. The close spacing of the joints gives the spine an apparently distributed yielding characteristic and comparatively severe curvatures can be accommodated. Even within the length of each duck the spine flexes significantly. Joint rotation occurs about the side of the spine which is bending compression, (not the centreline), and an appreciable change in the length of the spine occurs. This results in considerable increase in the stress in the Parafil cables. The Consultants do not believe the spine concept to be in any sense proven, and note the following problem areas.

a) Strain capacity too little - Making allowance for the length of spine without joints adjacent to the power take-off at each end of each duck. the extension corresponds to an average extreme fibre strain of about 1% when all the joints are fully open.

The moment capacity of the section before the spine opens is somewhat less than 18000 tons x 5.5m. The Consultants have made approximate calculations of how the strain in a spine will vary according to the wave height H, and crest length Lc, assuming some defined maximum moment capacity which is independent of strain. A fully compliant spine will follow the profile of the wave. If moment capacity is added the spine can remain rigid up to a certain wave height, and even thereafter will be straighter than the wave profile. The vertical axis of the plot in Figure 7.3 defines the strain that will occur, assuming constant moment capacity in different sea conditions. For a safe spine this therefore defines the strain capacity for which the spine must be designed.

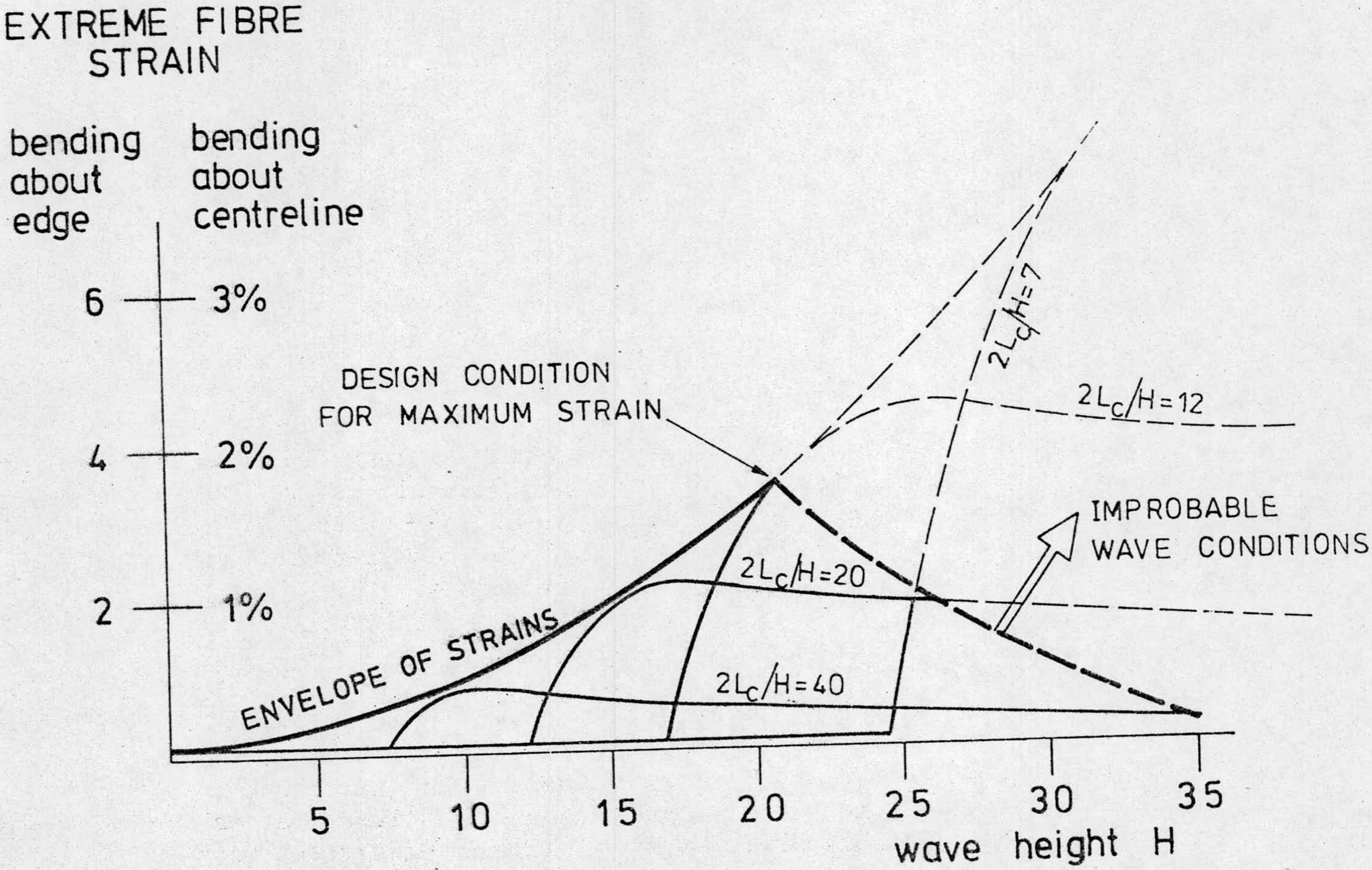
It appears that in the worst sea conditions the strain capacity required will be considerably greater than the SEA joint design allows, particularly for the case of heave motion for steep waves travelling along the spine (see Figure 7.3).

b) All strain concentrates at a single joint - Figure 7.3 does not reveal the full extent of the problem. The concept requires that the strain be divided fairly uniformly between the required number of joints. The first joint to open will be at the point of maximum moment. It is essential that the moment carried by that joint should increase as it separates, otherwise none of the other joints will have reason to open. If this happens, all of the strain will be concentrated at the one joint. This is known to occur in all structures with unbonded prestressing tendons, as in this case. At the breaking joint no local increase in the stress in the tendon will occur. This will be aggravated by the loss in stiffness due to reduction in lever arm as the tendons move across the section. The latter effect could cause buckling of the spine. Diaphragms would prevent this, but would cause an unacceptable fretting problem due to the movement of the cables, and the angular change in cable direction at the diaphragm.

It seems that the present Design does not include adequate provision to prevent the spine flexing at one joint only.

c) Duty imposed on the rubber - A third problem concerns the detailing of the rubber pot bearings. The outside of the rubber pad is required both to seal the inner space filled with gas under high pressure when the joint is closed, but must also allow the required joint opening. Rotation of the spine on opening will transfer all of the prestress load onto a very few pots diametrically opposite to the point of maximum opening. The combination of duties imposed on the rubber, in respect of extreme stress, cyclic loading, and edge effects will be extremely, and perhaps prohibitively, severe.

d) Longitudinal Shock-loads on closing - Particularly for a long spine, the ends will be moving in and out over many metres. The inertia of hundreds of metres of spine, even at low speed, is such that when the last joint closes there will be an impact generated stress wave sent down the spine. The Team do not consider this to be a major problem, but until this is proven by testing or analysis the Consultants would see this as a very important point of concern.



$$\text{PEAK STRAIN} = 2\pi^2 \frac{H}{4L_c^2}\,\frac{D}{2}\left[1 - \frac{4\pi^2 M_L}{H c_f \rho g D (4L_c^2)}\right] (\times 2 \text{ if bending about an edge})$$

M_L = maximum moment capacity : $18000^T \times 5{\cdot}5$ m (S.E.A. Reference Design)

c_f = force coefficient taken as $0{\cdot}5$

L_c = crest length

H = wave height

D = diameter $11{\cdot}0$ m

PEAK STRAIN IN SPINE AS A FUNCTION OF WAVE CONDITIONS

FIGURE 7·3

e) Dynamic behaviour - In any structure built to operate in the range between fully rigid operation and fully compliant, dynamic effects must be expected to be very important; and these can only increase the problems anticipated in simple quasi-static analysis.

f) Fatigue - Structures with self limiting force characteristics are more vulnerable to fatigue damage due to the high ratio of operating load to design load.

g) Sealing - Finally, there is a problem in making the spine water-tight, particularly at the joints, and in providing adequate damage stability in the event of leakage at one of the many joints. SEA have proposed to incorporate deflated butyl bags in each spine. These can be inflated at the appropriate time and are apparently very effective in filling the whole of an irregular space. Clearly the prestressing cables present a problem in this context. Whether such bags would be able to seal the back of the flexing joints without wear or fatigue is not known and the sealing of the joints must be regarded as an unsolved problem. Clearly a length of spine could not rely solely on the integrity of such a bag in an inaccessible position, and watertight bulk heads might be necessary in some of the precast spine rings. As stated previously this is not possible because of the fretting problem, especially if a watertight seal were required.

7.2.3 DUCKS

These are constructed in steel with a ballast box near the inside face. The duck is not continued around the back of the spine, an innovation that will considerably simplify manufacture and installation. The top of the duck is shown with an open topped tray with drainage holes. This shape has not yet been tested by the Team but it is hoped that in extreme waves the tray will be flooded and the angular motion of the device reduced. Trim of the duck is easily adjusted by ballasting the end tanks. Manufacture of the ducks themselves will not present any major problems.

7.2.4 DUCK LOCATION BEARINGS

The main bearing is situated between the torque ring and the bearing deck and is 12m diameter. There is a rubber journal bearing using the low pressure hydrodynamic principle of the "cutless' type. Normal working pressure will be around 10psi with survival pressures up to 100psi. The Team believe that the location bearing is the most difficult problem in their design. The bearing relies on the maintenance, by the natural action of the bearing of a layer of sea water between the two bearing surfaces. If the hydraulic bearings could be made to work the advantages would be as follows -

a) Simplicity - no moving parts
b) Long life with no maintenance required
c) Tolerance to spine movement
d) Ample reserve strength for extreme duck loads
e) Low friction losses
f) Sea water lubrication

This proposal is so novel that a proper appraisal is not possible until a considerable amount of research has been completed. The only related systems are the water skates used for slow movement of very heavy loads, and the rubber hydraulic bearings now frequently used for ships' propellers. One disadvantage of such bearings, seizure after prolonged periods at rests, should not be a problem for the continuously nodding duck. The principle problems are likely to be life, replacement, detection of defects and consequences of premature failure.

7.2.5 POWER TAKE-OFF - GENERAL

Very briefly, interaction between wave motion and the duck beak produces cyclic angular movements at two torque rings about a relatively stable spine. The spine in turn is provided with sets of low speed hydraulic pumps driven by gearing on the torque ring. The pumps thus produce a pulsating flow of high pressure hydraulic fluid which, after smoothing both by virtue of the diversity of wave energy aided by somewhat limited hydraulic accumulation, is used to drive a high speed oil turbine.

7.2.6 GEARS

Conventional gears require accurate location of the gear axis relative to the track. The Team have proposed a gear system which is intended to overcome this problem. The sprockets are made of rubber and have a rolling profile. The track has polymer inset teeth. It is claimed that there is evidence from other applications that this type of gear system would have adequate strength and indefinite life. Again the principle problems are likely to be associated with the unusual location of the gears, i.e.

a) Tolerances to very large movements, 6 inches parallel to the gear axis, considerable movement radial to the spine due to movement of the duck and manufacturing tolerances, and about 0.8° of rotation from the same source.

b) Detection of faults, consequences of local failure and replacement.

c) Working environment. Gears designed for long life, operating in salt water with no maintenance, have not yet proved a practical possibility.

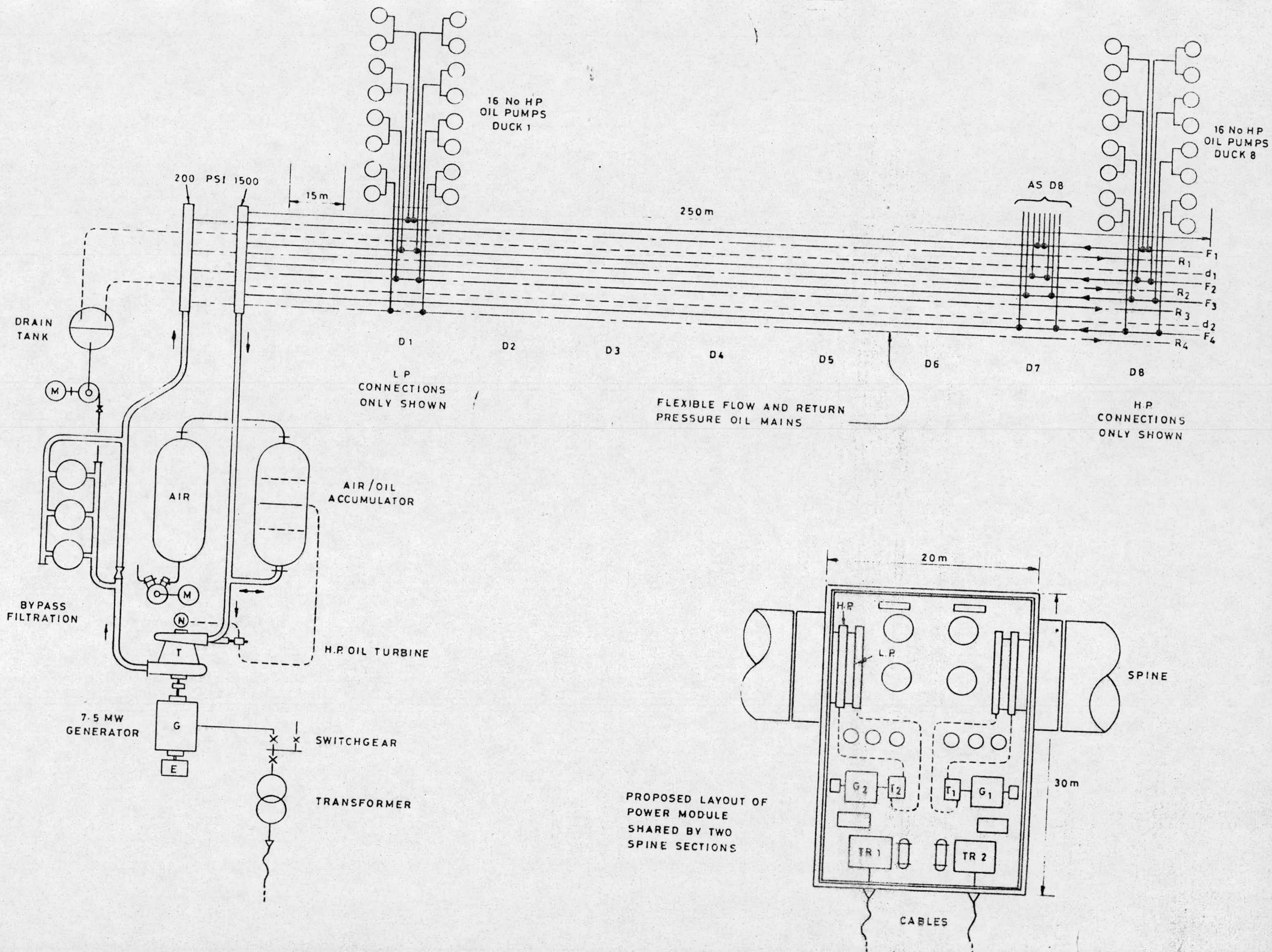
7.2.7 HYDRAULIC PUMPS

The type of hydraulic pump proposed is a radial piston multi-cam design, in which the piston assembly rotates with the drive shaft, adapted from high torque low speed hydraulic motors, and driven by the pinion. These deliver hydraulic fluid at a pressure of up to $14 MNM^{-2}$. The pumps reverse every half wave period, i.e. between four and six million reversals annually. The pumps normally operate at low speed (50 - 80 rpm) but must be capable of withstanding three times this order of speed under storm conditions. Pump reversal will entail provision of some form of reliable flow reversing valves. If possible, a pump design requiring only suction and discharge valves would be advisable.

The pumps proposed are modified Hägglund motors having a normal maximum working speed of 50 rpm and a maximum working pressure of 210 bar. These motors will be required to be designed and developed specifically to operate well as pumps in this application of varying speed and reversal of

FIGURE 7.4

HYDRAULIC SYSTEM DIAGRAM – SINGLE SPINE – 8 DUCKS SEA DEVICE



rotation, and could be subjected to thermal shock.

The above type of pump adapted from the standard motor design does not have the capability of varying its output flow except by use of the 'half flow' facility. However, other types of radial piston high-torque low-speed motor have been more successfully adapted to operate as pumps, namely the radial piston crankshaft and eccentric design which uses a hydrostatically balanced eccentric which in turn drives hydrostatically balanced pistons in which the piston assembly does not rotate with the drive shaft.

This latter type of pump appears to suffer less from thermal shock and can have the facility of varying the output flow in proportion to the pumping pressure, so that power shedding could be more controllable. To produce a pump of the size required and having a proven reliability the manufacturer would have to carry out a development programme.

7.2.8 PUMPS AND THE CONTROL SYSTEM

The hydraulic oil pumps are subject to peak momentary wave power inputs. Consideration of sea conditions off the Outer Hebrides indicates that if each duck has 16 h.p. pumps, each should have a rating equivalent to 400 KW. A further complication arises since performance of the duck in differing sea states requires adjustment of the torque reaction. Positive displacement pumps have a relatively constant torque characteristic. The requirements of the duck can only be met by varying the number of pumps or even pump pistons in action under particular sea conditions. Pumps of the swash plate type, whilst varying the stroke and work done per cycle, would not meet variable torque requirements. An alternative would involve manipulation of the pump delivery valves so that driving torque could be adjusted as required in steps. Arrangements for doing this would be complicated. The possibility of delivery pressure reduction has not been overlooked. It would not, however, suit the fluid power recovery turbine at the other end of the cycle.

7.2.9 DRIVE DISCONNECTION CLUTCH

From the point of view of servicing, maintenance and the protection of the whole power generation system, it is considered essential that a de-clutching facility be built into the drive to the pumps. This would allow any one pump to be taken out of service for maintenance purposes, and could ensure that the failure of one pump would be less likely to cause damage to the ring gear and pinion assembly by overloading, or to the other pumps and rectifier valves in the system by drastic contamination of the hydraulic fluid and consequent possible filter failure.

7.2.10 FILTRATION

The high pressure oil pumps and foreseeably, the turbine will require very clean hydraulic fluid. Full flow filtration, although desirable, would be impracticable and inordinately expensive. Filtration at the offtake points where low pressure oil is returned to the pumps would be ideal but would hardly be practicable in the exposed location, since routine filter changing will be necessary. A rather poor compromise would be to provide proportionate flow filtration through a bypass filter circuit located in the power module. As an aim, filtration should cover 25 per cent of the flow and achieve particle removal down to 25 microns.

7.2.11 MAINS COOLING, ETC.

In the Lanchester design the space existing between the elliptical spine and the circular duck torque rings and their support collars, has been used to accommodate a number of flow and return oil mains. Since the spine is necessarily constructed as a flexible central core, the pressure oil mains must have equivalent flexibility. They are also exposed to sea water, and whilst this inherently solves any oil cooling problem, it might result in too great an increase in the oil viscosity. The pipe section required for oil transmission with low friction loss will be considerable. A number of pipes connected in parallel may be necessary.

7.2.12 ACCUMULATORS AND TURBINES

Before applying the pressure oil flow to drive a high-speed specially designed oil turbine, it would be advisable to achieve further smoothing and local pressure regulation by providing an oil/air accumulator with sufficient additional air volume to keep the pressure variation within small limits. Having done this it is estimated that under the maximum sea conditions considered, the power available to the hydraulic turbine might be about 10.75 MW. In passing through the turbine, the oil would be reduced in pressure so that it is discharged at about 200 p.s.i. (1.4 MNm^{-2}). The back pressure is required to avoid cavitation in the pump suction passages, and also to meet head loss in the return lines.

The high speed - probably 3000rpm - single or two stage Francis type oil turbines would have an output of 12750 h.p. but would be quite small dimensionally. Manufacture of the turbine runners would be difficult. Consideration would be given to a stainless steel casting produced by the lost wax process. Power output would be regulated not by small guide vanes, as in conventional water turbine practice, but by some form of sensitive spear valve subject to over-riding speed control.

7.2.13 HYDRAULIC FLUID

The type of hydraulic fluid used will have to be carefully chosen for its lubricating and viscosity characteristics, but a number of the current suitably sized hydraulic motors adapted to operate as pumps have been proved suitable for use with water based fluids such as the 5% oil and 95% water fluid, invert fluids, or water glycol. However, for using these fluids, the maximum operating pressure must be reduced due to the inferior lubricating properties, but pressures up to 10 MNm^{-2} will still give an acceptable unit life.

7.2.14 ALTERNATOR

The 3000rpm alternator directly coupled to the turbine would have a nominal rating of 9 MW. The power plant would be duplicated within the power module so that it handles the output of two spine units with a total of 16 ducks. The generators would not be operated in parallel since the proportional adjustment of generated voltage according to the electrical output must apply individually. The module would therefore include power and auxiliary switchgear and set-up transformers leading to twin 22 KV flexible submarine cables to transmit the total of 18 MW to a pair of rectiformer units in the nearby converter station.

7.2.15 SUMMARY OF PROBLEMS FOR THE POWER TAKE-OFF

The preceding description summarises the means of converting wave energy to an electrical output. Areas in which design needs detailed investigation and development have been mentioned. It is equally important to be aware of a number of the practical difficulties. The more obvious ones include reversing high pressure pumps with an attendant fatigue failure problem; the risk of hydraulic fluid leakage with so many high pressure oil mains exposed to the sea; the problem of accommodating very large strains in the high pressure mains; location of the pumps is unsuitable for complicated pump control equipment; none of the duck mounted equipment is really accessible and this is contrary to the engineering principle of achieving high annual availability by regular plant maintenance. This should be a guiding principle when further developing power plant for the SEA system.

7.2.16 CONSTRUCTION METHOD

The basic spine modules are precast concrete rings 2m long, 11m diameter (flattened at the top), and weighing about 120 tonnes each. The spine is assembled in lengths of approximately 250m in a dry dock. Rubber pads are included in the joints between the modules except for the 5 units at the 12 positions of the power take-offs. The latter are rigidly connected with a grouted joint. The whole 250m length is prestressed together in one operation, plus a power house at the appropriate end. Quoting from the SEA report:- "After the flooding of the dry dock, individual lengths of the spine would be floated into calm water where the individual 250m lengths would be connected together by divers into an appropriate final length. The lower clamp ring will first of all be positioned round the spine from a floating barge at appropriate spacings along the spine and the upper clamp ring complete with hydraulic pumps and assembled drive wheels and torque ring will then be lowered onto the spine to connect with the lower clamp ring by a floating crane. A man inside the upper power unit housing tightens the bolts into the pre-tapped holes in the upper flanges of the power unit lower floatation clamp. Between the spine and the power unit fabrications will be a 50mm rubber waffle sheet. This work could, of course, be performed in dry dock given appropriate craneage and a 14m float-out capacity. The power unit housings are assembled complete with hydraulic stub pipes, hydraulic pumps, torque ring and any electronic control gear. After assembly the stub flanges of the pipes and appropriate power and control cables are connected between the power unit housings by fitters assisted by floating cranes. Slave floatation devices will be put round the spine at this stage to keep the top 2m of spine out of the water to facilitate this work.

After everything is connected and its integrity proved, the ducks themselves are offered to the torque rings with the assistance of a floating crane and clamped to the ring against driving faces with clamping cables. The duck itself has three conditions of floatation. The first is a working condition in which the nose of the duck is the optimum height out of the water when attached to the spine for efficient working. The second is the assembly condition where some floatation tanks inside the duck are filled with water to give the correct natural approach of a duck to the torque ring during assembly and the third is a fail-safe condition in which the duck is completely flooded except for integral floatation tanks and will hang below the spine but with an absolute minimum of negative buoyancy so that the overall buoyancy of the complete device is not changed much".

The spine would be towed in 1 kilometre lengths to its final location, where the moorings and power take-off cable would be attached.

7.2.17 MAINTENANCE

It will be possible in favourable weather for a man to enter the upper power unit housing on location. However, only minor maintenance will be possible in this way and the Team are proposing that the whole 1 kilometre device should be disconnected and towed back to a calm water site once every few years. At this site the ducks' power packs, and hydraulic mains if necessary, would all be taken off and replaced. This operation is planned to take 7 weeks from disconnection to recommissioning.

7.3 EDINBURGH TEAM - IDEAS FOR FULL SCALE DEVICES

7.3.1 GENERAL

This Team have not so far been required to prepare a Reference Design, but they have taken a very active interest in proposing new ideas for a full scale system. The drawings in this report are no more than a presentation of these ideas, and do not constitute a full Reference Design.

7.3.2 SPINE

The Team wish to take advantage of their recent discovery that ducks do not need heave (vertical) restraint. The spine consists of short sections, one section for each duck. The joints between these sections are intended to give the spine complete freedom to bend about a horizontal axis. Only surge bending is restrained. It is hoped that this will allow optimum use of spine material, and the spine cross-section now begins to look more like an "I" girder with flanges. To simplify the power take-off system, spine joints are included at the end of each duck but not within the length of each duck. The Edinburgh Team consider that the spine will need to accommodate about 3 per cent extreme fibre strain (bending about the centre line). This implies that each joint will have up to 600mm of movement.

At each joint the bending moment is carried by rubber "struts". For normal operating waves the deformations are small enough for the spine to be effectively rigid in surge, and the device is able to work at maximum efficiency. The struts are carefully designed to buckle above a known limiting compression force, and to lift off when the precompression in the struts is exceeded by tensile bending. Hence in more extreme seas the joints allow rotation. The 2m length of the rubber struts is determined largely by the deflection they must accommodate ($\pm$600mm). The area is determined by the loading carried. The maximum stress in the rubber will be less than half that in the concrete, and it will not be possible to generate the whole of the moment capacity of the concrete. The struts must be kept as close as possible to the horizontal centreline of the spine cross-section for the joint to work properly.

The principle problems for this spine concept are as follows.

a) The load carrying capacity of the struts.

b) Sharing of the prestress and bending loads between the spherical bearing and the rubber struts - If the prestress is taken by the spherical bearing in calm water, then the area of concrete behind the bearing will need to be about the same as the flanges of the spine. If the prestress is taken by the rubber struts the load will tend, in time, to creep out of the rubber into the hard bearings. Replacement of the bearings would also need release of the prestress.

c) The maximum deflection - The maximum deflection necessitates a rather long and unwieldy rubber strut unit.

d) Strain concentrations in the rubber - The mean strains in the rubber strut where buckled may appear reasonable, but it will be a very difficult design problem to limit the peak strain concentrations sufficiently to achieve an acceptable fatigue life. This will occur at re-entrant corners and at the junction of the rubber with the end support plates.

e) Shear capacity of the struts for torsion - The spine must remain torsionally stiff and strong even after the struts have buckled. The torsion must be carried by shear in each strut. In general, buckled components have very little stiffness for any mode of loading and it is not clear whether the struts will be able to take this shear force.

f) Providing space for the prestressing cable and hydraulic mains to pass through the spherical bearings.

g) Design of the prestressing 'saddle' at each joint for the prestressing strand - This saddle must be watertight, provide side restraint to deal with the change of tendon direction in a flexing spine, while preventing fretting or fatigue of the tendons.

h) Installation, inspection and maintenance.

7.3.3 DUCK

The form of construction is similar to that proposed by SEA, but the Edinburgh Team are using a profile for the top of the duck which has been used in their tank tests.

7.3.4 DUCK LOCATION AND POWER TAKE-OFF BRACELETS

7.3.4.1 Description

This concept is the key to the Team's proposals for maintenance of the duck location and power systems. Each duck has ten bracelets attached to it by means of friction clamps attached to the bogies. These grip a fin on the inside face of the duck. Each bracelet consists of four bogies of twelve wheels with pin connections. The bogies are kept apart by a few radial axis rubber tyred wheels. At the ends of the ducks on the duck inside face much larger radial axis wheels on the two front face bogies bear against spine fins to transfer the longitudinal loading from the duck. One of the four bogies per bracelet (a total of ten bogies per duck) houses the power take-off plant.

7.3.4.2 Proposal for Maintenance

Each bracelet can be separately removed leaving the duck adequately supported on the remaining nine - (except for longitudinal support in the case of the end bracelets). This is accomplished by disengaging the hydraulic main connector, the power take-off belt end clamp and the friction clamps by remote control, releasing the preload in the eccentric pin connecting the two top bogies, releasing the back cover flaps (fairings), disengaging one bracelet pin and rolling out the bracelet. A refurbished bracelet can be immediately substituted and the removed bracelet taken to base.

7.3.4.3 Duck Location Bearings

Each bogey has twelve tyres, a total of 480 tyres per duck. The peak load per tyre is about 10 tonnes and they are shown as about 1m in diameter. The tyres are not required to transmit torque which is hoped to greatly reduce the rate of wear. The wheel bearings are conventional sealed units with pressurised greasing to help them survive for long periods under the sea.

7.3.4.4 Power Take-off

Two belts per bracelet are clamped to an end fixing on the underside of the spine. One wraps round the back of the spine between the rubber tyres and onto a pulley on the power take-off bogey, the other wraps round the spine in the other direction to a separate pulley on the same bogey. Each pulley drives two hydraulic pumps (i.e. twenty per duck). To accommodate the maximum excursions the belts wrap round the pulley between 0 and 14 times at either extreme.

7.3.4.5 Hydraulic Equipment on the Bogey

The hydraulic circuit is based on an ingenious idea for separating the hydraulic oil needed for the pumps from the bulk of the fluid, in this case water used in the hydraulic mains. Apart from saving expense, it is hoped to reduce the problem of oil filtration and to isolate the effects of malfunction, contamination or leakage of one defective pump. The pump contains no valves and produces an alternating flow. This is fed to a cylinder and drives a piston over a fixed range which in turn pumps water at the other end. At this stage a pressure reduction or increase would be introduced. The water then passes via a two way poppet valve rectifier into the hydraulic mains. These are connected via a retractable coupling into the hydraulic mains inside the duck which collect power from all ten power bogies.

7.3.5 HYDRAULIC CIRCUIT ON THE DUCK

The high pressure main on each duck may be connected to a hydraulic accumulator. The high and low pressure mains are taken to opposite ends of the duck. At each end the main is connected via a rotating seal to a flexible hose wrapped round a large diameter reel. (The rotating seal may not be necessary, it may be possible to use the torsional flexibility of the main and a rigid connection). The flexible hose is played out by the reel as the duck rotates, and wraps around the spine where it is again connected into a more conventional main.

7.3.6 POWER GENERATION

As for the SEA design the hydraulic mains collect power over a convenient length and are used to drive turbogenerators in special power houses at perhaps 500m intervals.

7.4 CRITICAL ASSESMENT OF TECHNICAL FEASIBILITY

It would not be appropriate for the comments in this section to be restricted to the Reference Designs. The Reference Designs serve only to illustrate the problems for this Device.

As stated in the Consultants 1977 Report, the Device is compact but complex in comparison to the other Devices. The Teams' ideas for a full scale device are in fact very little firmer than last year. It might be thought that the more options that are kept open, the more likely it is that a winning combination might be found. In this case the Consultants now feel this would be falsely optimistic. The problems that the Teams are trying to solve are inherent in the Device Concept, and are not particular to the full scale proposals made so far. They are numerous and some of them are quite formidable. Both Teams are being forced to adopt unpalatable solutions, and this is the reason for so many options being offered. In many cases a choice will have to be made between the lesser of two evils. The problems are not restricted to just one or two key components, but have arisen throughout the device. With many major problems in series the chance of a successful outcome becomes very small.

Finally, the range of ideas proposed has not been matched by the depth of analysis of particular components. This Chapter can only report on problems already recognised. The Consultants are convinced that with so many novel features, detailed study of components will identify further problem areas.

System Appraisal

As systems become more complex it is essential that components are studied in the context of the whole system, particularly in determining operational requirements for the components, and the consequences of failure. The Consultants would be confident that many of the components proposed, if used in isolation, would have a fair chance of being engineered successfully, and would have a reasonable life. However, the system cannot be appraised on this basis. A comprehensive appraisal would need to identify in detail all possible malfunctions and their probabilities of occurence and then predict the consequences of each incident, particularly regarding progressive failures or failures leading to total shut-down. Plant reduncancy and necessary fault detection and isolation equipment would be included in this appraisal, and maintenance schedules prepared. The current proposals are insufficiently detailed to allow a full appraisal on this basis, but the Consultants believe this Device would be likely to fail this test. All previous experience suggests that regular inspection and maintenance of plant is essential, access to key plant is vital, and that only the most rudimentary plant will operate unattended in seawater for long periods.

In these respects the power take-off of the SEA Reference Design looks more feasible than the Edinburgh proposals, but the above comments still apply to the gear system, the spine bearings, the hydraulic bearings and the hydraulic mains.

In conclusion,

a) To date, no credible solution to the spine problem has been proposed.

b) There is no solution which looks in any way encouraging for the power take-off.

c) The device remains a concept attractive in its elegance and fairly economical in its use of materials, but after a further years work the Teams appear to be no closer to finding an engineering interpretation of their ideas.

7.5 FUTURE RESEARCH AND DEVELOPMENT

In the Consultants opinion a breakthrough in concept is required to solve both the spine and power take-off problems, and effort should be concentrated in this direction.

The Teams' proposals do not yet merit a full development programme. If, however, it is felt that further clarification of the proposals is required, or a radically improved system is proposed, then it is essential that the proposals are comprehensive and sufficiently detailed for engineering design calculations to be made.

At this stage no further experimental evidence is needed to define the performance specification for the power take-off. Any gaps in the data can easily and rapidly be filled by tests using the Edinburgh Team's excellent narrow tank facility.

For the spine problem, experimentation is only just beginning. Unfortunately, the tests will need to include power extracting ducks as the ability of spine to provide an effective reactive frame is as important as its ability to survive. Both Teams are very undecided about the best type of spine and its optimum strength and stiffness, so it is essential that the techniques used allow rapid changes in the modelling of spine properties. Again, it is essential that this experimentation is fully integrated with realistic engineering study of achievable system performance.

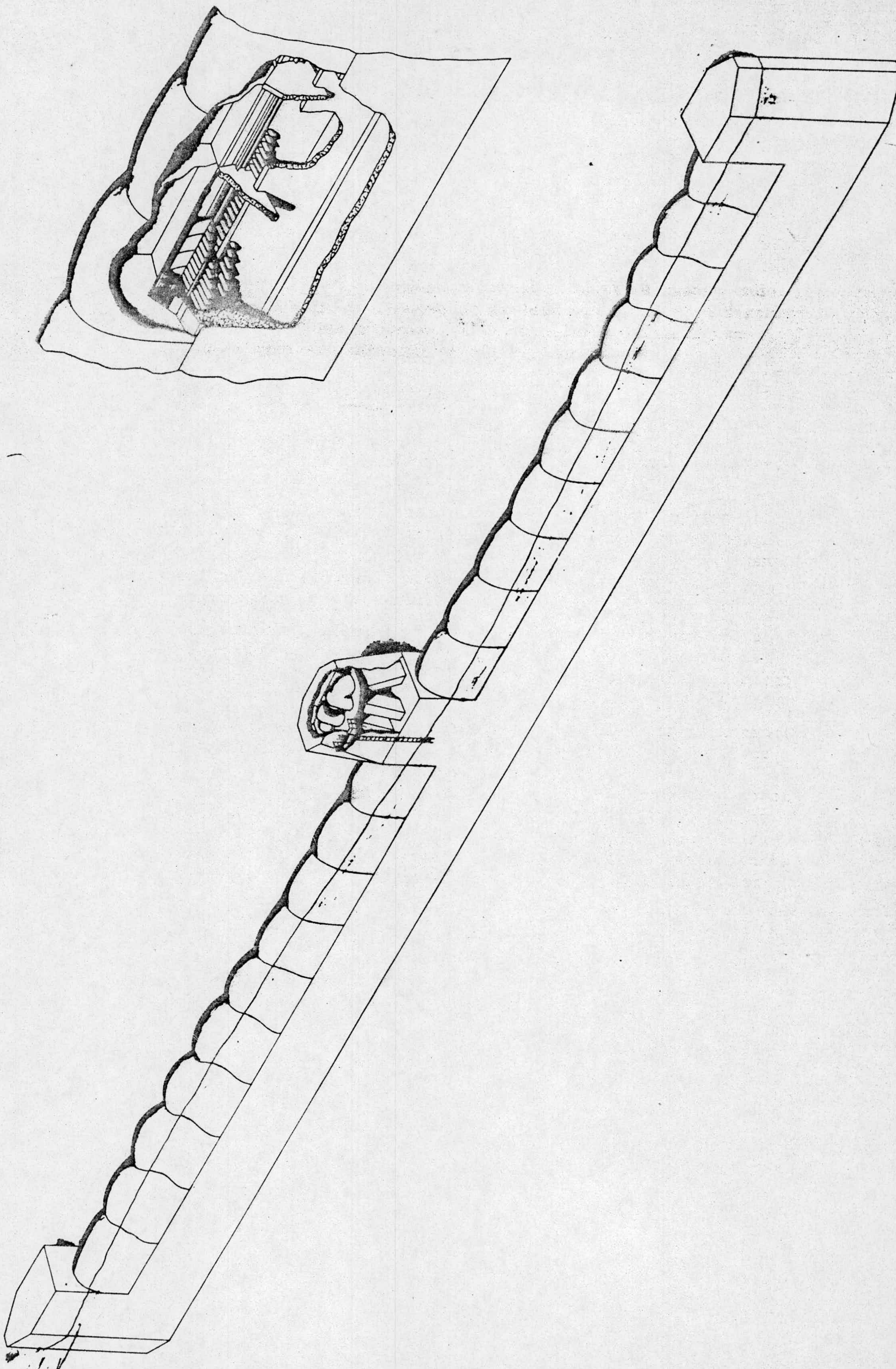


FIG 8·1