All electric drive train for wave energy power take off

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Abstract

This paper tackles a fundamental weakness in all wave energy converters: the electro-mechanical power conversion. Initial progress on a project focusing on an all electric drive train with speed enhancement and power converter topologies is presented. Simulation results of three case studies are used to discuss the possibilities of speed enhancement in wave energy converter all electric power take off.

1 Introduction

Wave energy research in the UK started in the 1970s and focused on large single devices. Initial cost calculations showed these large structures were unlikely to be economic. Focus shifted to smaller devices, fuelled in part by the discovery of the point absorber effect, whereby a resonating device which is small relative to the wavelength can capture more energy than that contained in a wave front of its own width [1]. The effect is achieved with the device resonating at the predominant wave frequency, the wave force continuously in phase with the device velocity and the assumption of zero viscous losses. Devices must be small (~10m) relative to the wavelength (~100m). Since 2000 there has been a broad range of devices proposed, mostly between these size extremes, some of which have been optimised and tested. EMEC, Wave hub, NaREC and various universities meant the UK had a thriving wave energy testing seen. Sadly, in recent years, several of the more commercially advanced devices have folded, such as Pelamis, Aquamarine and WaveDragon. Many factors have caused this, of which a lack of supply chain of appropriate power take off technology is one. Devices were forced either to develop their own bespoke components, such as the Pelamis power module [2], or rely on power take of components developed for a completely different market.

Electrical machines have traditionally been designed to be driven at high speed rotary motion, typically from an internal combustion engine. For example, a 3000rpm electrical machine with an active diameter of 200mm has an air gap speed of 30 m/sec. A typical wave energy converter, however, can expect to produce oscillatory motion with velocities in the region of 0.5-2m/s. This discrepancy of velocity profile can be accommodated by a range of mechanical linkages used in devices, predominantly hydraulic or pneumatic in the systems demonstrated to date due to their availability off the shelf. There are concerns about the limitations including low efficiency at part load, ability to control over a wide range of frequencies and displacement leading to potential end-stop problems. Efficiency of a hydraulic Power Take Off (PTO) is dependent upon pressure and flow rate, with values quoted as high as 90%, but also as low as 40% at part load, where a device will spend much of its time. The alternative to hydraulics is an all electric direct drive, whereby the electrical machine is coupled directly to the moving part of the wave energy device.

In this paper all electric power take off will be discussed, using simple simulation of three case studies. It is the first step of a project which will make use of wave to wire models to fully evaluate the system [3]. Simulated results of architectures capitalising on amplitude amplification by integrated spring electrical machine, linear gearing and variation of an external spring will be presented.

2 Features of all electric power take off

In conventional electric machines, active area, mass and cost is predominantly dictated by required peak torque – slower direct drive power take off machines are physically larger than geared systems. This can be offset by increasing electric loading or using high force dense machines, for example modulated pole machines [4], which capitalise on magnetic gearing within a machine. Alternatively, speed amplification can be implemented via a separate magnetic gear, or inclusion of a spring element.

Increasing electric loading is discounted due to the squared relationship with losses. A magnetic gear is a non contact method of amplifying motion, where the input and output shafts are the only moving parts. The gearing effect is obtained by coupling harmonics of the magnetic field, rather than physical coupling of teeth as in a conventional gearbox [5, 6].

Inclusion of a spring element introduces the potential of mechanical resonance as a mechanism to amplify oscillation amplitude. The stiffness can be provided by mechanical, pneumatic or magnetic springs, and if it is variable will allow resonance across a range of mechanical frequencies. A magnetic spring is a force controlled motor, and it could be developed as a separate machine to the power take off.

Alternatively, the spring force can be emulated by controlling the reactive force of the electric machine to be in phase with its displacement – effectively integrating a magnetic spring with the power take off. Spring elements have been proposed for wave energy power take off elsewhere, either as an alternative to tight mooring for the provision of a reactive force or used to increase relative speed to benefit the electrical machine.

For a linear machine, increasing oscillation amplitude can only be accommodated by a corresponding increase in either stator or translator length. The potential active mass reduction associated with amplitude amplification is potentially offset by an almost linear increase in mass with stroke length. There is no such limitation in a rotary machine, where machine mass is constant regardless of oscillation magnitude.

To design an all electric power train machine which is mass competitive, the system likely requires some sort of amplitude amplification, or a high force density electrical machine is required –effectively a magnetic geared solution where pole size is reduced to give a high change in flux over a small displacement.

3 Case Study 1: A large Pitching device – constant frequency

3.1 The device

This section investigates the use of a spring element to implement amplitude acceleration. It considers the case where the power take off force does not affect the fundamental motion of the wave energy converter – a high inertia system.

Wave energy converters will be required to operate in a wide breadth of wave energy spectrum. Designers must optimise devices for a mean or significant energy band whilst ensuring the device can survive in energy ranges vastly away from this design point. Often in wave energy research, the hull is designed for optimum power take off in idealised sea conditions. Devices tend to be an order of magnitude shorter than the dominant wavelength (100m) in order to resonate at an appropriate frequency. Devices are made to survive extreme waves through robust design, resulting in over engineered solutions.

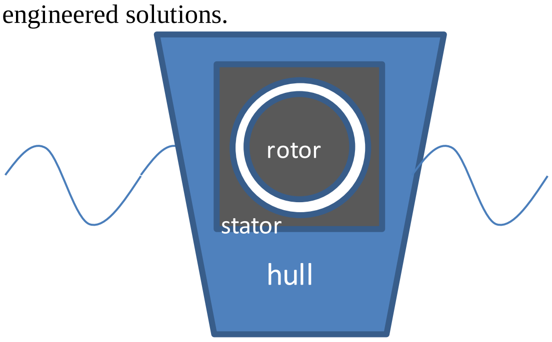

Figure 1: Pitching hull for case study 1

This case study assumes a hull with similar dimensions to a ship – in the region of 100m by 30m, with the capability to ballast in order to meet the desired motion characteristics. In wave energy terminology this is classified as a *terminator* device, with the ballast alteration being *slow tuning*. A schematic of this case study is shown in Figure 1. Power take off is from rolling of the hull and the electrical machine consists of a stator rigidly fixed within the hull, encompassing a rotor. The anticipated moment of inertia of the hull is such that its frequency of oscillation will change slowly over time.

Typical North-Atlantic wave-length is approximately 130m with a period of 9 seconds. To approximately follow the wave slope an object should be about one quarter of this (32.5m). Modern ships have a typical beam close to 32.5m and a natural roll period of about 9 seconds; making resonance mating realistic. Typical dimensions of a ship of this beam give a displacement of around 14,000 tonnes. In waves of 3m height, the roll angle will be approximately 8 degrees off vertical, more if resonance is achieved. Energy contained in this sea state is approximately 40kW/m wave front, equivalent to 5MW incident over the length of the hull. The power take off would be perhaps rated at one third of this value. For reference, the Pelamis is 140m long with a total displaced mass of 700 tonnes.

As the hull is large it is assumed the power take off force is negligible compared to the wave force and so the oscillation of the hull is considered constant over a short number of wave cycles and unaffected by the power take off. To extract 1MW peak from a hull pitching +/-8 degrees in a 9 second period, a peak torque of 10 x 10^3 kNm (1 per unit) would need to be applied against a stationary body. This is the torque required assuming the rotor is coupled to a stationery inertial reference frame.

3.2 Modelling equations

Introducing a spring between the rotor and stator removes the need for an inertial reference frame and the equation of motion is now given by Equation (1).

$$\ddot{\theta}_2 = \frac{1}{J}\left[B_{pto}\left(\dot{\theta}_1 - \dot{\theta}_2\right) + k\left(\theta_1 - \theta_2\right)\right] \qquad (1)$$

Where J is the moment of inertia of machine rotor, B_{pto} is the damping coefficient of the damping, k is a spring force and θ_1 and θ_2 are angular position of the rotor and stator.

The resonant system can be used for amplitude amplification. For example the rated power can now be extracted by amplifying the relative displacement twelve times to +/- 100 degrees and reducing the peak torque to a more manageable 800 kNm. The magnitude of amplitude amplification is a function of the moment of inertia of the rotor (or any mass coupled to it), combined with the spring and damping coefficients. The damping coefficient was set to 0.005 of the unsprung value. Angular displacement of the system is shown in figure 2.

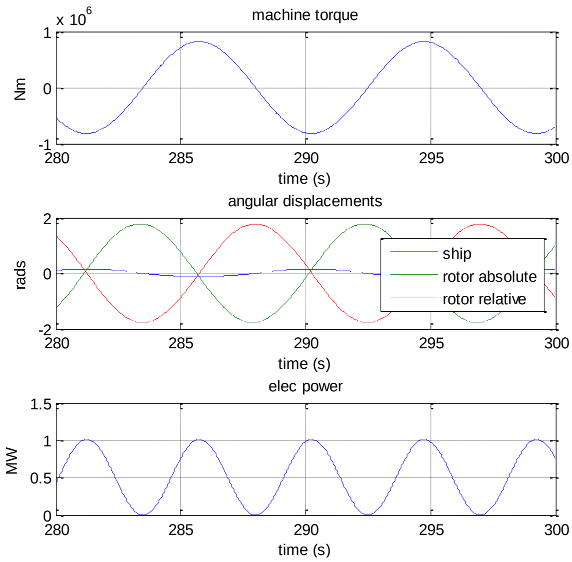

Figure 2 Resonating power take off

3.3 Amplification by integrated spring

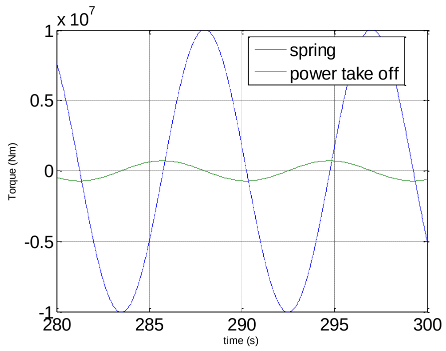

Figure 3: relative size of spring and power take off torque

Whilst this reduction in rated torque appears to be an attractive option, it is only achievable if a perfect spring is assumed. The spring force is shown in figure 3. The saving in machine torque comes at the expense of needing a large spring torque. The effect of varying the damping coefficient (per unit on the base value for the un sprung system) is shown in Figure 4.

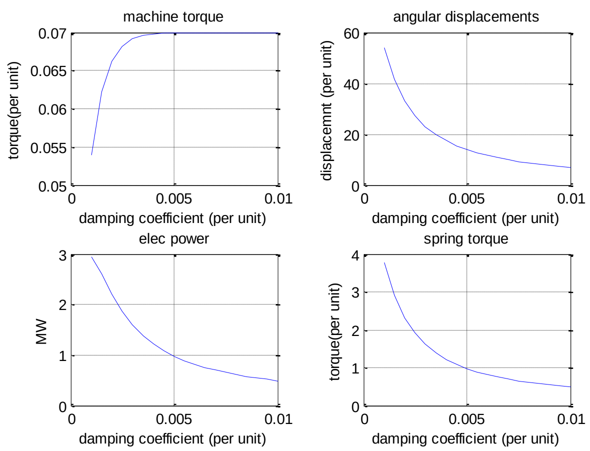

Figure 4: The affect of varying damping in resonant system

If the spring torque must also be provided by the electrical machine, this amplification only saves active machine cost for systems where the spring torque is less than 1 per unit. As shown in figure 4, this corresponds to systems where the output power is also less than 1 per unit. A similar study on varying the spring coefficient between +/- 3%, in an attempt to curtail spring force demonstrated that at any point off resonance, the maximum spring force required reduces quickly, but so too does the power captured. For this scenario, amplitude amplification can only result in a reduction in the peak machine torque if the spring force is externally applied.

4 Case Study 2: A Large Heaving Buoy coupled to a linear generator – excited at a single frequency

4.1 The device

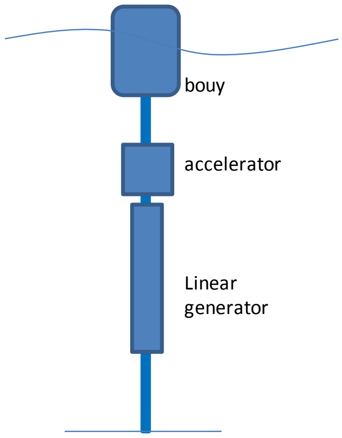

Figure 5: Heave buoy of case study 2

This case study investigates the use of a perfect mechanical amplifier. Consider a cylinder with a 12m diameter and 12m draft – as used by J Falnes in much of his work on resonating devices [1]. If it is excited at its resonant frequency (0.128Hz) by waves of 1m amplitude, it might be expected to yield 400kW with an amplitude of oscillation of say 5m. Assuming an instantaneous peak power of 700kW and PTO damping set equal to radiation damping, the peak force required by the linear generator is 160kN.

Figure 5 shows the heaving buoy coupled to the power take off via an ‘accelerator’. This could either be a magnetic gear [5], or a simple lever and is here assumed to be a 100% efficient system which amplifies displacement and divides required force by a gear ratio.

Extracting the same power from an increased amplitude of 10m halves the peak force required from the generator. In a rotary machine this would correspond to halving the required airgap area and hence active mass. In this case study a machine with inherent magnetic gearing is used to investigate alternative gear ratios in linear machines.

4.2 The Linear Electrical Machine

Several linear machine topologies have been proposed for wave energy, many of which capitalise on rare earth permanent magnets, such as the air cored topologies of [7,8]. Alternatively, some form of magnetic gearing is employed such as the transverse flux [9], flux switching [10], and the Vernier Hybrid [11] machines. The basic concept is that multiple small pitch magnets link a single coil, so that a small physical displacement is amplified to a high rate of change in magnetic flux. Investigation into linear machines for other applications have shown that topologies where the magnets are located on the stator give a less efficient magnet use on short stroke machines, are likely to prove favourable for long stroke machines [12].

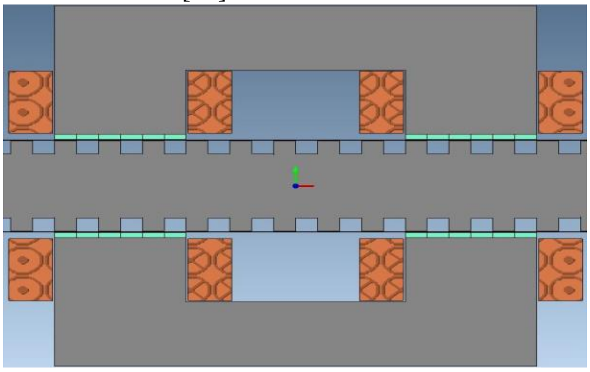

Figure 6: the Vernier Hydrid Machine

Initial reporting of the VHM for wave energy [11] has the magnets and exciter coils mounted on C-core stators surrounding a slotted steel translator, Figure 6. Translator pitch and stator magnet pole pitch are equal and maximum force is produced when the translator tooth is aligned with join between adjacent magnet poles – the q-axis. There are two C-cores for each phase. In the three phase machine, three 3 C-core units separated by 120 electrical degrees, mechanically separated by multiples of one third of the translator pitch.

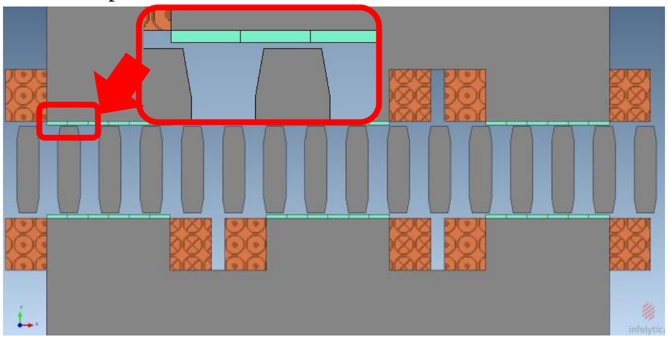

Figure 7: Improved performance vernier hybrid machine

Recent work by the authors has integrated the C-cores together to make two 3-phase E-cores that reduce the volume of the machine, Figure 7. Adjacent stator iron protrusions of the E-cores are 120 electrical degrees out of phase whereas neighbouring E-cores are 180 electrical degrees apart. It is possible to reduce the cogging and force ripple and thus improve the performance of the machine by independent variation of the magnet and tooth width. The design of a single three phase module of the improved E-core Vernier Hybrid Machine, consisting of a set of four cores compared to the original design of six cores is shown in table 1.

	C-core	*E-core*	
Magnet mass	2.6	1.3	kg
Active mass	820	740	kg
Force	120	80	N
Force ripple	5500	5460	N
Axial length	950	400	mm
Total MMF	100	100	A
tooth width	3000	3000	mm
Magnet width	12	10	mm

Table 1: Single module parameters of improved VHM

4.3 Linear Amplitude Amplification

Using this design, any rated force can be achieved by either scaling the axial length (axial scaling), or by increasing the number of E cores (modular scaling). For a fixed force, the volume of active material in the translator is constant regardless of the scaling method used, whereas the mass of the translator is linear with axial scaling. For minimum total mass, it is favourable to increase the modular length and decrease axial length. I.e. it is better to use numerous modular machines rather than a single axially long machine. Arbitrarily setting the minimum axial length to be 0.5m requires 5 units to react 160kN, with the parameters shown in the Table 2.

Amplification	*2*	*1*	
Force	80.0	160.0	kN
Amplitude	10.0	5.0	M
Units	3.0	6.0	
steel (translator)	1.22	0.94	tonne
steel (stator)	0.97	1.95	tonne
steel (total)	2.2	2.9	tonne
Copper	0.23	0.46	tonne
Magnet	20.1	40.2	Kg
axial length	0.5	0.5	M
active length	2.1	4.2	M

Table 2: Linear machine for alternative gear ratios

Also shown in the table are results for the amplitude amplified version. The active mass of the stator decreases and the active mass of the translator increases with amplification. In total, the amplified version gives a saving on all materials. Amplitude amplification is beneficial in this situation.

5 Case study 3: Archimedes Wave Swing excited by alternative frequencies

5.1 The device

This case study investigates the use of a variable pneumatic spring on device resonance. Figure 8 shows a simple representation of the Archimedes Wave Swing (AWS), a sub-surface wave energy device demonstrated with an all-electric power take off [13]. The AWS consists of an oscillating hood coupled to a linear generator. Within the hood, an air pocket acts as a variable pneumatic swing. It can be modelled in steady state by the mass-spring system, where M_a is a measure of the added mass (assumed here to be 0.6*dry mass), the mass of water affected by the oscillation. K_1 is the hydrodynamic stiffness, and C_1 is the radiation damping (function of frequency) and excitation force (varies with frequency). The power take off force can add to either of these forces.

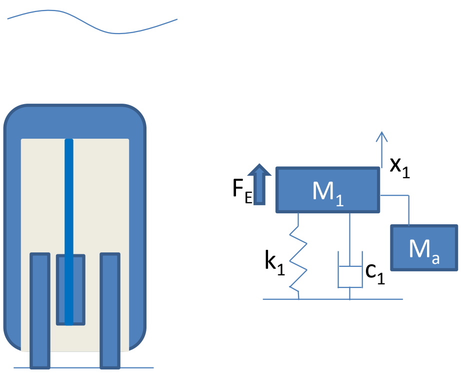

Figure 8: Case study 3 the AWS

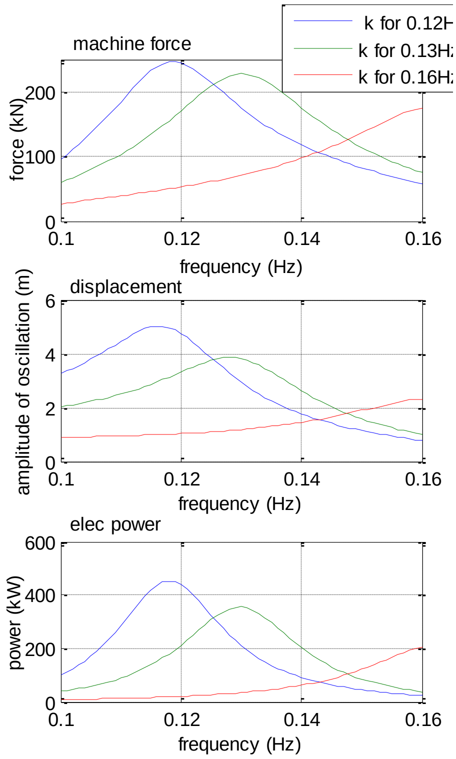

Figure 9: Unrestrained AWS results

5.2 Modelling with variable hydraulic spring

The most common wave spectrum for South Uist, using data described in [14], cited in [15] has an amplitude of 0.972m with time periods varying from 6.75-10.25 seconds, i.e. frequency 0.09 to 0.14. Using dimensions, mass, natural stiffness, excitation and damping coefficients from [13], the unrestrained response to this amplitude of wave is shown in Figure 9. Three values of stiffness, corresponding to device resonance at three frequencies are compared.

The advantage of variable resonance on yield is clear, and is well reported for various wave energy converters elsewhere. The average power of 450kW with the lower value of k_h corresponding to resonance at 0.12Hz is obtained with a force of 250kN and an oscillation of almost 5m, i.e. a peak to peak oscillation of 10m. Oscillation (and power) can be curtailed by altering the spring force or damping force within the AWS. In the deployed prototype, oscillation amplitude was limited by a combination of the generator damping and some additional water dampers over-damping the system. For the purposes of this design study with pure electric drive, the water dampers are ignored and amplitude is limited solely by over damping of the generator. The resulting yields are shown in Figure 10.

For the 0.12Hz device, the peak force required is now seen to be over 300kN for a peak power of 420kW. For devices with a low resonant frequency, amplitude limitation is likely to be necessary and this is shown to require the peak power of the machine to be increased compared to the non restrained version.

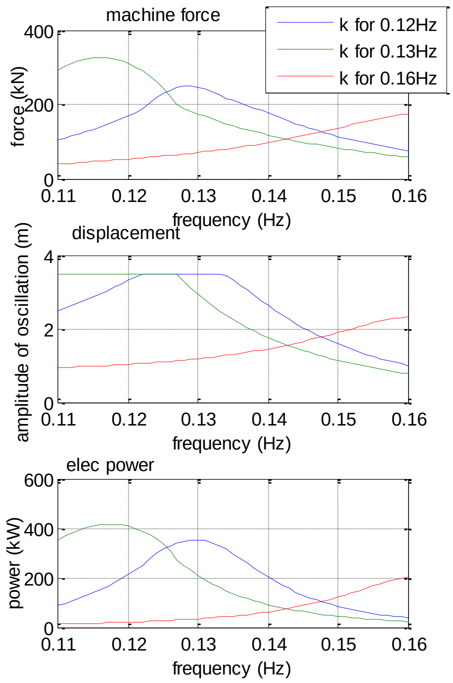

Figure 10: Restrained AWS results

5.3 Modelling with additional spring

Figure 11 shows a variation of the AWS, where an additional spring element and damper is added between the hood (M_1) and the translator (M_2).The dynamic behaviour can be described by Equations (2).

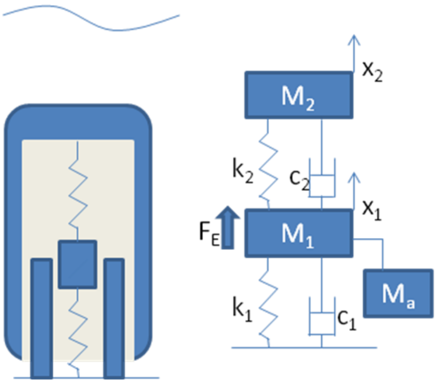

Figure 11: AWS with spring mounted power take off

$$\ddot{x}_2 = \left(C_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) - m_2 g\right)\frac{1}{m_2} \qquad (2)$$

$$\ddot{x}_1 = \left(F_E \sin \omega t - k_2(x_1 - x_2) - B_2(\dot{x}_1 - \dot{x}_2) - x_1 k_1 - \dot{x}_1 C_1 - m_1 g\right)\frac{1}{m_1 + m_a}$$

Steady state analysis of these equations, carried out in [15] for another wave energy converter provides the values of k_p and B_p for maximum power in any given sea state. Looking at k_h corresponding to 0.16Hz resonance and again limiting the generator amplitude (x_1-x_2) to 3.5m, the characteristics are shown in Figure 12.

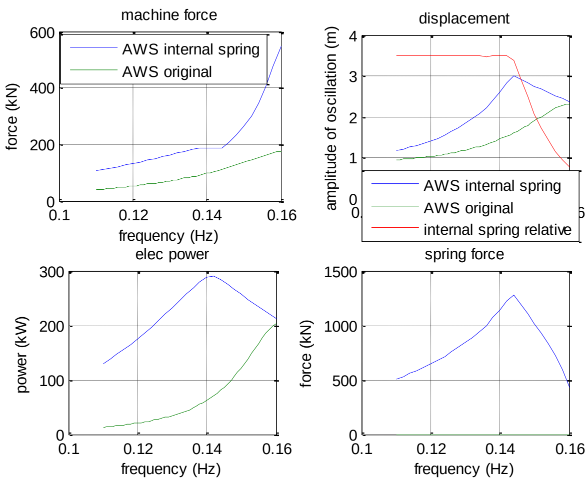

Figure 12: Results of spring mounted PTO AWS

Higher power can be obtained from the new model at high frequencies, yet as with the previous case study, the magnitude of the additional spring force is much larger than that required for power take off. It does not make sense to add the spring force by the electric power train in this case study therefore. It may be possible to capitalise on the resonance effect under some scenarios, and full wave to wire modelling will be used to do this over the course of the project. Below 0.14Hz, for example, (x_1 –x_2) is seen to be amplified, power is increased, and dynamic amplification is achieved. More accurate modelling would be required to investigate the validity of this result.

6 Conclusion

Three case studies have been presented to investigate some aspects of an all-electric drive train in wave energy converters. Case study 1 showed that adding the spring can induce resonance, increase velocity and reduce the force rating of the power take off. However, the spring force can be many times greater than the power take off force, and so if this is provided by the machine than the advantage of lower rated machine is offset. Case study 2 showed that amplitude amplification was advantageous even in linear machines, where the increased translator mass required to accommodate the increased oscillation amplitude can be offset by reducing the axial length of the machine due to the reduced force requirement. Case study 3 showed that a variable air spring will alter device yield and power take off requirements, yet if over damping is required to curtail oscillation this will dominate the machine requirements. Also, introducing an additional spring into the device can be used for amplitude amplification at some frequencies, but again magnitude of the spring force can offset any gains in machine ratings. It is clear that implementing spring or over damping control through all-electric power take off will not result in a reduced machine rating in all situations. However, in some scenarios, it is shown that amplitude amplification can allow all electric drive train development in wave energy converters. Full wave to wire models are required to investigate this fully.

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