

Progress in the Development of a Multi-mode Self-reacting Wave Energy Converter

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Abstract—This paper describes the rationale behind a novel multi-mode self-reacting wave energy converter (WEC) being developed by the New Zealand Wave Energy Technology project (WET-NZ). A computer modeling system used to develop an understanding of the WET-NZ device operation is described. A 2 kW experimental test device, built to demonstrate the concept in real sea conditions and to provide verification data for the model, is reported.

I. INTRODUCTION

Many nations are setting targets for renewable power generation, including New Zealand, and the opportunity to extract energy from ocean waves is immense. However, it is very difficult to harness this energy and convert it into usable electricity and at present there is no successful commercial wave energy converter. Drew et al. [1] reviewed the technology and concluded that despite considerable research and development, the concepts for converting wave energy into electricity “show no signs of converging to a preferred solution”.

The WET-NZ project was established in 2004 to develop technical knowledge within New Zealand to prepare for future marine energy technology adoption. The project focuses on surface wave energy extraction and the major outcome is a demonstrable prototype in real sea conditions. The strategy was firstly to develop a thorough understanding of the source of energy, followed by establishing essential features for the transfer of energy from the source into the device, and from the device into a power-take-off (PTO) system. A review of the status of past and current WEC concepts was carried out after a clear understanding of the wave energy conversion has been grasped by the team. It is surprising that the concept we proposed has not been previously pursued. This paper describes the rationale behind our multi-mode self-reacting wave energy converter (WEC).

The second part of the paper describes a computer modeling system used to develop an understanding of the WET-NZ device operation. Commercial hydrodynamic analysis software is available for examining the interaction of structures with waves and marine currents. However, these general purpose tools have limitations when analyzing the real-time control of power flow through a WEC device. The main areas of limitation are identified and discussed in the paper.

A multiply-connected structure subjected to arbitrary loadings has been modeled using rigid-body dynamics. Newton-Euler formulation was used to derive the governing equations where coupled surge-heave-pitch motions are represented properly.

Various aspects of the numerical model have been verified with small scale laboratory wave tank experiments. However such experiments could not provide adequate information regarding the power absorption capability of the device. A 2 kW peak power experimental device has been built and is being deployed in real sea conditions to provide on-going verification cases for the numerical model.

Finally, a preliminary case study using the computer model to extrapolate the experimental scale device to a commercial scale device is reported.

II. WET-NZ CONCEPT

A. Energy source characteristics

Ocean wave energies are stored and transmitted by oscillating between potential and kinetic forms, and an efficient energy converter should aim at extracting both forms of energy. Two-dimensional linear wave theory was considered to be adequate to establish the concepts of energy transfer. Initially, it was confusing to find that surface wave potential energy has been described in the literature as being an oscillating source at the surface, whereas the kinetic energy is constant and distributed far below the surface [2]. This description relates to the concept that the potential energy is due to gravity (that is only in the vertical direction) and can be seen as the rise and fall of the surface displacement. The kinetic energy is however derived from the movement of the water particles which circulate at a constant speed, the diameter of the circular motion at the surface being the wave height, but exponentially reduces below the surface to the point of being practically negligible at depths below half of the wavelength.

Our physical interpretation is to separate the water particle motion into two components, vertical and horizontal. The circular motion of the water particle simply consists of two oscillating motions, one in the horizontal direction and one in the vertical direction, both oscillate at the same frequency but 90° out-of-phase. From simple harmonic motion, when the

water particle reaches the top or bottom, its vertical velocity and hence vertical kinetic energy is zero, the kinetic energy has been transferred into the potential energy, which subsequently returns to kinetic energy again when the particle moves in the opposite direction. This energy transfer must also occur in the horizontal direction, but it is not clear where the energy can be stored as there is no gravitational potential energy in the horizontal direction. If we considered the pressure differential on a small volume of water we can see that there is a fluctuating horizontal component of the pressure differential. We conclude that the potential energy presented in the wave field is in the form of pressure differential which also oscillates in both the vertical and horizontal directions. Each component is 180° out-of-phase with the corresponding kinetic energy component. So it is not so much a matter of whether kinetic or potential energy is extracted from the waves, but rather which direction is the energy being extracted from. We aim to extract energy in both vertical (heaving mode) and horizontal (pitching or surging mode). Intuitively, the phasing of the two modes is the key to an efficient mechanism of energy extraction.

B. Survivability

An essential design requirement of any WEC is survivability. An overarching facet of the WET-NZ concept has been to design a structure that can survive extreme wave events, whilst efficiently converting moderate waves into useable energy. The WET-NZ device relies on extracting energy from the differential motion of two bodies of different mass: the reactive hull and the active float. The reactive hull should be sufficiently long to extend below the wave motion, whilst the float should be at the surface, where maximal wave energy is located. To survive extreme wave conditions, the device should not be excessively constrained or require reciprocal motions (as other WECs do), so that extreme waves do not generate extreme internal forces. The WET-NZ device is designed to be i) self-reacting (to limit the internal reaction forces), ii) a point absorber (thus small and relatively low risk), and iii) slack-moored in shallow-to-intermediate water depths (25 - 50 m), thus avoiding the nearshore low energy regions and the extremes of deep ocean waves.

C. Structure

The device consists of four major components: an active float, a reactive hull, a power take-off system (PTO) and a mooring system (see Fig. 1). The energy is absorbed through a single pivot rotating motion. This reduces chances of mechanical failure, and the design means that it can extract energy in different modes of motion (heave, surge or pitch). The active float is designed to be able to rotate fully about the pivot and hence does not induce any limiting forces such as hitting end stops. Water lubricated bearings are used at the pivot so that no sealing is required.

Slack moored floating structures are inherently more robust than rigid structures attached to the seabed. However to absorb wave energy, the floating structure must include a reactive

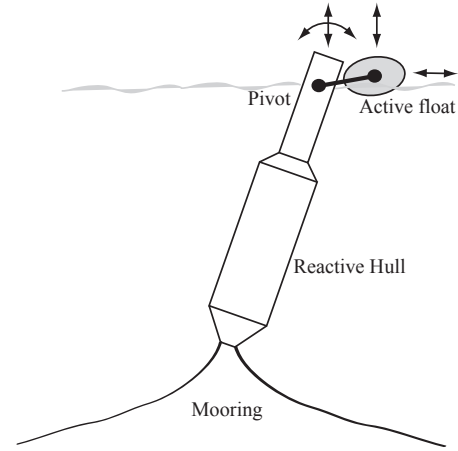


Fig. 1. Schematic diagram of the device

hull which must have adequate volume relative to wavelength, leading to design trade offs. A cost saving feature of the hull is that it can be mostly filled with water — the water mass providing much of the inertial mass required to react against the active member.

The power take-off (PTO) system converts the relative angular motion between the active float and the reactive platform into electricity. Two hydraulic cylinders, connected to the crank arms on each side of the pivot shaft, are used to drive a hydraulic motor to convert a typically slow oscillating motion between 5 to 12 s period to a high speed unidirectional motion of the motor shaft, approximately 360 to 900 rpm. The motor is then coupled to a permanent magnet generator to generate electricity.

III. NUMERICAL MODEL

There are many commercial software packages available for modeling floating structures interacting with ocean waves. Computational fluid mechanics (CFD) packages, which are available with most commercial finite element (FE) suites, may be able to provide the most accurate models of the device. But they are very expensive, and require huge computational power. The end results focus on simulating the fluid motions rather than the response of the float structure to waves forces. A class of numerical models used in designing ships and off-shore structures uses the diffraction theory. These packages do not attempt to model the water particle motions, instead they calculate hydrodynamic parameters which are unique to the geometry of the submerged structure and the wave parameters. An industry standard package for the diffraction theory is WAMIT [3], but other suitable systems are MOSES or ANSYS-AQWA. These packages would be useful in verifying aspects of our device, however, there are many restrictions at the developmental levels, for examples in modeling the non-linear real-time control of the PTO system, or the slip-stick friction model of the pivot bearings.

Modeling the complete system from first principle has some advantages over general commercial solutions [4], [5].

This is our preferred approach since we need to model the PTO system and the wave-dependent active control, which would not be possible or very difficult to represent in other general purpose hydrodynamic software. This method also produces very fast simulation time as the number of degrees-of-freedom of the model is small, hence a large range of parameters space can be considered to explore the device response characteristics.

For the WET-NZ project, a multiply-connected structure subjected to arbitrary loadings has been modeled using 2D rigid-body dynamics. The theory also applies to 3D systems since we use standard vector algebra for the derivation of the model. A modified Newton-Euler formulation was used to derive the governing equations where coupled surge-heave-pitch motions are represented as off-diagonal terms in the governing equations. External forces on the structure include: wave-structure interactions (Froude-Krylov forces [6], and Morison's hydrodynamic forces [7]), and tension-only mooring forces. Internal forces are from friction at various moving parts, and from the hydraulic/electric PTO system. The model can also handle arbitrary adaptive control strategies for the PTO. This feature will be used to achieve quick response to varying wave conditions to maximize power absorption. The 2D shape of a rigid body can be represented by a polygon with a constant thickness, and any forces acting on the surface of the body can be calculated by summation of the forces on each individual edge.

A. Coordinate systems

We use the standard finite-element formulation concept of two coordinate systems: a global coordinate systems that is fixed to the earth; and a local coordinate system that is attached to a rigid body. Each body has its own local coordinate system. Any point G in space can be defined by a global displacement vector $\mathbf{r}_G$. If point G is a point on the i^{th} rigid body, and the i^{th} body has a local coordinate system with the origin at point P which is also attached to the body, then the local coordinate vector of point G is ${}^i\mathbf{r}_G$. Another important vector that is needed in assembling the local system into the global system is the global vector of point G relative to point P , denotes as $\mathbf{r}_{G|P}$, which can be found from

$$\mathbf{r}_{G|P} = [R_i] {}^i\mathbf{r}_G, \quad (1)$$

where

$$[R_i] = \begin{bmatrix} \cos \theta_i & -\sin \theta_i \\ \sin \theta_i & \cos \theta_i \end{bmatrix},$$

and θ_i is the rotation angle of the i^{th} body relative to the directions of the global coordinate system.

The conversion from local to global coordinate system is

$$\mathbf{r}_G = \mathbf{r}_P + \mathbf{r}_{G|P}. \quad (2)$$

B. Kinematic

From (2), the velocity, $\dot{\mathbf{r}}_G$, and acceleration, $\ddot{\mathbf{r}}_G$, of G can be found as

$$\dot{\mathbf{r}}_G = \dot{\mathbf{r}}_P + \boldsymbol{\omega}_i \times \mathbf{r}_{G|P} + {}^i\dot{\mathbf{r}}_{G|P}, \quad (3)$$

and

$$\ddot{\mathbf{r}}_G = \ddot{\mathbf{r}}_P + \dot{\boldsymbol{\omega}}_i \times \mathbf{r}_{G|P} + \boldsymbol{\omega}_i \times (\boldsymbol{\omega}_i \times \mathbf{r}_{G|P}) + {}^i\ddot{\mathbf{r}}_{G|P}. \quad (4)$$

Noting that ${}^i\dot{\mathbf{r}}_{G|P} = {}^i\ddot{\mathbf{r}}_{G|P} = 0$ since there is no relative motion between G and P , and for the 2D case $\boldsymbol{\omega}_i = \dot{\theta}_i \mathbf{k}$, where $(\mathbf{i}, \mathbf{j}, \mathbf{k})$ is the basis of the global coordinate system.

C. Model of the WET-NZ device

The device is represented by two rigid bodies: (1) reactive hull and (2) active float. They are kinematically constrained at the pivot point P , denotes as P_1 on the reactive hull and P_2 on the active float. The constraint equation is simply

$$\mathbf{r}_{P_1} = \mathbf{r}_{P_2} = \mathbf{r}_P. \quad (5)$$

The two bodies can however be rotated independently about the pivot point P , so that the global position of the whole device can be fully represented by a state vector, $\mathbf{X}$,

$$\mathbf{X} = \{ x_P \quad y_P \quad \theta_1 \quad \theta_2 \}^T, \quad (6)$$

where x_P and y_P are the global coordinates of P , θ_1 and θ_2 are rotation angles of bodies 1 and 2 from the global coordinate directions, and T denotes the transpose of the vector.

1) *Reactive Hull*: Fig. 2 shows a free-body diagram of the forces imposed on the body. These are:

- The gravitational force, $\mathbf{F}_G = mg$ acting downward at the center of mass, where m is the mass of the hull and g is gravitational acceleration;
- The buoyancy force and moment acted at point P , $\mathbf{F}_{FK}$ and $\mathbf{M}_{FK}$, calculated from the Froude-Krylov integration of the hydrostatic pressure over the wetted area of the body;
- The hydrodynamic force and moment acted at point P , $\mathbf{F}_H$ and $\mathbf{M}_H$, approximated from the Morison's forces;
- Mooring forces acting at J mooring points, where $\mathbf{F}_{M_j}$ is the mooring force acted at the j^{th} mooring point;
- The reaction force and moment from the pivot bearing, $\mathbf{F}_R$ and $\mathbf{M}_R$. The moment is caused by the bearing friction and is also a function of $\mathbf{F}_R$. Any arbitrary torque induced by the PTO can be added to $\mathbf{M}_R$ if necessary.

2) *Active float*: Similar forces to the reactive hull are present on the active float, except that the reaction force and moment are in the opposite direction. It is also possible to include mooring forces in the active float, however, we have not investigated the effects of mooring from the active float at this stage.

3) *PTO*: The PTO system consists of a hydraulic ram to convert a slow oscillating torque between the active float and the reactive platform into hydraulic pressure to drive a hydraulic motor which is then coupled to a generator. The forces generated by the generator load, viscous loss and frictional loss can be modeled as forces which are dependent on the motion of the piston relative to the cylinder. The cylinder is attached to the active float at point C and to the reactive hull at point B as shown in Fig. 3. The hydraulic ram

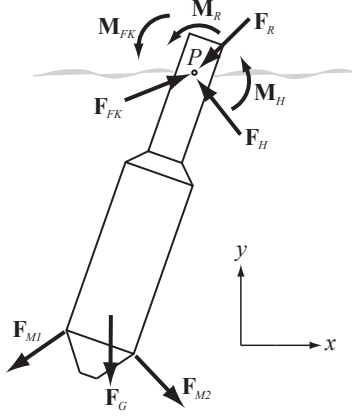


Fig. 2. Forces and moments on the reactive hull

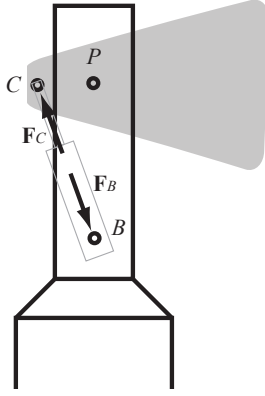


Fig. 3. Schematic diagram of the hydraulic cylinder

force is represented by $\mathbf{F}_C$ and $\mathbf{F}_B$ acting at point C and B , respectively, where $\mathbf{F}_C = -\mathbf{F}_B$, along the line $\overline{CB}$.

4) *Mooring force*: The mooring system for the WET-NZ device is based on slack mooring, similar to ship mooring so that the device can rise and fall with tidal variation, and there should be minimal vertical forces imposed on the system. Mooring of floating devices is a complex problem that can be analyzed using commercial packages such as OrcaFlex [8]. At this stage of development, to approximate the effects of mooring forces on the response of the device, we have assumed that each mooring line produces a force at the attachment point on the device. The key parameters are the number of mooring lines (in the 2D plane), and the positions and angles of attachment to the device.

Each mooring line is represented by a tension-only spring force proportional to the distance between the mooring point M_i on the floating body and the anchor point A_i , and a damping force which is proportional to the rate of change of the distance between M_i and A_i ,

$$\mathbf{F}_{M_i} = K_i (|\mathbf{r}_{M_i|A_i}| - L_i) - C_i (\dot{\mathbf{r}}_{M_i|A_i}), \quad (7)$$

for $(|\mathbf{r}_{M_i|A_i}| - L_i) > 0$,

where K_i , C_i and L_i are the stiffness, damping coefficient and

initial length of the i^{th} mooring line, respectively.

D. Hydrostatic effects

The pressure, p , at any point in a regular wave field can be found from

$$p(x, y, t) = \rho a g \frac{\cosh(kh + ky)}{\cosh kh} \cos(kx - \omega t) - \rho g y, \quad (8)$$

where ρ is the water density, a is the wave amplitude, $\omega = 2\pi/T$ is the angular wave frequency and T is the wave period, $k = 2\pi/\lambda$ is the wave number and λ is the wavelength, and h is the depth of the water.

The force on the submerged volume, V , can be found by finding the Froude-Krylov integration of the hydrostatic pressure over the wetted area of the body, that is

$$\mathbf{F}_{FK} = -t_k \int_S p \cdot \mathbf{n} dS, \quad (9)$$

where t_k is the sectional thickness of the device, and $\mathbf{n}$ is the vector normal to the perimeter, S , of the submerged volume.

If there is no wave, then $\mathbf{F}_{FK}$ simply equals to the buoyancy force, that is $\mathbf{F}_{FK} = \rho g V$ acting upward at the center of buoyancy of the submerged volume.

For generality, this force is shifted to act at the pivot point P and a counter moment, $\mathbf{M}_{FK}$, about P is added.

E. Hydrodynamic effects

Chitrapu & Ertekin [4], [9] investigated two methods of simulating large-amplitude response of floating platforms. One method uses Morison's forces to represent hydrodynamic effects, and the other method uses the potential theory. They concluded that the Morison's method gave reasonably accurate results as compared to the more accurate potential theory method, and that any methods must resort to experimental data for verification as there are no exact solutions to this complex problem. For our model, it is necessary to consider large-amplitude motion and so Morison's forces have been adopted to represent the hydrodynamic effects on our device. The Morison's equation for a vertical pile is given as [7]

$$F_x = \underbrace{C_m \frac{\rho \pi D^2}{g} \ddot{u}_x}_{\text{inertia}} + \underbrace{C_d \frac{\rho}{2g} D |\dot{u}_x| \dot{u}_x}_{\text{drag}}, \quad (10)$$

where F_x is the horizontal force, C_M is the inertial coefficient, C_d is the drag coefficient, D is the diameter of the cylinder, and $\dot{u}_x$ is the horizontal water velocity.

We have assumed that the reactive hull is a long slender object, and by breaking up into several sections each section, i^{th} can have different cross-sectional dimension and coefficients, we can determine the hydrodynamic forces and moments as followed

$$\mathbf{F}_H = \mathbf{F}_{C_a} + \mathbf{F}_{C_d}, \quad (11)$$

$$\mathbf{M}_H = \mathbf{M}_{C_a} + \mathbf{M}_{C_d}, \quad (12)$$

where $\mathbf{F}_{C_a}$ and $\mathbf{M}_{C_a}$ are the force and moment acting at P caused by the hydrodynamic inertial effects, and $\mathbf{F}_{C_d}$ and $\mathbf{M}_{C_d}$ are the force and moment acting at P caused by the hydrodynamic drag effects.

These vectors can be found by integrating along the axial direction of the hull, with l being the coordinate along the axis, and l_i and l_{i+1} are the two end points of the i^{th} section

$$\begin{aligned}\mathbf{F}_{C_a} &= \sum_i \int_{l_i}^{l_{i+1}} (\rho A_i C_{a N_i} (\ddot{\mathbf{u}}_l - \ddot{\mathbf{r}}_l) \cdot \mathbf{n}) \mathbf{n} dl \\ &\quad + \rho V C_{a T} \mathbf{t}, \\ \mathbf{M}_{C_a} &= \sum_i \int_{l_i}^{l_{i+1}} l (\rho A_i C_{a N_i} (\ddot{\mathbf{u}}_l - \ddot{\mathbf{r}}_l) \cdot \mathbf{n}) \mathbf{n} dl, \quad (13)\end{aligned}$$

where A_i is the cross-sectional area, $C_{a N_i}$ is the inertial coefficient normal to the axial direction of the i^{th} section, and $C_{a T}$ is the inertial coefficient tangential to the axial direction, and $\mathbf{n}$ and $\mathbf{t}$ are the normal and tangential unit vectors in the axial direction.

Similar equations can be derived for the drag forces noting that the velocity term in the Morison's equation is $\dot{\mathbf{u}}_l - \dot{\mathbf{r}}_l$, that is the relative velocity between the water and the hull.

Another reason for using the Morison's equation is that we can extend our model to include irregular waves by using the method described in [10].

F. Pivot bearing model

Although it is possible to algebraically manipulate the equations to eliminate the internal reaction force, $\mathbf{F}_R$, from the equations of motion, it is very tedious and prone to errors. We used a more generic approach of treating the two x, y components of the reaction force in the equations as the unknown variables and they will be solved as parts of the solutions. This simply means that we have to include additional constraint equations in the system equations. It should be noted that $\mathbf{F}_R$ is added to the derivative of the state vector, $\dot{\mathbf{X}}$, and so the solutions in $\mathbf{X}$ are the x, y components of the integral of $\mathbf{F}_R$.

Another complication is that the bearing friction loss is dependent on this reaction force, where

$$\mathbf{M}_R = \mu |\mathbf{F}_R| r_b \text{sgn}(\dot{\theta}_2 - \dot{\theta}_1), \quad (14)$$

where $\mathbf{M}_R$ is a moment acted on the reactive hull, μ is the bearing friction coefficient, and r_b is the radius of the pivot shaft.

G. Wave model

Wave motion, $\mathbf{u} = \{ u_x \ u_y \}^T$ can be determined from the 2D linear wave theory for monochromatic waves as

$$u_x = -\frac{agk}{\omega^2} \left(\frac{\cosh(ky + kh)}{\cosh kh} \right) \sin(kx - \omega t) + v_c t, \quad (15)$$

$$u_y = \frac{agk}{\omega^2} \left(\frac{\sinh(ky + kh)}{\cosh kh} \right) \cos(kx - \omega t), \quad (16)$$

where v_c is the constant horizontal water current velocity, and t is time.

It should be noted that these equations are also valid for shallow water. In shallow water, the horizontal motion is larger than the vertical motion, and it will be interesting to see if our device shows significant improvement over heave-only devices.

H. Newton-Euler equations

Multi-body dynamic models are usually derived from Newton-Euler equations, and there are several generalized methods for assembling sub-systems into the main system. We adopted the algorithms in [11] to construct the non-linear dynamic equations of motion for the WET-NZ device.

Three equilibrium equations in x, y , and θ can be formed for each object. Using the acceleration equation in section III-B, the Newton-Euler equations of the reactive hull can be expressed as

$$\begin{bmatrix} m_1 & 0 & -y_{G_1|P} m_1 \\ 0 & m_1 & x_{G_1|P} m_1 \\ -y_{G_1|P} m_1 & x_{G_1|P} m_1 & M_{33} \end{bmatrix} \begin{Bmatrix} \ddot{x}_P \\ \ddot{y}_P \\ \ddot{\theta}_1 \end{Bmatrix} - m_1 \dot{\theta}_1^2 \begin{Bmatrix} x_{G_1|P} \\ y_{G_1|P} \\ 0 \end{Bmatrix} - \begin{Bmatrix} \sum F_x \\ \sum F_y \\ F_3 \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix}, \quad (17)$$

$$M_{33} = I_{G_1} + m_1 (x_{G_1|P}^2 + y_{G_1|P}^2),$$

$$F_3 = \sum \mathbf{M}_x + \sum_k (\mathbf{r}_{k|P} \times \mathbf{F}_k),$$

where G_1 is the center of gravity of the hull, m_1 and I_{G_1} are the hull mass, and mass moment of inertia about G_1 , $\mathbf{r}_{G_1|P} = \{ x_{G_1|P} \ y_{G_1|P} \}^T$, $\mathbf{F} = \{ F_x \ F_y \}^T$ is any force vector acting on P , $\mathbf{F}_k$ is any force vector applied at point k on the hull, and $\mathbf{r}_{k|P}$ is the global position vector of point k relative to point P .

A similar system of equations can be formed for the active float to provide 6 equations, and since the state vector (6) has only 4 components, adding the 2 integral components of the reaction force $\mathbf{F}_R$ to the state vector gives 6 unknowns which can be solved by the 6 governing equations. Since the equations of motion is a second-order equation, additional 4 velocity terms and 4 trivial equations are added to the system equations to convert it to a first-order system, that is the total WET-NZ device is represented by 10 first-order nonlinear differential equations.

Coupling between surge, heave and pitch are all taken care of by the algorithm as all hydrostatic and hydrodynamic forces are calculated at each time step from the instantaneous position of the device.

I. Numerical integration

Because the bearing friction is dependent on the reaction force which is a solution of the system, it is necessary to solve using a fully implicit differential equations solver, such as the `ode15i` solver in MATLAB, which seeks the solution for a set of differential equations of the form $\mathbf{f}(t, \mathbf{X}, \dot{\mathbf{X}}) = \mathbf{0}$.

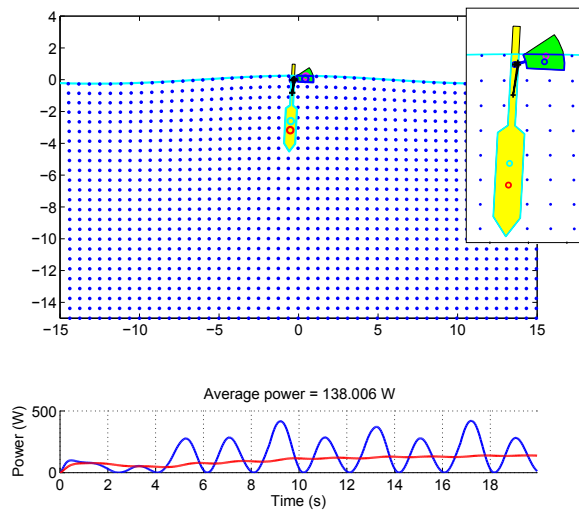


Fig. 4. A simulation of the WET-NZ experimental device

The model has been verified with numerous known cases using hydrodynamic coefficients estimated from the literature as well as data from small scale laboratory tank tests.

IV. EXPERIMENTAL PROOF-OF-CONCEPT (POC) DEVICE

An experimental device was designed to be deployed 0.5 km off the east coast of Christchurch in 15 m deep water. This site was chosen because of easy access from Lyttelton harbor, cellphone coverage for control and data transmission, and within visual inspection from shore to allow video capture of device motion.

The device was designed to be as large as possible within constraints such as: cost, capability of our workshop, transportability by a Hiab truck, and able to be towed to deployment site by a small work vessel. Based on these constraints, the device dimensions were chosen to be approximately $5 \times 1.7 \times 0.5$ m for the hull, 3.5 tonnes with entrained water; the active float has a waterline area of 1 m^2 and weighs 300 kg. The total dry weight of the whole device is approximately 1.7 tonnes.

Fig. 4 shows a typical simulation of the device operating in 0.5 m, 4 s monochromatic waves.

Figures 5 – 7 show photographs of the device being dropped into the water using the Hiab, being towed to deployment site by an inflatable jet-boat, and the device in operation during 2 months in the water. Fig. 8 shows a plot of power captured over an 18-hour period. The overall trend of the power curve relates to the wave conditions at the time, and peak electrical power of nearly 400 W has been observed. Pre-deployment testing and analysis showed that substantial losses occur in the drive train at this scale and the mechanical power transferred to the device by the waves is estimated to be two to three times the measured electrical power.

After two weeks in the water, we lost communication with the device and it was found that water ingress into the battery compartment was the cause. The hydraulic system continued to operate over the 2 months in the water until the



Fig. 5. The WET-NZ device being deployed at Lyttelton harbor



Fig. 6. The device being towed to deployment site

device was removed for maintenance and data retrieval. The communication system is being repaired and the device will resume operation in the near future.

Preliminary observation indicates that the numerical model provided realistic responses of the device under wave actions when resistances from mechanical losses and electrical power generated are taken into account, although quantitative analysis has not been completed at this stage.

V. FULL-SIZE DEVICE

It is envisaged that the first generation full size device would operate between 100 kW to 500 kW peak, and a commercial wave farm would consist of arrays of multiple devices spreading over a wide area in intermediate water depth, but sufficiently far from land to minimize environmental impacts. As the technology matures, larger scale devices are



Fig. 7. The device in water

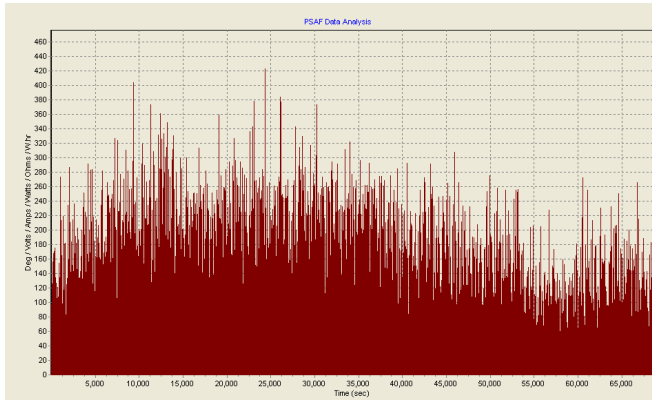


Fig. 8. Power captured over an 18-hour period

expected.

As an example of the full size device, the simulated experimental model was scaled up approximately 6 times to give a hull size of $30 \times 10 \times 6$ m, and entrained mass of 1,500 tonnes. The active float has a water line area of 36 m^2 and a mass of 70 tonnes. Fig. 9 shows an example of the device in 3 m, 12 s regular wave regime, which transfers 95 kW average power to the device, peaking at 300 kW. Quantification of the device operation in irregular sea states is a key element of ongoing research.

VI. CONCLUSION

A multi-mode self-reacting wave energy converter developed by the WET-NZ project has been described in this paper. A comprehensive model that can handle complex multi-mode energy transfer into and through the device has been developed. The model was based on multi-body dynamics but it also relied on empirical knowledge of the hydrodynamic interactions using Morison's method. An experimental device has been built to provide verification data for the model and to demonstrate its power generation capability in real sea environment. Sufficient knowledge of the device operation has been gained from the models to extrapolate the design to a half-scale (20 kW peak) device. Development of this device is now underway.

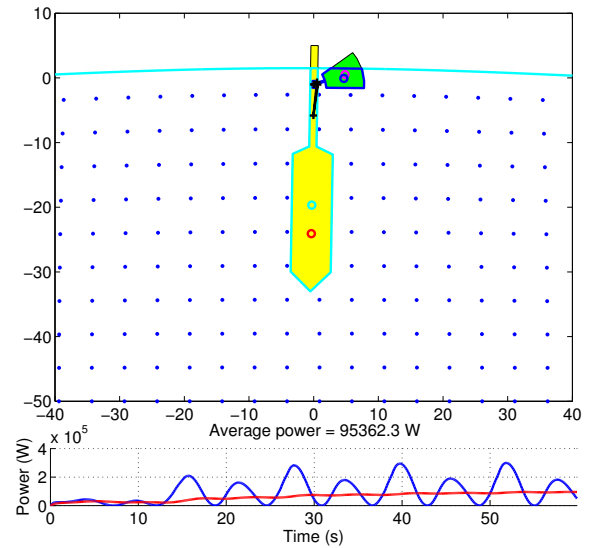


Fig. 9. A simulation of the full size device

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