Investigating the Performance of a Hydraulic Power Take-Off

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Abstract

Many wave energy converter (WEC) developers are in the process of scaling up previous point absorber concepts to 100kW+ devices. As sizes increase, serious challenges in power take off (PTO) design are being encountered. Hydraulic transmissions are favoured as gearing up to give high speed rotary motion is easily achieved, and power density is high. They are also highly controllable, for example allowing take-off cylinders to be locked or free-running as conditions require via simple valve control.

Theoretical concepts for extracting the maximum power from waves already exist, using reactive or latching control for example, but these do not account for the real engineering limitations or losses in practical PTO systems. The concepts assume that the PTO can generate any given force/motion relationship with equal efficiency.

This paper presents the results of a simulation study on the design and control of a hydraulic PTO. The control strategies for maximum power extraction are compared with and without consideration of the characteristics (losses) in the PTO. It is shown that efficiency can be surprisingly low. It is also shown that using components (motor and pump in this case) sized for smaller wave conditions may be preferable as their efficiency is better.

Keywords: wave energy converters, power take-off, hydraulics.

Nomenclature

s	differential operator
M	buoy mass
C, K	linearised resistance (drag), buoyancy stiffness
C_p, K_p	PTO damping, stiffness
f_e, f_p	excitation force, PTO force
r_c	ratio PTO damping / hydrodynamic damping
P_{av}	power absorbed from buoy
v, z	force, velocity, displacement
H, T	wave height, wave period
D	pump/motor displacement

1 Introduction

This paper is concerned with the design and control of hydraulic power take offs (PTOs) for point absorber wave energy converters (WECs). Many previous papers have addressed the modelling and control of point absorbers [1,2], and large scale (100kW+) point absorbers are currently being developed by a number of companies, e.g. [3]. However hardly any theoretical investigations have properly accounted for PTO losses, and the aim here is to show that these losses can have a dramatic effect on performance. Although focused on point absorbers, the results shown are of relevance to any design requiring conversion of reciprocating to high speed rotary motion.

Hydraulic power transmissions are normally chosen for high power WECs as they are easily controlled and have a high power density [4]. The hydraulic PTO shown here is used for illustrative purposes; it is not proposed as necessarily the best design. It uses a single reciprocating cylinder attached between a buoy and a stationary reaction point, driving flow through a fixed capacity hydraulic motor. This is coupled to a variable capacity pump delivering flow to a large accumulator at nominally constant pressure. One or more constant speed hydraulic motors and synchronous generator units can then be driven from the constant pressure supply (or disabled when insufficient power is available). This part of the system is not modelled and not within the scope of the current analysis – it should be relatively efficient if optimised for the correct steady state conditions.

The variable capacity pump allows the PTO to be controlled to vary the force between cylinder and buoy. Varying the force allow strategies such as reactive control to be implemented, where the PTO impedance is varied to maximise power absorption (in theory) by achieving resonance and an optimal level of damping. Alternative methods for controlling force include:

- replacing the variable capacity pump with a torque-controlled asynchronous generator;
- having several cylinders of different piston area which can be switched into the hydraulic circuit connected to the (nominally) constant pressure accumulator.

Simpler, uncontrolled, systems use a bridge of four check valves to rectify the flow from a cylinder to charge an accumulator.

2 Force Control Strategy

To test the efficiency of the PTO, a control strategy is required. This will be based on reactive control, but with limits on buoy displacement and velocity for practical reasons (PTO stroke and flow limits).

Using a linear hydrodynamic model,

$$(Ms^2 + Cs + K)z = f_e - f_p \tag{1}$$

where M is the buoy mass and added mass, C is a linearised radiation resistance/drag term, and K is the buoyancy stiffness. The buoy upward displacement is z, upward wave excitation force is f_e, and downward PTO force is f_p.

The force control algorithm is chosen as:

$$f_p = (C_p s + K_p)z \tag{2}$$

From Eq. (1) and (2), the buoy velocity is:

$$v = \frac{s}{Ms^2 + (C + C_p)s + (K + K_p)} f_e \tag{3}$$

A method for choosing C_p and K_p is required. Consider a regular (sinusoidal) wave of frequency ω. In the frequency domain, the amplitude of velocity and excitation force are related by:

$$V = \frac{1}{\sqrt{(C + C_p)^2 + \left(\frac{K + K_p - M\omega^2}{\omega}\right)^2}} F_e \tag{4}$$

or

$$V = \frac{1}{\sqrt{(1 + r_c)^2 + \left(\frac{K + K_p - M\omega^2}{C\omega}\right)^2}} \frac{F_e}{C} \tag{5}$$

where $C_p = r_c C$ (thus PTO damping is chosen as a multiple r_c of hydrodynamic damping). The average power absorbed from the buoy by the PTO is the averaged product of the damping force $C_p v$ and velocity v, which for a sinusoid is:

$$P_{av} = \frac{1}{2} r_c C V^2$$

so

$$P_{av} = \frac{4r_c}{(1 + r_c)^2 + \left(\frac{K + K_p - M\omega^2}{C\omega}\right)^2} \left(\frac{F_e^2}{8C}\right) \tag{6}$$

It can be shown that to maximise the power from the buoy (i.e. maximise P_{av}):

$$K_p = K - M\omega^2$$

$$r_c = 1$$

This is reactive control: K_p (which is usually negative) alters the natural frequency to match the wave frequency i.e. the buoy is in resonance. The 'reactive term' – the second term in the denominator of Eq. (6) – disappears because the inertia and stiffness forces cancel.

However this choice for K_p and r_c may give too large a motion amplitude (from Eq. 6). To reduce the amplitude K_p or r_c can be increased, but it is clear from Eq. (6) that the latter is best as r_c also appears in the numerator and power will not be so greatly affected.

In summary, the desired PTO force is given by Eq. (2) with

$$K_p = K - M\omega^2 \tag{7}$$

and $r_c = max(1, r_{cv}, r_{cz})$ (8)

where (from Eq. 5):

$$r_{cv} = \frac{F_e}{CV_{\lim}} - 1 \tag{9}$$

$$r_{cz} = \frac{F_e}{C\omega Z_{\lim}} - 1 \tag{10}$$

and where V_{lim} is the velocity limit, and Z_{lim} is the displacement limit (typically half the cylinder's working stroke).

Power from the buoy (Eq. 6) reduces to:

$$P_{av} = \frac{4r_c}{(1 + r_c)^2} \left(\frac{F_e^2}{8C}\right) \tag{11}$$

Such a control strategy requires an estimation of the dominant wave excitation force amplitude and frequency; how this might be done is not addressed here.

3 Ideal PTO Simulation Results

The parameters of Table 1 were used to test the control strategy. The values of r_c which are greater than 1 result from the velocity limit; the stroke is not a constraint in this case. The wave force is approximated by the product of buoyancy stiffness and wave position:

$$F_e = K \frac{H}{2} \tag{12}$$

Figures 1 to 3 show the behaviour for the three wave conditions. The average powers are calculated numerically according to Eq. (11). As expected, the controller works well to maximise the power absorbed within the physical constraints.

Hydrodynamic parameters	Mass+added mass, M	180 tonne
	Damping, C	60 kN/(m/s)
	Buoyancy stiffness, K	277 kN/m
Constraints	Velocity limit, V_{lim}	1.16m/s
	Displacement limit, Z_{lim}	2.0 m
Wave conditions (sinusoidal: amplitude $H/2$, period T)		
a)	$H = 1$m, $T = 6$s	$r_c = 1.0$
b)	$H = 2$m, $T = 8$s	$r_c = 3.0$
c)	$H = 3$m, $T = 10$s	$r_c = 5.0$

Table 1: Parameters for ideal PTO testing.

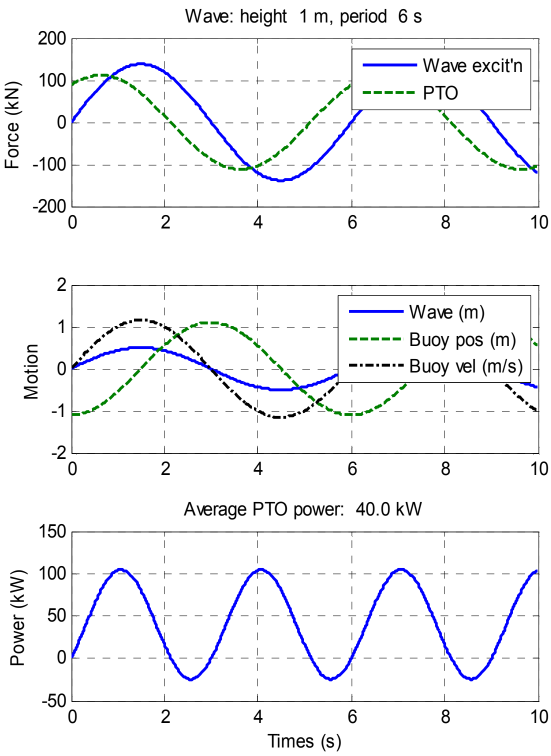

Figure 1: Ideal PTO with 1m wave, $T = 6$s, $r_c = 1$.

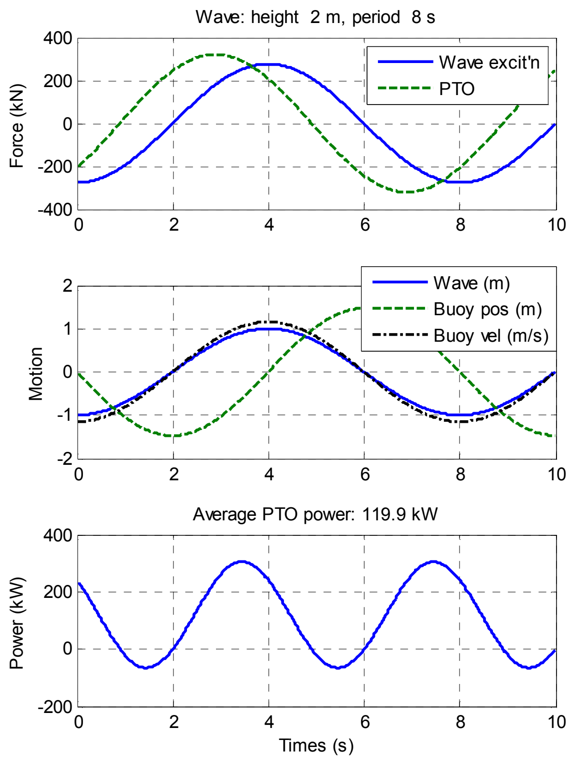

Figure 2: Ideal PTO with 2m wave, $T = 8$s, $r_c = 3$.

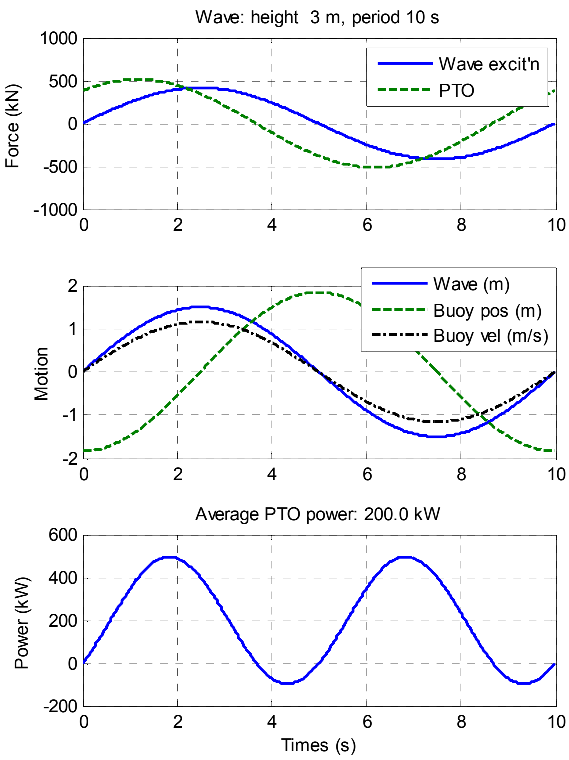

Figure 3: Ideal PTO with 3m wave, $T = 10$s, $r_c = 5$.

4 PTO Design Including Loss Model

Figure 4 shows the PTO concept. The accumulator is assumed to be sufficiently large so that its pressure can be assumed constant, and thus the pump (and motor) torque is proportional to the pump displacement. Losses aside, the cylinder force is proportional to motor torque, so the pump displacement is the control input used to alter the PTO force. An over-centre pump is used, meaning that the flow can reverse direction with unidirectional shaft rotation; this occurs when the pump displacement is negative. The part of the system which draws oil off to drive the generator is not shown, and losses for this part are not accounted for.

In reality both pump and motor can transmit power in the opposite direction, i.e. the pump can act as a motor, and the motor as a pump.

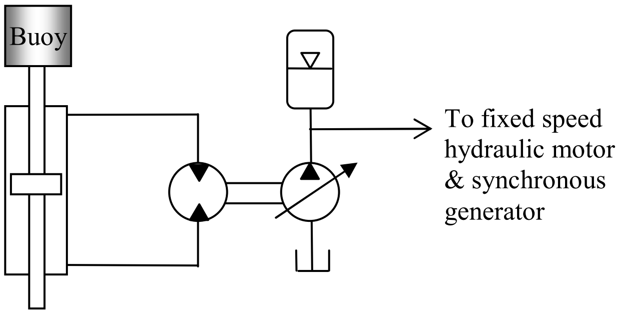

Figure 4: Hydraulic PTO concept.

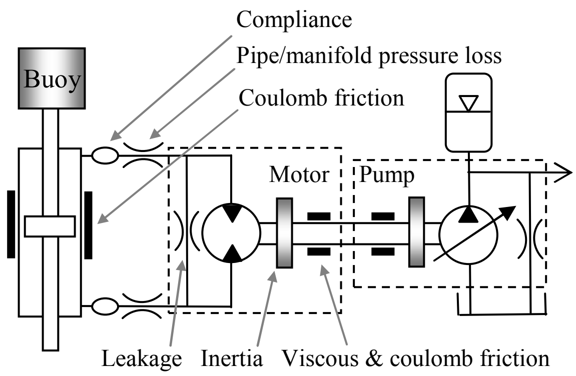

Figure 5: Hydraulic PTO with non-ideal characteristics.

Figure 5 indicates the main non-ideal characteristics which occur in this PTO. The losses are:

- coulomb friction in the cylinder
- viscous and coulomb friction in the pump and motor
- leakage in pump and motor (laminar)
- pipe and manifold pressure loss (turbulent)

Other unwanted characteristics, but ones which do not dissipate energy, are:

- oil and hose compliance
- pump and motor inertia

Note that pressure loss and compliance are really distributed effects, but a lumped parameter model is used here. Pump and motor losses conform to the well established Wilson model [5] except that compressibility loss is neglected.

5 Realistic PTO Simulation Results

Table 2 shows a set of PTO parameters. The same type of unit is used for both pump and motor (albeit the former is a variable capacity version), so the parameters are the same for each. The values are for a Denison Hydraulics Gold Cup Series 30 unit [6]. The losses (leakage, coulomb friction and viscous friction) approximate the loss characteristics measured by the manufacturer and shown as part of the performance curves reproduced in Fig. 6. Note that the ideal relationships are:

flow rate = $D \times$ angular velocity
torque = $D \times$ pressure difference

where D is the pump (or motor) displacement.

The oil/hose stiffness value is based on the approximate volume of trapped oil, a bulk modulus of 0.9 GN/m^2 and a 2.5m length of hose at each cylinder port with effective bulk modulus of 0.23 GN/m^2. Cylinder fiction and pressure loss values are the best expected for a well designed system.

Pump & motor parameters	
Maximum pressure	350 bar
Maximum speed	1800 rev/min
(Maximum) capacity (D)	0.502 L/rev
Inertia	0.286 kgm^2
Leakage	0.167 L/min/bar
Coulomb friction	70 Nm
Viscous friction	0.06 Nm/(rev/min)
Cylinder parameters	
Area (A)	0.013 m^2
Coulomb friction	5 kN
Stroke	4.0m
Lumped model parameters	
Oil/hose stiffness (referred to cylinder)	1.89 kN/mm
Pipe/manifold pressure loss (proportional to square of flow)	20 bar @ 900 L/min
Derived parameters	
Cylinder stall force	455 kN
Velocity limit (V_{lim})	1.16 m/s
Displacement limit (Z_{lim})	2.0 m

Table 2: PTO parameters (Series 30 pump & motor).

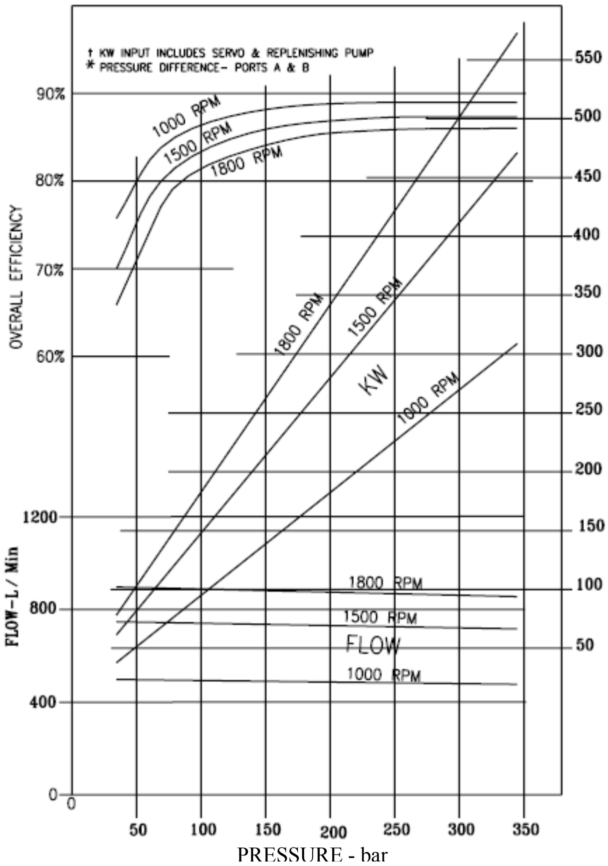

Figure 6: Pump performance curves (Denison Hydraulics Gold Cup Series 30) [6].

The system has been designed to work in waves of up to 3m, and the piston area has been chosen to give a stall force equal to the 3m wave excitation force amplitude.

Results are presented for the same hydrodynamic and wave parameters as before (Table 1), and as the velocity and displacement limits have not changed, the controller parameters are also the same. The desired PTO force is now used to adjust the pump displacement , and the achieved force will deviate from the desired due to various losses. The simulation model is implemented in Simulink; Fig. 7 shows the top level block diagram.

Figures 8-10 show the results equivalent to Figs. 1-3. As the piston force needs to overcome the coulomb, viscous and pipe friction forces, the PTO force tends to be higher than desired, and buoy movement is less. Leakage losses throughout the cycle add to the problem. Clearly the device should be deactivated in the smallest waves , and shows about 30% efficiency in 3m waves (or about 20% of the theoretical maximum power generation with no velocity limit, i.e. $r_c = 1$).

In an attempt to improve efficiency for smaller waves, smaller pump and motor units could be used. The new pump/motor parameters are shown in Table 3, together with the new velocity limit; other parameters are the same. The same tests are performed, but due to the lower V_{lim}, the PTO damping needs to increase; from Eqs. (8) and (9), r_c = 2.3, 5.5 and 8.8 for waves a), b) and c) respectively (refer to Table 1).

The average powers transmitted by the PTO in the ideal case (no losses) are 34kW, 83kW, 132kW respectively (smaller than the previous values – 40kW, 120kW and 200kW – due to the larger r_c values). With the realistic PTO simulation, Figs. 11 to 13, there is an improvement for the two smaller waves compared with using the larger pump/motor units.

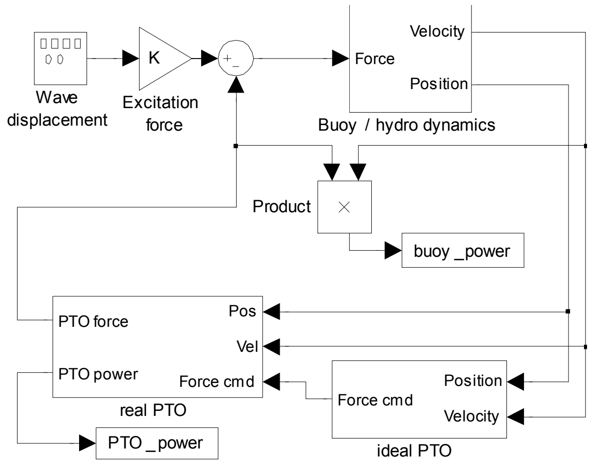

Figure 7: Simulink model of the PTO system.

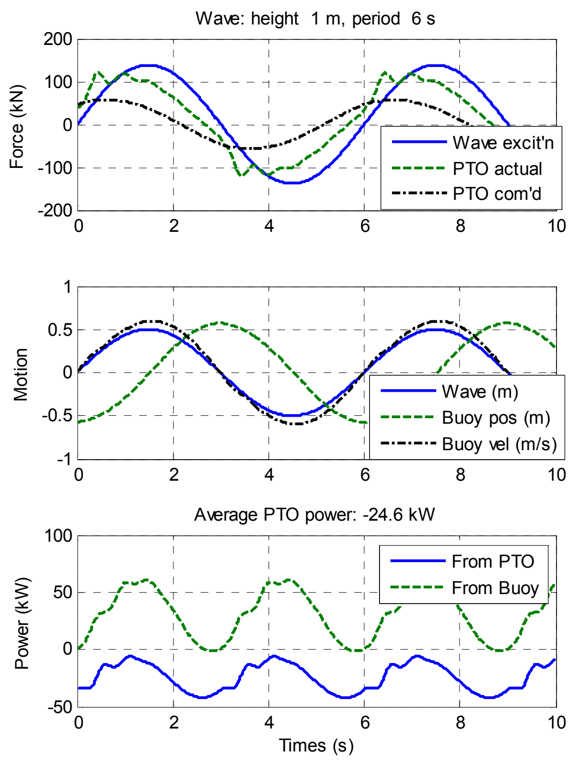

Figure 8: PTO with losses, 1m wave. $T = 6$s, $r_c = 1$.

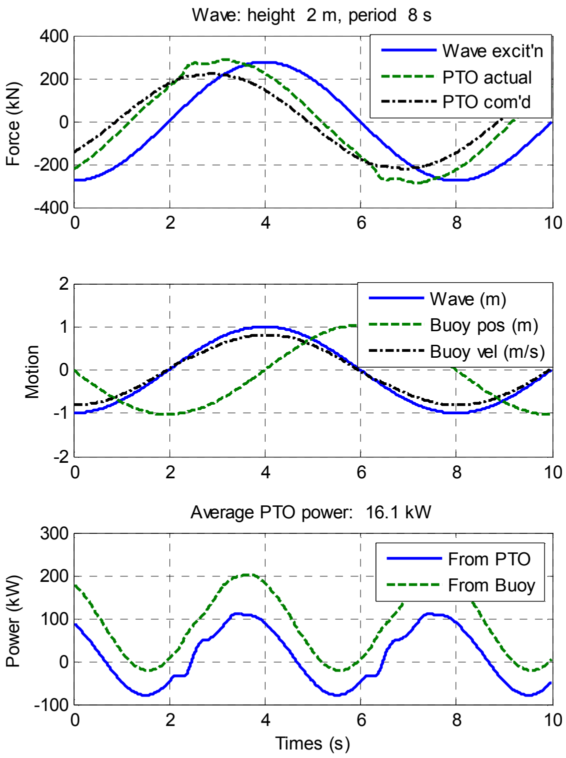

Figure 9: PTO with losses, 2m wave. $T = 8$s, $r_c = 3$.

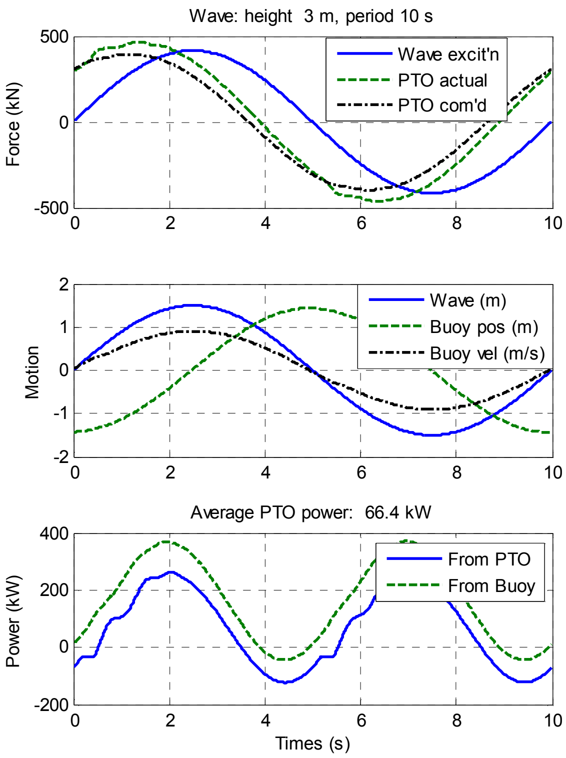

Figure 10: PTO with losses, 2m wave. $T = 10$s, $r_c = 5$.

6 Conclusions and Discussion

Simulating system behaviour including PTO losses can reveal that the power generated might be much lower than expected – only about 30% of the ideal value for 3m waves in this case. For 1m waves no power is generated. For 2m waves, the smaller pump and motor units investigated give better power absorption due to improved efficiency. In fact, as the smaller waves are likely to be more common, the smaller pump/motor is probably preferable.

Both the design and control of PTO's must be based on an understanding of real-life characteristics and realistic loss models. The PTO simulated here is only one possible design, but other designs will exhibit similar problems. The fundamental issue is that power transmissions (both hydraulic and electrical) can be designed to be fairly efficient around one operating condition, but the extremely variable quantity of power available from waves means that for most of the time the operating condition will be far from the optimum. This variability occurs both within one wave and between sea states. Systems designed to extract power from large (high power) waves, are likely to be unable to generate any power in more prevalent calmer conditions.

Open loop PTO control is used here – it is a common assumption that this is sufficient. However it is clear that due to compliance, inertia, and various losses the desired force is not accurately achieved. A closed-loop force control method would be better, enabling the PTO behaviour to be much closer to the ideal impedance characteristic. Note also that the calculation of the force command depends on predominant wave frequency, thus this needs to be estimated.

Pump & motor parameters	
Maximum pressure	350 bar
Maximum speed	2400 rev/min
(Maximum) capacity (*D*)	0.23 L/rev
Inertia	0.085 kgm^2
Leakage	0.10 L/min/bar
Coulomb friction	20 Nm
Viscous friction	0.026 Nm/(rev/min)
Derived parameters	
Velocity limit (V_{lim})	0.71 m/s

Table 3: Alternative PTO parameters (Series 14 pump & motor).

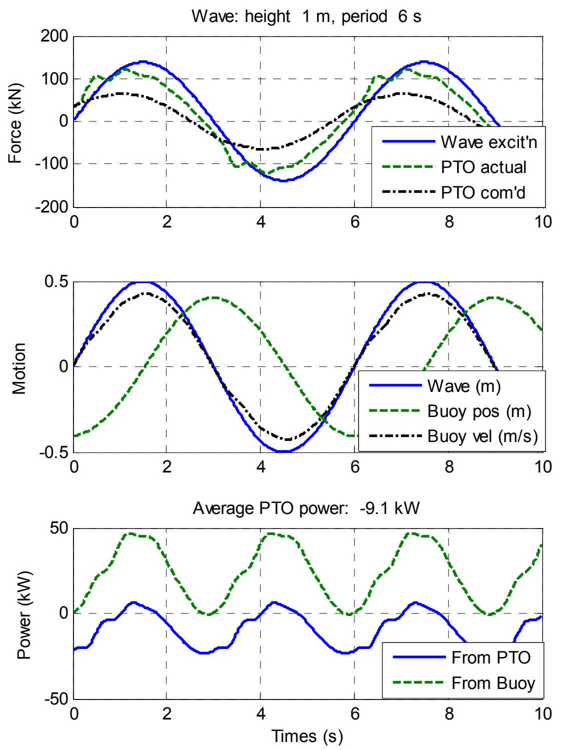

Figure 11: PTO with smaller pump and motor, 1m wave. $T = 6$s, $r_c = 2.3$.

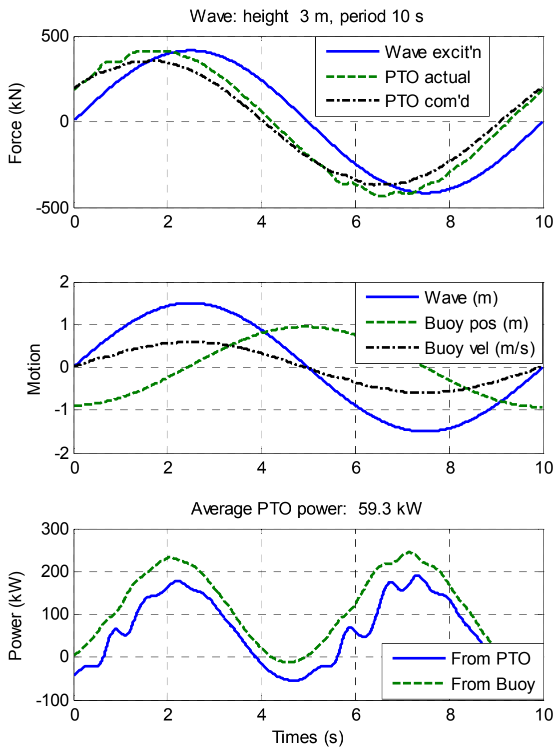

Figure 13: PTO with smaller pump and motor, 3m wave. $T = 10$s, $r_c = 8.8$.

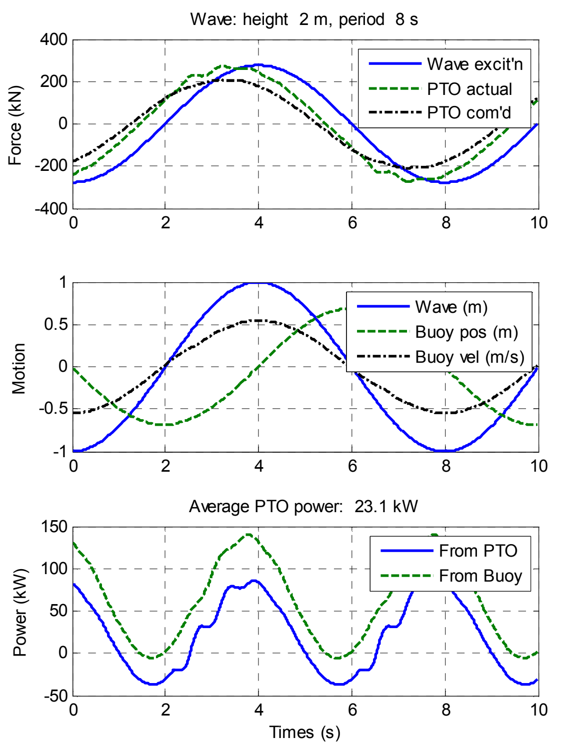

Figure 12: PTO with smaller pump and motor, 2m wave. $T = 8$s, $r_c = 5.5$.

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