Project Know How

PTO Dampers

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Abstract

In 2015, Aquamarine Power Limited (APL), developer of the Oyster 1 and Oyster 800 wave energy converters, went into administration and their intellectual property (IP) was acquired by Wave Energy Scotland Limited (WES).

The IP contained a significant body of information from exploratory small-scale tank tests completed during the company's trading history. This information exists across multiple internal documents which are not suitable for public publication.

This report presents the details of a range of Power Take-Off (PTO) dampers used for small-scale experimental wave tank WEC testing. The dampers discussed are those investigated, designed and used by Aquamarine Power and their research partners Queens University Belfast in the development of Oscillating Wave Surge Converter devices. Ten different systems are presented spanning a range of different damping mechanisms including: friction; hydraulic actuation; and electric/magnetic actuation. A detailed description and examples of the damping characteristics each system delivers are presented, along with its historical use by APL/QUB. The advantages and disadvantages of each system are also discussed, and a critical evaluation against key assessment metrics is presented in an attempt to identify a preferred system for future WEC development research.

Executive Summary

A damper mechanism is used in small scale experimental wave tank tests to replicate the behaviour of the Power Take-Off (PTO) system of a full-scale Wave Energy Convertor (WEC). During their trading history, Aquamarine Power Ltd (APL), in conjunction with their research partner Queens University Belfast (QUB), developed and tested many damper systems applicable to flap-type WECs. This document presents the details of the damper systems developed, along with a critical evaluation of its performance against key assessment metrics such as: cost; size; ease of use; waterproofing; ability to deliver required damping strategy; repeatability of damping; etc.

Ten PTO damper systems are presented and assessed in this report, which are:

1. Clutch Brakes
2. Disc Brakes
3. Linear water hydraulic piston (open loop)
4. Linear water hydraulic piston (closed loop)
5. Rotary water hydraulic system
6. Oil hydraulic system
7. Linear Motor
8. Electric Hysteresis Brake
9. Magnetic Particle Brake
10. Force Feedback Dynamometer (FFD)

A summary comparison of the advantages and disadvantages is made in Table 1 between all dampers in an attempt to identify a preferred system. However, it should be noted that such a comparison is dependent, and somewhat subjective, on the intended use and functional scope to which they were designed. Many of the design constraints may be unique to the requirements of APL in their quest to develop a large bottom-mounted flap-type WEC, namely Oyster. There is no guarantee that the same conclusion(s) can be applied directly if implemented on a different technology. Thus, the reader must use care when interpreting the results and recommendations presented in this report.

From Table 1 it can be seen that the Magnetic Particle Brake (MPB) system scores the highest across all the assessment criteria. Indeed, the positive experience of the research group in the use of this system would verify this conclusion as a preferred solution. However, this is under the caveat that a suitable water proofing solution must been developed independently. Such an innovative solution has been developed and successfully implemented by the QUB research group, making use of co-axial magnetic couplings, the details of which are given in Section 15.

A similar system based on an Electric Hysteresis Brake (EHB) was rank in second place to the MPB due only to a higher cost and large geometric size, making integration into a small-scale flap model more difficult. Again, this system is also dependent on the waterproofing solution of Section 15.

The third highest ranked damper system is arguably the Disc Brake system. It has many attractive features such as: low cost; inherently waterproof; very wide damping torque range. This system was the most extensively used by APL in the design, development and performance testing of the Oyster800 WEC. However, poor damping repeatability and high equipment maintenance (short lifespan) resulted in a lower ranking. Despite these shortcomings APL demonstrated that, once correctly set-up, it has the ability to deliver quality results from small scale wave tank tests.

Specification	Clutch Brake	Disc Brake	Open H_20 Hydraulic	Closed H_20 Hydraulic	Rotary Hydraulic	Oil Hydraulic	Linear Motor	Electric Hysteresis Brake	Magnetic Particle Brake	Force Feedback Dynamometer
Damping Actuation	Friction	Friction	Hydraulic	Hydraulic	Hydraulic	Hydraulic	Electrical	Electrical	Electrical	Electrical
Geometric Config.	Rotary	Rotary	Linear	Linear	Rotary	Linear	Linear	Rotary	Rotary	Rotary
Damping Profile	Constant	Constant	Approx. Constant	Quadratic	Approx. Constant	Constant	Other	Constant	Constant	Constant
Type of Control	Manual	Manual	Manual	Manual	Manual	Automatic	Automatic	Automatic	Automatic	Automatic
Max Damping [Nm]	1.3	∞	20	30	1	10	5.2	2.6	3.95	66
Min Damping [Nm]	0.2	0	0.75	1	0.1	1.7	0.2	0.04	0.07	<0.1
Assessment Criteria										
Repeatability	Poor	Poor	Poor	Poor	Poor	Good	Good	Good	Good	Good
Ease of Use/Control	Poor	Medium	Poor	Poor	Poor	Good	Medium	Good	Good	Good
Control outside tank (Y/N)	N	Y	N	Y	Y	Y	Y	Y	Y	Y
Dynamic Control (Y/N)	N	Y-manual	N	Y-manual	N	Y	Y	Y	Y	Y
Ease of use under water	Good	Good	Good	Good	Good	Medium	Poor	Poor	Poor	Poor
Lifespan / robustness	Poor	Poor	Good	Good	Poor	Good	Good	Good	Good	Good
Cost (approximate)	£10	£100	£100	£100	£100	£19,000	£3,000	£650	£350	> £20,000
Relative Size (comparison)	Small	Medium	Large	Large	Small	Large	Large	Medium	Small	Large
Actual Size (main component, e.g. Ø, stroke, etc.)	26 mm	140 mm	200 mm	200 mm	50 mm	270mm	300 mm	113 mm	86 mm	250 mm

Key:

Good
Medium
Poor
N/A

Table 1. Evaluation of 10 PTO damper systems developed and tested by the wider APL-QUB research group.

Table of Contents

1 Purpose

This document presents the details of the Power Take-Off (PTO) dampers developed by Aquamarine Power Ltd. (APL), in association with their research partners Queens University Belfast (QUB). PTO dampers are used in small scale experimental wave tank tests to replicate the behaviour and effect the full-scale power conversion system has on a Wave Energy Converter (WEC). During its trading history (2006 – 2015) APL accumulated a wealth of valuable knowledge and experience in experimental wave tank testing and physical modelling through its extensive R&D activities. Wave Energy Scotland (WES) has since taken ownership of the Intellectual Property assets of the company and wish to capture and disseminate this knowledge to the wider wave energy industry so that the experience and learning of APL can be utilised and built upon by others.

2 Background Information

Aquamarine Power Ltd was a wave energy company who developed a wave energy converter called Oyster. The Oyster system consisted of a WEC located in shallow water close to the shore, with a bottom-hinged, surface-piercing flap which oscillated due to wave action. Double acting pistons on each side of the WEC pumped water through a high-pressure pipeline back to shore, where high pressure water drove a Pelton wheel turbine connected to an electrical generator. The flow from the Pelton wheel discharged to a header tank and returned to the WEC via a low-pressure return pipeline.

APL deployed a full-scale 315 kW Oyster 1 system at the European Marine Energy Centre (EMEC) in August 2009, followed by a second-generation machine rated at 800kW, Oyster800, in August 2011.

APL ceased trading in November 2015 and the intellectual property was acquired by WES.

APL had a longstanding relationship with Queen's University Belfast (QUB) and the core of the APL R&D team were based permanently at the university's experimental wave tank facility in Belfast. Together they conducted extensive research into Oscillating Wave Surge Converter (OWSC) technology, in particular the Oyster device, which belongs to this classification of WEC. Physical modelling and experimental wave tank testing was the primary research technique which they used. As such, valuable knowledge and experience was acquired by APL on the physical modelling of the PTO system at small scale. A wide range of PTO damper systems were developed and tested throughout company's research activities.

3 Prerequisite Information

The damper systems presented in this report were designed under a series of constraints and functional specifications. Many of these were unique to the requirements of APL in their quest to develop a large bottom-mounted flap-type WEC, namely Oyster. There is no guarantee that the same conclusions or constraints can or should be applied directly to different technologies and so, care must be taken by the reader when interpreting the result for different applications. The key specifications which influenced the design of the PTO damper systems developed by APL were:

1. A constant damping[1] torque profile was sought (often referred to as Coulomb damping)
2. The damper hardware must be integrated with a hinge-line torque-tube sensor.
3. Ideally, the entire flap-PTO system is connected to the tank floor via a load cell (or multiple load cell) arrangement to measure the foundation reaction loads.
4. The scale of the wave tank tests and range of wave heights considered dictate the size, geometry and torque capacity of the damper.

These high-level specifications are discussed separately in more detail in the following sub-sections.

3.1 Constant Damping Torque

The PTO system on the full-scale Oyster device(s) is a closed-loop, pressurised water hydraulic system consisting of two doubling acting linear cylinders/pistons, a circuit of flow-rectifying valves and a serious of high- and low-pressure accumulators. The system configuration and interface geometry with the flap is such that it delivers an approximately constant amplitude torque to the pitching motion of the device. Figure 1 shows an example of the damping torque recorded from the Oyster800 device during full-scale ocean trials in 2014. This illustrates the characteristic 'square-wave' torque profile of the PTO system. (It can be seen however that there is a directional bias in the magnitude of the damping torque (positive torque is larger) due to the difference in active area between the bore and annulus side of the double-acting hydraulic pistons). Thus, for APL's wave tank R&D activities, constant damping was sought which best replicates the full-scale system. (Note: this does not suggest that this is the optimal damping strategy for this type of WEC concept).

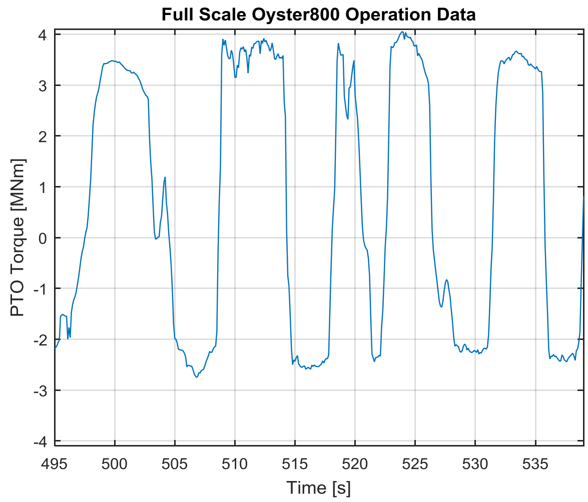


Figure 1. Damping torque profile of the full-scale Oyster800 PTO system recorded during ocean trials in 2014

3.2 Integration with a Torque-Tube Sensor

[1] Mathematical definition of an ideal constant damping strategy is given in the Appendix.

In wave tank testing, the hydrodynamic power capture of a pitch flap-type WEC is the product of the flap's angular velocity and resistive PTO torque experienced about the hinge line of the device. In order to reduce error in this critical metric, direct measurement of the torque induced on the hinge of the device is desired. Directly measuring the torque on the hinge also has the added advantage of being independent of the type of damper system implemented and so, different dampers can be integrated/exchanged into the same core flap model design.

QUB and APL developed a robust torque-tube design in-house which satisfies the requirement to directly measure the hinge torque. It is a thin-walled, strain-gauged design which measures the twisting force between the PTO damper system (which is attached to the shaft which acts as the pivot point for the flap) and the housing which is attached to the flap (and also acts as a water proof barrier for the strain gauges). Figure 2 shows a schematic diagram of the torque tube design.

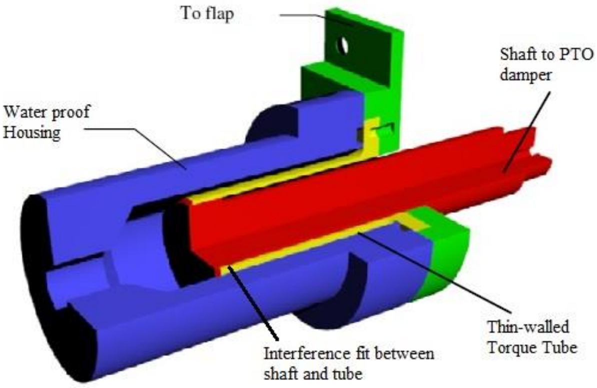


Figure 2. Torque Tube Design

Almost all of the PTO damper systems used by APL (and QUB) were designed such that they could be integrated with such a torque tube system, ensuring that the tube was the load path route between the PTO mechanism and the pitching flap. More detailed discussions of torque tube design and general physical scale models design criteria can be found in [R1] and [R2].

3.3 Integration with a 6 degree-of-freedom load cell

When performing wave tank tests, it is usually desired that both the power capture and loading characteristics of the WEC model are simultaneously recorded. For a bottom-mounted flap-type WEC such as Oyster, the anchoring forces between the flap and the seabed (tank floor) is the primary loading information. When testing in a wave tank, the scale model (including all of the PTO damper equipment) must be design so that it 'sits on top' of a 6 degree-of-freedom (dof) load cell (or equivalent sensor(s)) so that the load cell is the sole load path to ground. This requirement can have a significant influence on the size, weight and geometric configuration of the PTO damper system.

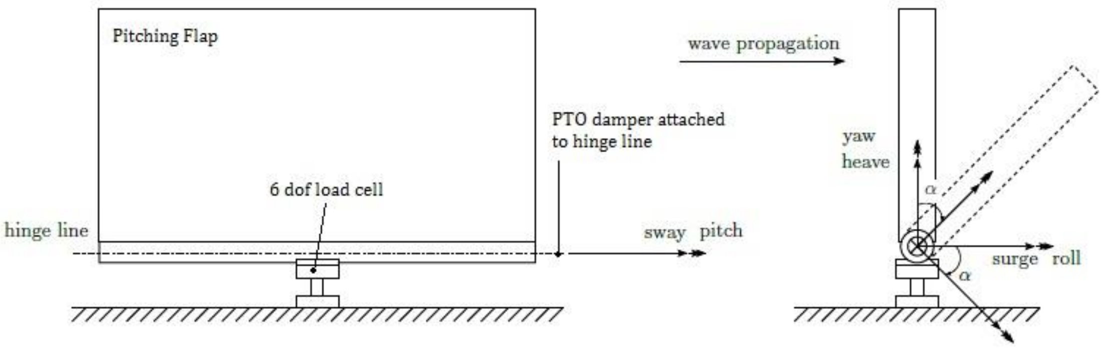


Figure 3. A simplified schematic diagram of the 6 dof load cell attachment point on a wave tank flap model.

3.4 Torque Capacity

In order to design an efficient damper system and specify the necessary components, the required torque range must be well established. However, for small scale wave tank tests this torque range is heavily influenced by the scale at which the tests are being conducted. Froude scaling laws specifies that torque scales as the fourth power of the scale factor. For example, on the full scale Oyster800 device the hydraulic PTO system can deliver a maximum Root-Mean-Square (RMS) torque of circa 8MNm (which is capped by a pressure relief system). For 40th scale wave tank tests, this limit corresponds to circa 3Nm while at 25th scale the same limit is over 20Nm. Thus, the PTO damper design and equipment required for each of these scales would be significantly different.

Conversely, in small scale testing, inherent friction in the equipment and model configuration may inhibit a sufficiently low level of PTO damped to be achieved. Thus, the choice of equipment may continuously overdamp the model if testing at smaller scales. It is also worth noting that because a 'square-wave' or constant torque profile was sought by APL, the RMS measure of the torque signal is close to the instantaneous torque amplitude. This is not necessarily true for other types of damping strategies. Unless otherwise stated, any torque limit or range specified in this report is referring to the RMS measure.

APL performed tests at (and thus designed dampers for) scales ranging from 40th scale to 20th scale and so, different scales required different equipment and different geometric configuration. Where possible/applicable, the details of this will be discussed in the subsequent sections of this report.

4 Report Scope and Structure

Ten PTO damper systems have been identified for discussion in this report, each of which is based on a different physical technique/mechanism. The damper systems are:

1. Clutch Brakes 2. Disc Brakes	**Friction**
3. Linear water hydraulic piston (open loop) 4. Linear water hydraulic piston (closed loop) 5. Rotary water hydraulic system 6. Oil hydraulic system	**Hydraulic Actuation**
7. Linear Motor 8. Electric Hysteresis Brake 9. Magnetic Particle Brake 10. Force Feedback Dynamometer (FFD)	**Electrical Actuation**

Each of these dampers are discussed separately in the subsequent sections of this report. The information provided on each is structured as follows:

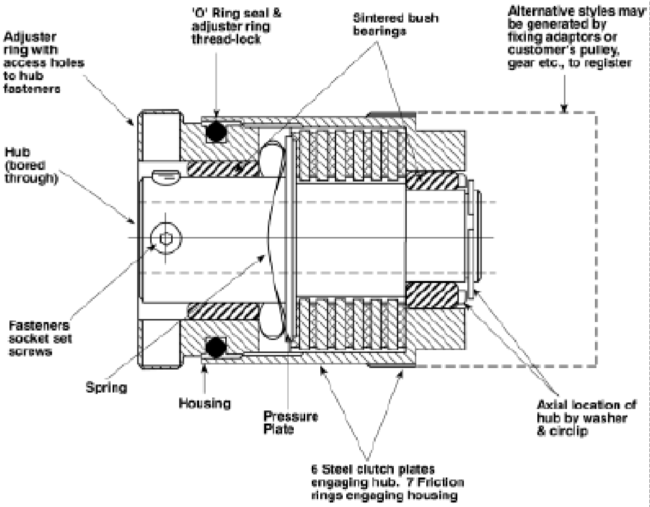
- Description of the damper system and historic use
- Example of the torque characteristics delivered
- Advantages and Disadvantages
- System evaluation

A summary comparison of the pro’s and con’s is made (in the Executive Summary) between all dampers in an attempt to identify a preferred system. However, it should be noted that such a comparison is dependent on the intended use and functional scope (i.e. use on a small-scale flap-type WEC device, in the instance of APL). There is no guarantee that the same conclusion(s) can be applied directly if implemented on a different technology. Thus, the reader must use care when interpreting the results and recommendations presented in this report.

5 Clutch Brake

5.1 Description & Use

The clutch brake is a unit which applies damping via direct friction. The model used by APL (see Figure 4) has six plates which can be forced together to increase the damping torque provided. By design it induces Coulomb damping, which produces a square wave damping time-trace profile.



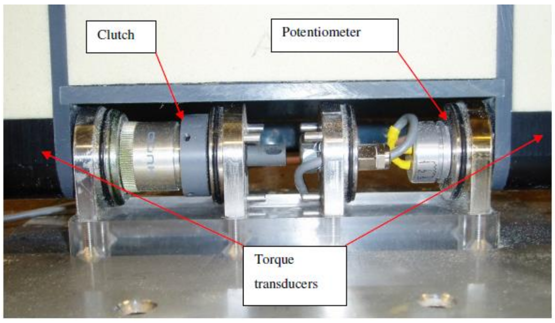


Figure 4. Schematic diagram (left) of the clutch brake mechanism used. (right) One example of how the clutch was integrated along the hinge-line of the flap model. Torque tubes are installed outside and in-line with the clutch and the flap is attached directly above the system.

This damper system was designed and first used by QUB (details of which can be found in [R1]) for 40th scale model tests. APL subsequently adopted this methodology in their early research investigations. Two clutch brakes were easily integrated into the flap model to increase the range of damping torque applied to the flap. However, as APL's research evolved to test wider and wider flaps, the clutch brakes could not provide a sufficiently high level of damping torque. Empirically, the maximum damping torque delivered by 1 clutch was found to be circa 1.3Nm.

APL used the clutch brake system for detailed WEC performance testing of the Oyster1 design (2008-2009). Following on, this system was also used for the early exploratory landscaping research of the second-generation device, namely Oyster800, circa 2009-2010. However, before the Oyster800 concept was defined, the clutches were replaced by a disc-brake system (see Section 6) which delivered the detailed performance and loading testing of this device design.

5.2 Torque Trace Profile

Figure 5 shows an example of the square-wave damping torque profile from a newly installed clutch brake system. Clutch brakes however are really only designed to operate in one direction. It was found that the oscillatory motion of the flap quickly wears out the friction plate mechanism. This causes a looseness in the damper called 'backlash' where, on motion reversal, the flap is allowed to move freely for a period without any damping torque being applied. This effect is uncontrolled and continues to degenerate (affecting test repeatability) and actually alters the entire damping strategy. Figure 6 shows an example of the backlash effect.

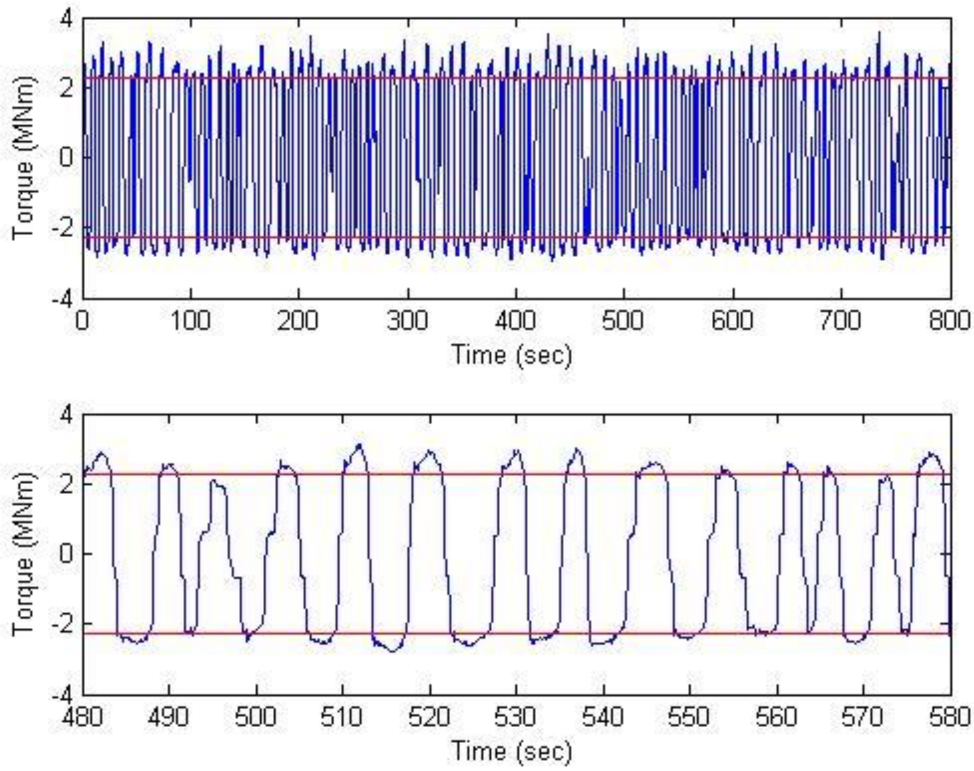


Figure 5. Time-trace example of the PTO damping applied from a fully functioning clutch brake system used on a 40th scale flap model. The red lines indicate the magnitude of the RMS damping. (All values shown have been scaled up to full scale units).

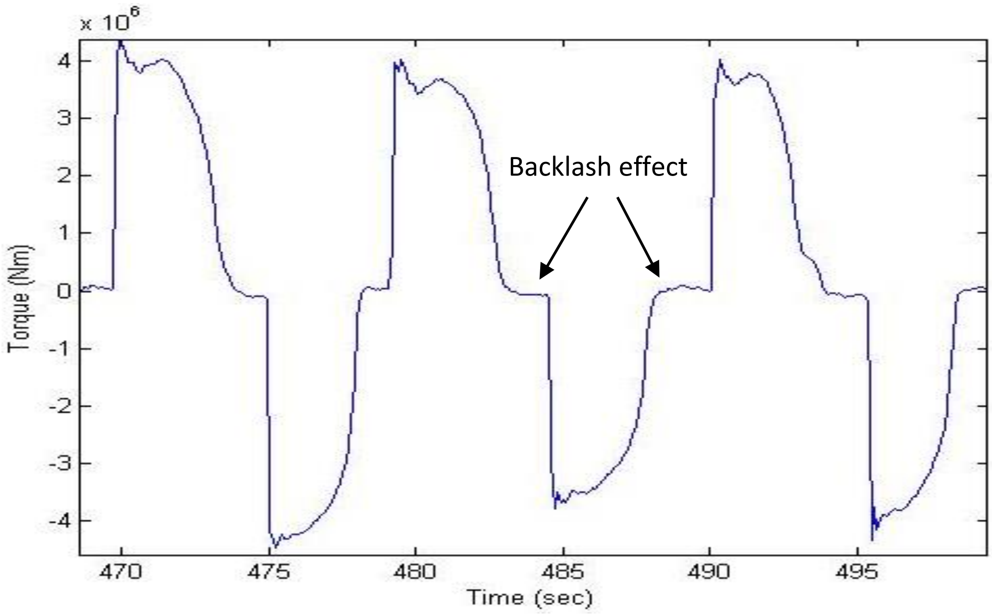


Figure 6. The effect of backlash of the damping torque characteristic profile.

5.3 Advantage and Disadvantages

Advantages

i. Can be directly used underwater.
ii. Very cheap.
iii. Small in size and weight and relative easy to integrate in a model design.
iv. Produces a square-wave torque profile.
v. Applies a direct torsional damping to the flap. This reduces the complexity of the geometry interface with the flap.

Disadvantages

i. Difficult to adjust. Requires a person to manually screw the clutch to change the damping level.
ii. Relatively small maximum damping torque capacity, ~1.3Nm per clutch.
iii. Wears out quickly (~1-2 weeks of use).
iv. Increased (and uncontrolled) backlash effects as clutch starts to wear.
v. Accurate test repeatability is difficult/impossible.

5.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Friction
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Manual
Max Damping [Nm]	RMS level achieved experimentally	1.3Nm
Min Damping [Nm]	RMS level achieved experimentally	0.2Nm

Assessment Criteria	Score
Repeatability	Poor
Ease of Use/Control	Poor
Control outside tank (Y/N)	N
Dynamic Control (Y/N)	N
Ease of use under water	Good
Lifespan/robustness	Poor
Cost (approximate)	£10
Size (outer diameter)	26 mm

Key:

Good
Medium
Poor
N/A

Table 2. Specification and assessment of the Clutch Brake damper system

6 Disc Brake

6.1 Description & Use

An evolution of the clutch brake damper system (Section 5) was to use disc brakes as the mechanism to apply direct Coulomb friction to the moving flap. The brakes used by APL were conventional mountain bike disc brake-callipers. The discs were rigidly attached to each side of the rotating flap (on the hinge line/bottom tube) and the brake callipers independently attached to a baseplate, all of which was mounted on top of a 6 dof load cell, anchored to the tank floor. Figure 8 shows how the disc brake damper system was integrated into a flap model and an example of a disc and brake calliper used by APL is shown in Figure 7.

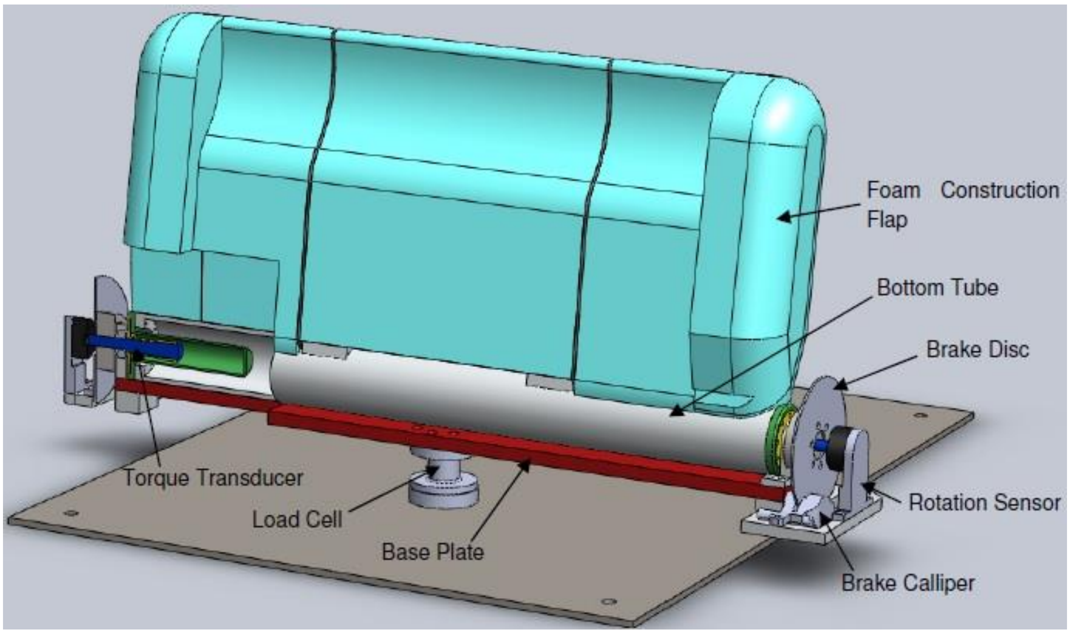


Figure 8. APL's disc-brake damper system and integration with a flap model.

The damping/friction is adjusted by tightening/loosening the brake pads (within the calliper) on either side of the rotating disc. This is done via a lever (a typical bicycle brake handle) suspended/mounted above the water level, which is attached to the calliper by a hydraulic hose. For ease of user access, the lever was connected to a point outside the tank and a simple mass-pulley system used to control the lever during tests, as shown in Figure 9. APL's preferred supplier of the brake discs was Avid.

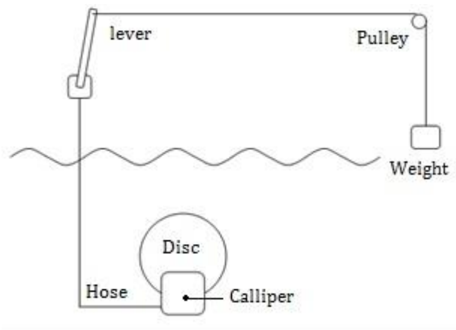


Figure 9. Schematic diagram of the pulley-weight system used to control the damping applied by the disc brakes

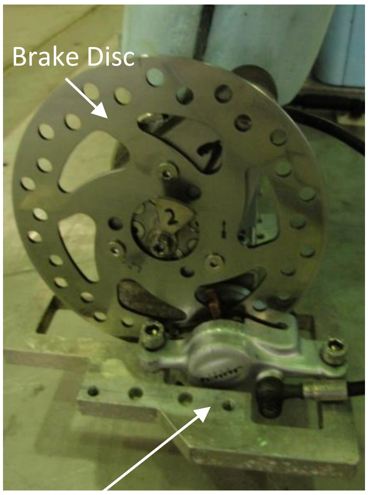


Figure 7. Example of a disc and brake calliper used by APL.

The disc brake damper system was used extensively by APL and QUB on a range of different wave tank tests from 40^{th} to 20^{th} scale models and on a variety of different flap-type WEC concepts and exploratory research activities. This system was used in 2010-2011 for scale model tests which informed the detail design and performance characteristics of the full scale Oyster800 device. This system continued to be the preferred damper of the APL research group for wave tank testing right up until the company ceased trading in 2015.

This damper system can span the entire torque range spectrum required from zero torque (the callipers do not touch the disc at all) to a torque large enough to stop the flap moving altogether (stopping torque) in performance sea states, at all scales considered by APL. This equates to torque magnitudes in excess of 130Nm. In addition, the extensive use by the APL/QUB research group was also promoted by the fact it inherently provided an approximate constant or square-wave coulomb damping, was easily integrated with the existing torque tube design and was sufficient small and light weight.

6.2 Torque Trace Profile

Figure 10 shows an example of the damping torque profile from the disc brake system used on a 40th scale model test. It can be seen that in general, this damper system delivers an approximate square-wave torque profile. However, torque spikes are also evident, examples of which occur at 25.5s, 30s and 33s in Figure 10. It was found that the discs actually wore/polished in a localised swept area about which the flap typically oscillated. If, however, in an isolated motion the flap moved beyond this typical range, the brake pads would encounter a rougher portion of the disc. This rapidly increased the instantaneous torque applied, thus manifesting as a torque spike in the time trace. The onset of this uneven wear/polish can happen quite rapidly within ~1 weeks of use. However, the discs could be detached, rotated and re-attached to use an untouched part of the disc to extend the working lifetime of the disc equipment.

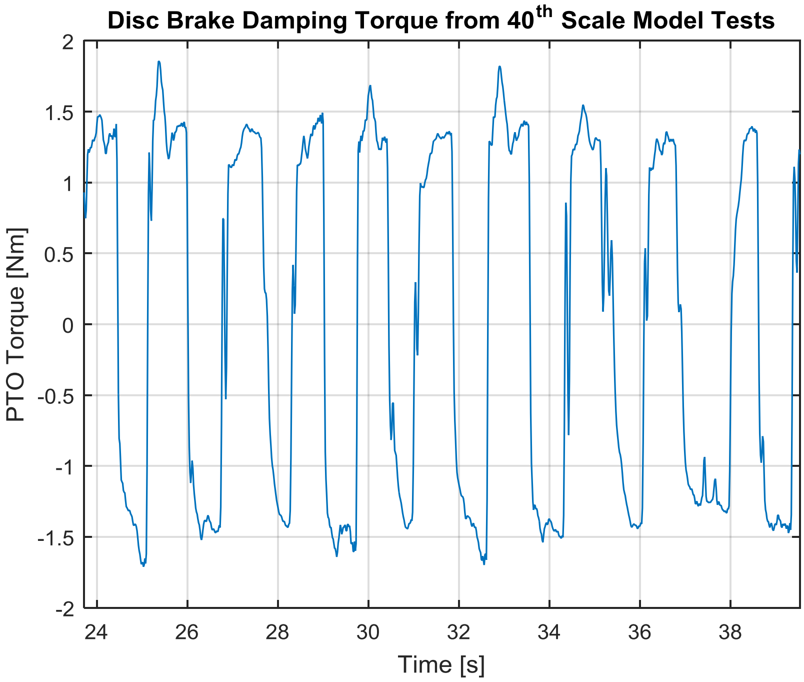


Figure 10. Damping torque profile from the disc brake damping system on a typical 40th scale model test. Square-wave characteristics are exhibited and the torque spikes are a result of wear/polish on the discs themselves after some use (~1-2 weeks).

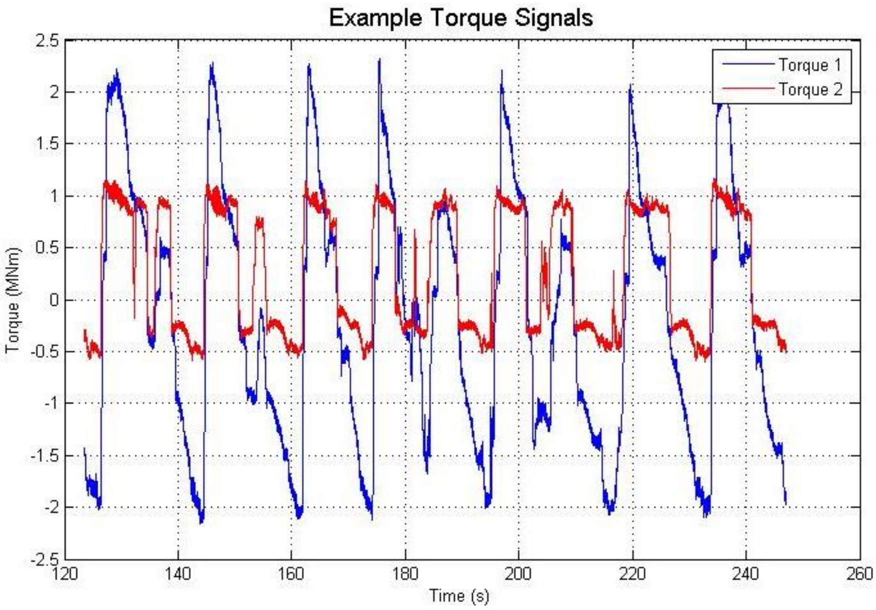


Figure 11. Examples of different damping levels on either side of the flap. Disc wear effects shown in blue.

A more severe example of disc wear/polish (torque spikes) is shown in Figure 11 in the blue torque signal. The two different signals represent a brake-calliper system attached at either end of the flap model (as shown in Figure 8). A thorough study of the disc wear/polish issue was conduced by APL and the results compiled in an internal report [R4], the reader is referred to this for more detail.

Due to the fact that each brake is controlled independently and manually, it is often laboursome to get both braking magnitudes the same during tests, as illustrated by the red and blue signals in Figure 11. In terms of power performance testing however, this does not have any effect as both torque signals are added together to get the total damping torque on the flap before mechanical power is calculated. However, if there is a large asymmetry in the damping torque applied, this could manifest as an additional twisting load on the strucutre. In APLs case, this would reuslt in a yaw load on the foundations and a racking force across the structure of the flap.

6.3 Advantage and Disadvantages

Advantages

i. Can be directly used underwater.
ii. Cheap & readily available off-the-shelf components from any good bike store.
iii. Relatively small in size and weight and easy to integrate in a model design.
iv. Can produce a square-wave torque profile.
v. Once the model is set up, the damping level is easy to adjust, albeit manually.
vi. Full torque range achievable from 0Nm to stopping torque.
vii. Applies a direct torsional damping to the flap, reducing complexity of the geometry interface.

Disadvantages

i. Uneven wear/polish of the disc can distort the damping profile applied.
ii. The onset of disc wear can happen quite readily, circa 1 week of use.
iii. Disc wear and manual adjustment of calliper 'tightness' makes accurate tests repeatability difficult/impossible.

6.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Friction
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Manual
Max Damping [Nm]	RMS level achieved experimentally	Stopping Torque †
Min Damping [Nm]	RMS level achieved experimentally	0 Nm

Assessment Criteria	Score
Repeatability	Poor
Ease of Use/Control	Medium
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	Y (manual)
Ease of use under water	Good
Lifespan / robustness	Poor
Cost (approximate)	£100
Size (typical disc diameter)	140 mm

Key:

Good
Medium
Poor
N/A

Table 3. Specification and assessment of the Disc Brake damper system

† *Stopping torque is the magnitude of damping required to stop the flap moving in the waves. For the type of scales APL tested, stopping torques of in excess of 130Nm were achieved with this system.*

7 Linear Water Hydraulic Piston (open loop)

7.1 Description & Use

The open loop hydraulic system consists of two double-acting hydraulic cylinders, similar in design to those installed on the full-scale Oyster device. An example of one such cylinder used by the APL/QUB research group is shown Figure 12, attached to a 20th scale model of the Oyster1 device. The hydraulic circuit, shown in Figure 13, is open so that water is drawn from the wave tank into each cylinder through a non-return valve. This water is then pumped though a throttle, through a second oppositely configured valve and returned to the tank. The damping level is set by manually adjusting the opening of the throttles, which involves physically getting into the tank.

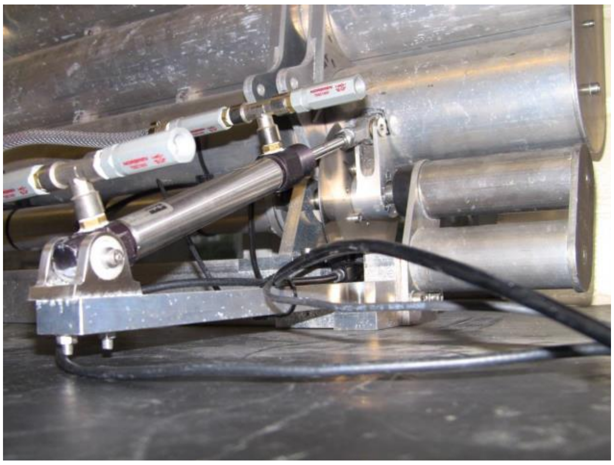


Figure 12. Double-acting hydraulic cylinder in the open loop hydraulic damper.

Due to the presence of the cylinder rod in chamber 2, the volume of water drawn into it (or active area of the annulus side of the cylinder head) is less than that in chamber 1. This results in a bias between the force applied to the flap when moving in the seaward and landward directions.

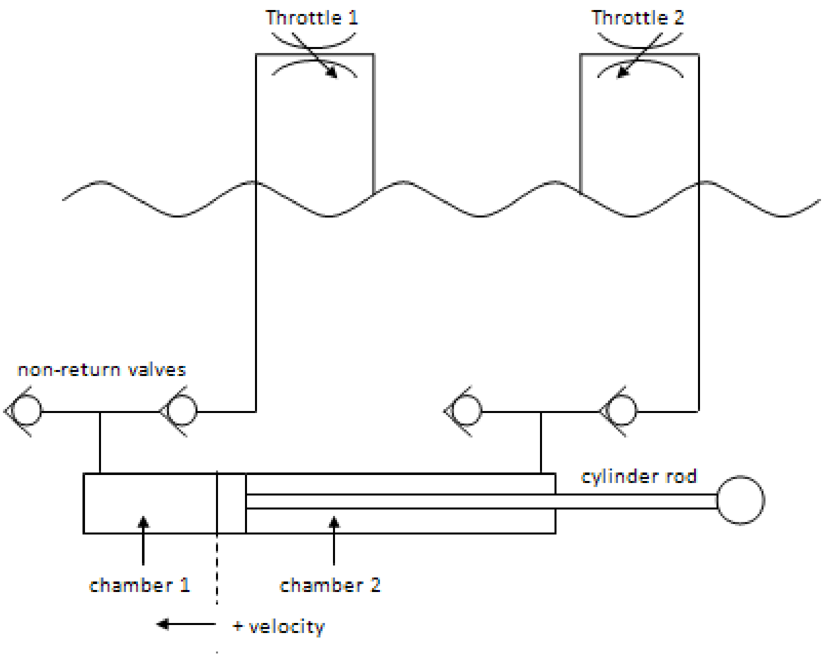


Figure 13. Hydraulic circuit for open-loop hydraulic damper.

This system was developed and used as part of a QUB PhD project which correlated the behaviour of the full scale Oyster1 prototype to 20th scale wave tank tests (2012). A detailed description of this project and the design of the open loop hydraulic damper system can be found in [R3]. In practise, it was not found to be that user friendly and this system was not tested beyond the remit of the PhD project. Thus, there is scope to develop this system further.

7.2 Torque Trace Profile

The characteristic of fluid flow through small pipes and orifices might suggest that the damping applied to the flap should be in some way be related to the flow velocity (i.e. velocity of the flap/cylinder head). Thus, it might be expected that the damping force is proportional to the flap velocity (linear damping) or proportional to the square of the flap velocity (quadratic damping). However, the torque profile measured from this system exhibited more characteristics of a square-wave, or constant damping profile (but with some distortion). Figure 14 shows a graph of damping torque verses the flap rotation angle. If perfect, constant damping was applied, this graph should have a square/rectangular profile. It can be seen that the system does indeed display this type of behaviour. The reason why a 'quadratic' damping (related to flow velocity) profile was not realised is because the throttle valves had to be almost completely closed to achieve the magnitude of damping required during the tests. This resulted in an accumulation of pressure in the cylinder chambers so that the damping was more related to this pressure rather than the flow/flap velocity.

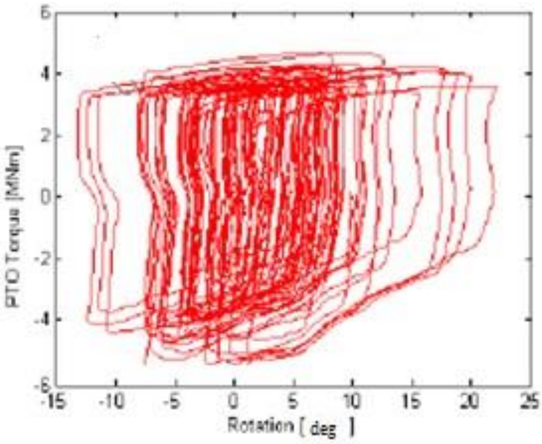


Figure 14. Torque-rotation graph illustrating the damping profile achieved with the open-loop damper. (Note: units in full scale)

It can be seen in Figure 14 that there is a distortion in the damping profile. This distortion on one half of the cycle comes from a combination of the difference in active fluid volume and pressure in each chamber and the fact that the effective lever arm of the cylinder-flap attachment also changes to allow the flap to move through its full range of rotation, (i.e. the cylinder itself move up and down as the flap pitches seaward and landward respectively). Further details of this are discussed in [R3]. Given the fact that this damper system applies a linear load (at a lever arm) to resist the motion of the flap, the interface geometry (i.e. the effective lever arm) can have a significant influence on the maximum magnitude of the damping torque supplied to the device.

7.3 Advantage and Disadvantages

Advantages

i. Can be directly used underwater.
ii. Cheap.
iii. Subject to the same geometric configuration and constraints as the full scale Oyster1 system (representative of full scale behaviour).
iv. Quite robust and is not subject to excessive wear.

Disadvantages

i. Difficult to categories the type of damping applied, (unique profile).
ii. Requires manual access to the tank to change the damping setting during tests.
iii. Test repeatability is very difficult.
iv. Linear load (not torsional) applied, creating a more complex geometric interface.
v. Larger mounting arrangement/footprint for the cylinders required, increase the complexity of attaching the model onto a load cell.

7.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Hydraulic
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Linear
Damping Profile	Constant/Linear/Quadratic/Other damping	Approx. Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Manual
Max Damping [Nm]	RMS level achieved experimentally (per cylinder)	20 Nm †
Min Damping [Nm]	RMS level achieved experimentally (per cylinder)	0.75 Nm †

Assessment Criteria	Score
Repeatability	Poor
Ease of Use/Control	Poor
Control outside tank (Y/N)	N
Dynamic Control (Y/N)	N
Ease of use under water	Good
Lifespan / robustness	Good
Cost (approximate)	£100
Size (Stator & stroke)	200 & 100 mm

Key:

Good
Medium
Poor
N/A

Table 4. Specification and assessment of the open-loop hydraulic damper system

† *Note: the magnitude of the damping reported is only that achieved during the PhD project [R3] with the component sizing and configuration implemented. This is very dependent on both cylinder and pipework diameters as well as the PTO-flap interface geometry (i.e. the effective lever arm). Thus, with further design and testing this torque range could be augmented.*

8 Linear Water Hydraulic Piston (closed loop)

8.1 Description & Use

Similar to the open loop linear hydraulic system (Section 7), the closed loop system consists of two double-acting hydraulic cylinders. See Figure 15 for another example of how the cylinders were applied by the QUB research group in model testing. In fact, due to size requirements the cylinders used are typically off-the-shelf pneumatic cylinders, but water is used as the medium instead of gas. The closed loop hydraulic circuit was developed in 2006-2008 under a QUB PhD project (see [R1]) on 20th scale model tests of flap-type WECs. The reader should refer to this thesis for more in-depth details on the motivations behind the design. Although used during this PhD project, APL never adopted this system for any testing or development of their Oyster device. The lessons learned from the PhD project highlighted that the system would not be appropriate for APLs requirements.

Figure 15. Double-acting hydraulic cylinder attached to a flap model.

Two variations of the close-looped system were developed. The first, shown in the circuit diagram in Figure 16, is where the motion of the flap pumps water from the cylinders through a combination of flexible and rigid tubing into a reservoir, located outside of the wave tank, via two throttle valves. The throttle valves have a parallel non-return valve so that the water is only restricted during its flow to the reservoir. The damping is controlled by manually adjusting the throttle values from outside of the tank.

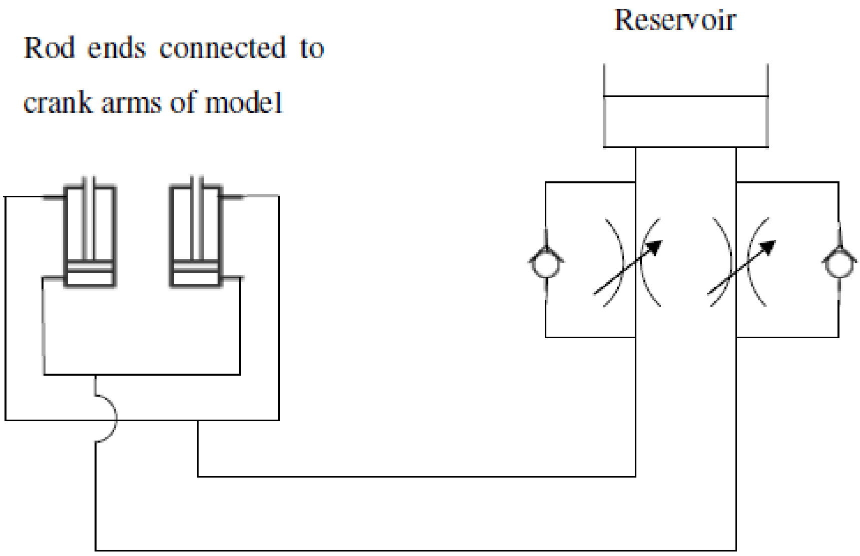


Figure 16. Hydraulic circuit of the first closed-loop hydraulic damper system.

The second system, shown in the circuit diagram of Figure 17, is a more complex arrangement where a pressure accumulator was added. In this system, a flow rectifying circuit is included so that flow is always pumped from the cylinders to the accumulator at high pressure and is always drawn back into the cylinders from the reservoir at low pressure. Conceptually, this is a similar pipework configuration to the full-scale PTO system which was installed on the Oyster device. However, it must be emphasised that flow characteristics in small pipes is not reflective of that in larger (i.e. full-scale) pipes. Thus, just because the pipework configuration is similar does in no way guarantee that the damping strategy provided on the scale model will be representative of the full-scale system. A throttle value between the accumulator and reservoir is used to control the level of damping applied and a pressure relief valve is used to limit pressure build up in the accumulator.

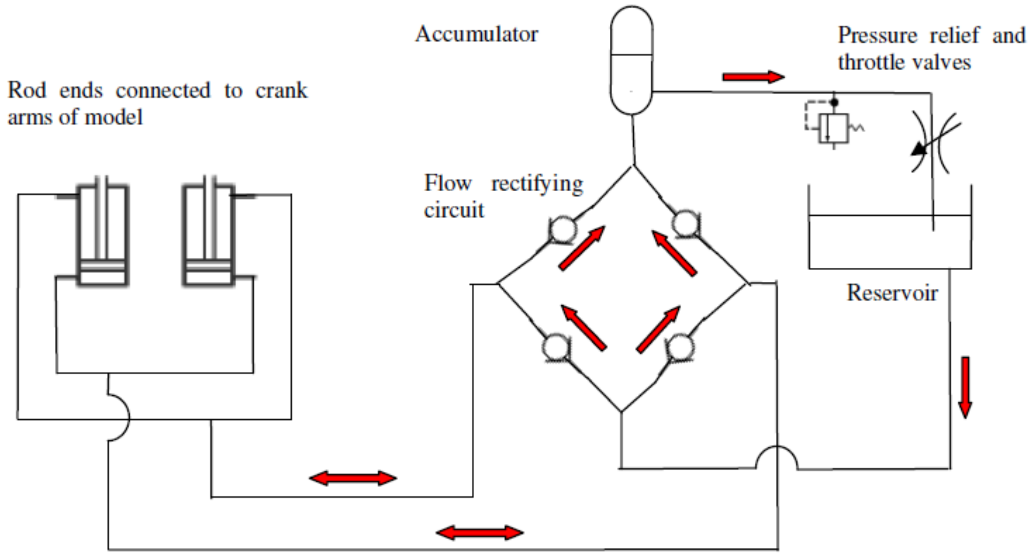


Figure 17. Hydraulic circuit of the second closed-loop hydraulic damper system

8.2 Torque Trace Profile

In the first configuration (see Figure 16) the damping is produced by turbulent flow restriction through the throttle valves. In this case, the damping provided is roughly proportional to the square of the flow velocity and thus is referred to as 'quadratic' damping. Figure 18 shows an example of the damping torque profile in monochromatic waves alongside the corresponding rotation of the flap. The phasing of the torque profile, relative to the rotation, indicates a close relationship to the flap velocity (and thus piston and flow velocities). It was concluded by [R1] that this damper system did closely approximate quadratic damping.

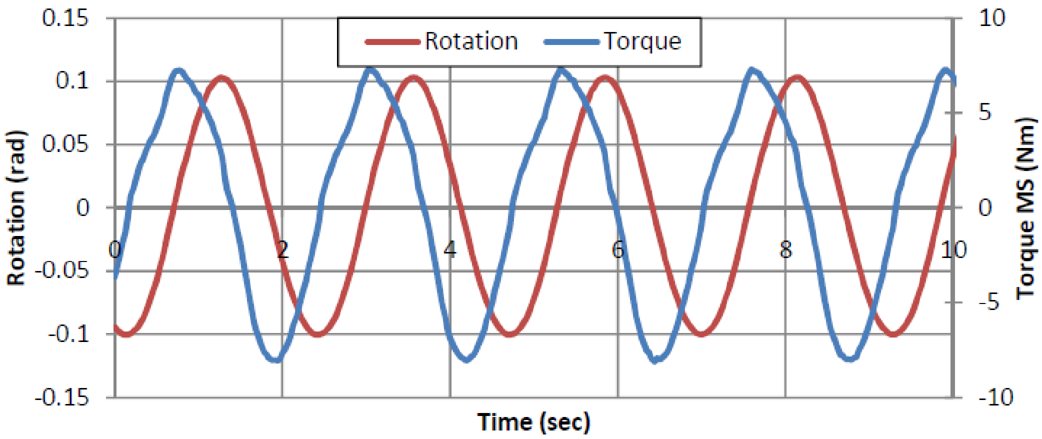


Figure 18. Quadratic damping induced by flow through the throttle values.

In the second closed loop system (see Figure 17), the inclusion of the accumulator was an attempt to keep a more constant pressure in the system and thus deliver a more square-wave or constant damping profile. However, as can be seen in Figure 19, although there was some modification of the signal it did not fully achieve a square-wave profile. It was found that the constant damping profile was still contaminated by quadratic effects created by flow losses (related to flow velocity) in the pipework and through the various valves in the circuit. In addition to this, it was difficult to control the pressure in the accumulator and it would fluctuate on a slow (circa 10 wave cycles) time scale, influenced by the groupiness of the irregular wave (and thus flap motion). However, the magnitude of this deficiency was due more by the component sizing and system design/assembly rather than a fundamental issue in the conceptual design. Although the flow through the throttle valve also increased (with increasing pressure) to partially stabilise the system, it was not effective on a wave-by-wave basis. Thus, the magnitude of the torque applied slowly modulates. However, it should be noted that this system was not extensive tested by [R1] and so could potentially be improved.

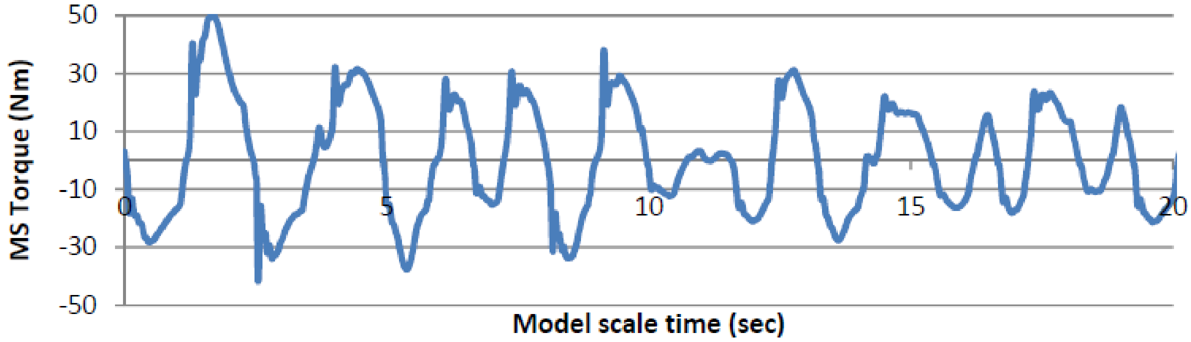


Figure 19. Damping profile (in irregular seas) of the close-loop system with an accumulator installed.

The damping torque range achievable by this system is very dependent on the component sizing (e.g. diameter of the cylinder and pipework) and geometric configuration. For example, similar to Section 18, given that fact that the damping is achieved by a linear force being applied to the flap at a lever arm, the interface geometry will significantly influence the magnitude achieved. For the given damper configuration and component sizing discussed in [R1], a minimum damping torque of 2Nm was reported due to the residual resistance from flow 'freely' circulating around the pipework. At the other end of the scale, a maximum RMS damping torque of circa 60Nm was achieved with this system. It should be noted however, that given the fact that the torque profile is not a square-wave, the instantaneous maximum can be almost twice as large as the RMS value. This system delivered an instantaneous maximum of almost 120Nm during testing.

The relationship between RMS and instantaneous torque values and the shape of the damping profile is a subtle but important factor when interpreting wave tank model test results. Often it is the RMS damping value which is used to assess the PTO system and make comparisons with a full scale WEC system. However, if the instantaneous torque required to achieve a target RMS value is very large (e.g. in the case of quadratic damping), this may not be physically achievable when scaled to full scale. Thus, WEC engineers must pay close attention to the shape of the damping profile, as well as just the magnitude of the damping delivered and account for any physical limitations or constraints that may exist in a full-scale system when interpreting wave tank test results.

8.3 Advantage and Disadvantages

Advantages

i. Can be directly used underwater.
ii. Cheap.
iii. Subject to the same geometric configuration and constraints as the full-scale Oyster system.
iv. Robust and is not subject to excessive wear.
v. Can modify the damping outside of the wave tank.

Disadvantages

i. Test repeatability is extremely difficult due to the manual adjustment of the valves.
ii. Difficult to set up and use, costing valuable staff and wave tank testing time.
iii. Frequent bleeding of air out of the system.
iv. Difficult to maintain a constant pressure in the accumulator
v. Applies a linear load (not torsional), increasing the complexity of the flap-geometry interface.
vi. Larger mounting arrangement/footprint for the cylinders required, increasing the complexity, or ruling out the possibility altogether, of attaching the model onto a load cell.

Disadvantages (ii), (iii) and (iv) applied specifically to the second system with an accumulator. All other points apply to both systems.

8.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Hydraulic
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Linear
Damping Profile	Constant/Linear/Quadratic/Other damping	Approx. Quadratic
Type of Control	How the damping level is controlled – Manual/Automatic	Manual
Max Damping [Nm]	RMS level achieved experimentally (per cylinder)	30 Nm†
Min Damping [Nm]	RMS level achieved experimentally (per cylinder)	1 Nm†

Assessment Criteria	Score
Repeatability	Poor
Ease of Use/Control	Poor
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	Y-manual
Ease of use under water	Good
Lifespan / robustness	Good
Cost (approximate)	£100
Size# (Stator & stroke)	200 & 100 mm

Key:

Good
Medium
Poor
N/A

Table 5. Specification and assessment of the closed-loop hydraulic damper system

† *Note: the magnitude of the damping reported is only that achieved by the QUB/APL group with the component sizing and configuration implemented. This is dependent on both cylinder and pipework diameters as well as the PTO-flap interface geometry. Thus, with further design this torque range could be augmented.*

 # Does not account for the auxiliary pipework and accumulator system.

9 Rotary Water Hydraulic Piston

9.1 Description & Use

This system works on the same principle as the hydraulic systems discussed in Sections 7 and 8 except, instead of using a linear piston to apply a force to the flap at a lever arm, a rotary vane pump is used to apply torsion directly on the hinge line of the device. This system is subject to the same options of using an open or closed hydraulic system as discussed previously. The rotary hydraulic actuator used, shown in Figure 20(b), was originally designed to be a pneumatic one. Water was used instead of air because of the former's lack of compressibility. A moving wall, called a 'vane' separates two chambers, pressurising the fluid in one chamber as the flap moves and the fluid is pushed through a port which can be connected to a hydraulic circuit, as shown in Figure 20(a).

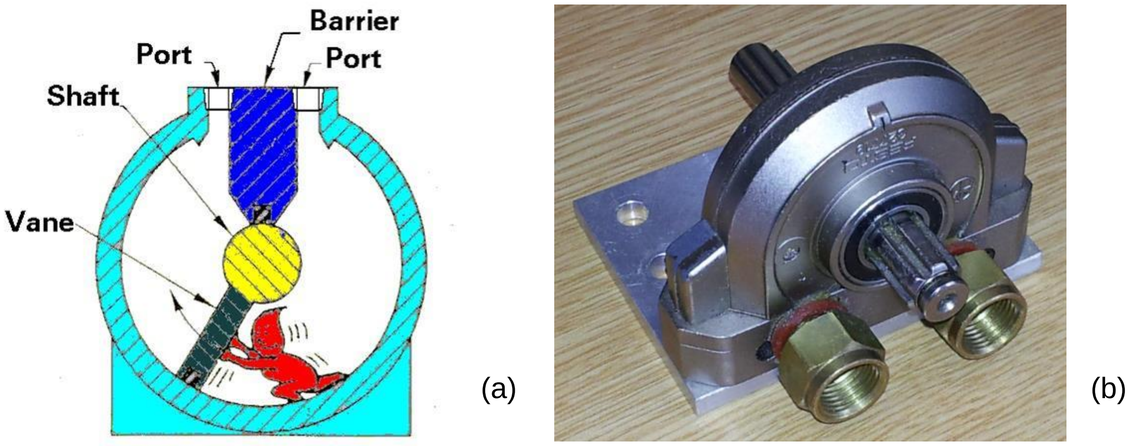


Figure 20. (a) Operation principle of a rotary vane pump (image credit to Micromatic LLC). (b) actual rotary pump used by APL/QUB.

The APL/QUB group tried to develop this system (circa 2009) for use on 40th scale model tests due to the compact geometric size (diameter ~50mm) of the pumps available. However, these small sizes are actually pneumatic pumps and so when used with water, it was found that there was too much flexibility in the vane mechanism as the damping increased. Beyond a torque magnitude of ~1Nm the vane separating the two chambers bends allow water to leak from one chamber to the next. It was found that over damping the system resulted in permanent damage of the pump mechanism.

In addition, it was also found that the accurate adjustment of the damping magnitude (below ~1Nm) was very difficult, again due to the small size of the pump. Only a small volume of water is being pumped by this damper, circa 0.03mL per degree of rotation, compare to ~0.17mL per degree for the linear cylinders discussed in Sections 7and 8. Thus, even the tiniest adjustment in a throttle value can have a significant effect in the damping in the pump. This is also illustrated in Figure 21.

In principle, a rotary hydraulic pump has features which makes it attractive for model testing of a pitching-flap. However, the APL/QUB group could not source sufficiently robust or adequate equipment to provide the level and profile of damping required for small scale model testing. This system was not exhaustively tested by the group or utilised in any WEC performance or design tests. Thus, there is scope to develop this system further to be a more suitable damper for scale model wave tank testing.

9.2 Torque Trace Profile

Figure 21 shows the torque-rotation profile recorded from monochromatic wave tank test. Within its operating range (low torque) the pump delivers a constant (or square-wave) torque profile illustrated by the rectangular profile in this graph. However, as the throttle valve is increasingly tightened (denoted on the graph by increasing the number of valve 'turns') the damping has a component related to the velocity of the flap. Finally, it can be seen that even when the valves are full closed, the flap is permitted to move by circa 10^{o} before the damping rapidly increases to stop this motion. This 'backlash' is due to the distortion of the vane inside the pump allowing water to leak from one chamber to the next. This happens until, ultimately, severe compression of the fluid suddenly induces a damping torque and stops the flap from moving.

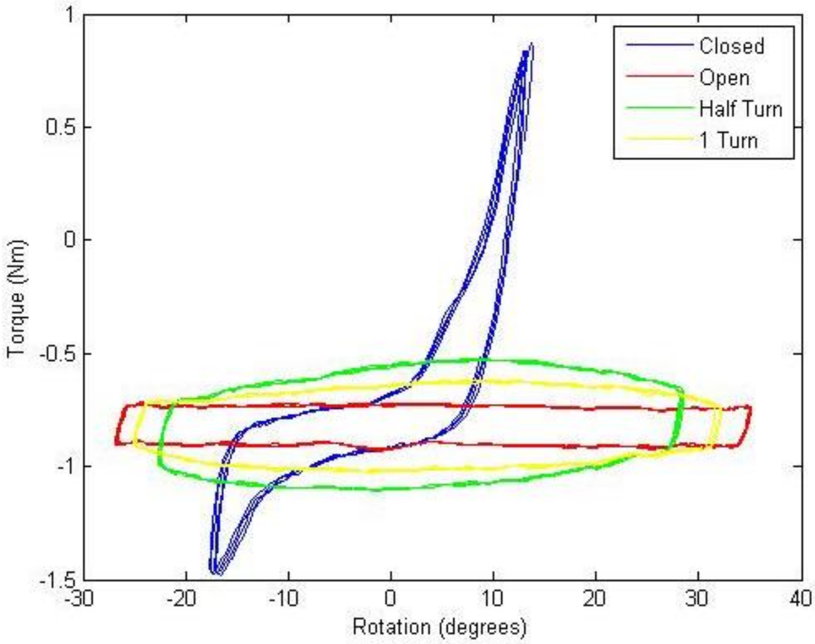


Figure 21. Torque-Rotation plots of the rotary vane pump with different adjustment of a throttle valve. Note, the reference to 'turns' in the legend denote turning the valve from the closed position. (The reader should ignore the bias torque offset on the y-axis as this is due to a test datum analysis error, thus -0.75Nm actually equates to ~0Nm in practise).

9.3 Advantage and Disadvantages

Advantages

i. Can be directly used underwater.
ii. Cheap.
iii. Applies a direct torsional damping to the flap.
iv. Can modify the damping outside of the wave tank.
v. Geometrical small and lightweight and easily integrated into a flap model.

Disadvantages

i. Test repeatability is difficult due to the manual adjustment of the valves.
ii. Very low maximum damping torque of ~ 1Nm.
iii. Adjustment of the damping magnitude is extremely sensitive.
iv. Significant 'back-lash' (~10^{o}) at high/maximum levels of damping.
v. Vane is not robust and can be easily damaged.

9.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Hydraulic
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Approx. Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Manual
Max Damping [Nm]	RMS level achieved experimentally	~ 1Nm
Min Damping [Nm]	RMS level achieved experimentally	~ 0.1 Nm

Assessment Criteria	Score
Repeatability	Poor
Ease of Use/Control	Poor
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	N
Ease of use under water	Good
Lifespan / robustness	Poor
Cost (approximate)	£100
Size (Diameter)	50 mm

Key:

Good
Medium
Poor
N/A

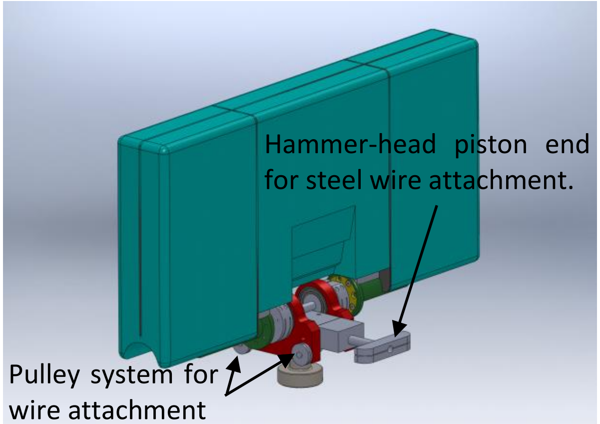
Table 6. Specification and assessment of the rotary vane pump damper system

10 Oil Hydraulic System

10.1 Description & Use

The development of Oyster800 (circa 2009-2011) was based extensively on 40th scale wave tank tests, employing the disc-brake damper system (see Section 6 for details). Although this experimental configuration delivered quality results, the system was found to be somewhat labour-intensive to set up, use and maintain in practise and accurate test repeatability was also difficult to achieve. So, in 2012, APL embarked on developing an entirely new PTO damper system which would address these deficiencies in the disc-brake system enhancing the ability to implement a highly controllable and repeatable damping strategy.

APL engaged the specialist services of the Institute for Fluid Power Drives and Controls (IFAS) based at RWTH Aachen University, Germany. IFAS are experts in hydraulic power systems and they were commissioned to provide a PTO damper for use in small scale experimental wave tank tests, subsequently referred to as the 'IFAS Damper'. Broadly speaking, the damper system was to be compatible with; 1) the Oyster scale flap model(s) (40th scale models in particular); 2) the data acquisition system used at the QUB wave tank facility; 3) key transducers such as load cells and torque tubes; 4) damping ranges experienced in scale model tests; and 5) survive the loads experienced in scale model tests.



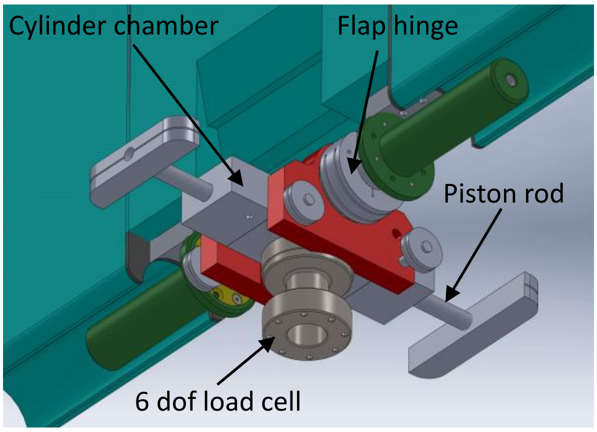


Figure 22. Geometric configuration of the IFAS damper concept. (left) attached to a generic flap-type device. (right) view from underneath the damper attached to a cylindrical/toroidal foundation load cell.

IFAS and APL brainstormed many design iterations but converged on a pressurised, closed-loop, oil-based damper system. The damping is provided by one horizontal, through-rod, double-acting linear hydraulic cylinder, mounded centrally below the hinge line of the flap, but on top of a 6 dof foundation load cell. Figure 22 shows an example of the geometric design of the IFAS damper concept. The pitching motion of the flap is transferred to the linear action of the horizontal piston via a stainless-steel wire-pulley system. A wire is slung around each hammer-head end of the piston rod and is attached to the main rotating flap hinge via the pulley system.

Figure 23 shows a photo of the actual damper hardware delivered by IFAS and an example of how APL integrated the damper into a flap model. The figure also shows the damper integrated into a 40th scale model of the Oyster800 device.

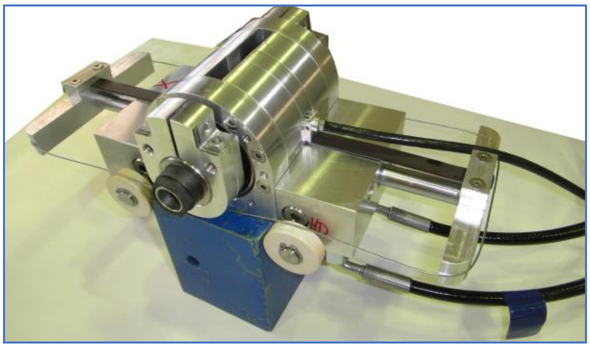
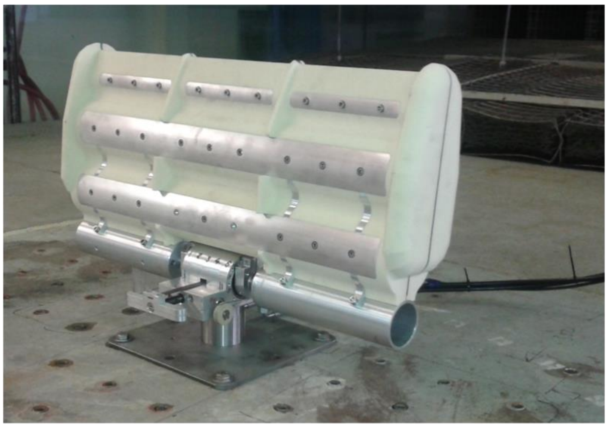



Figure 23. (left) The IFAS damper. (right) The damper integrated into a 40th scale Oyster800 model.

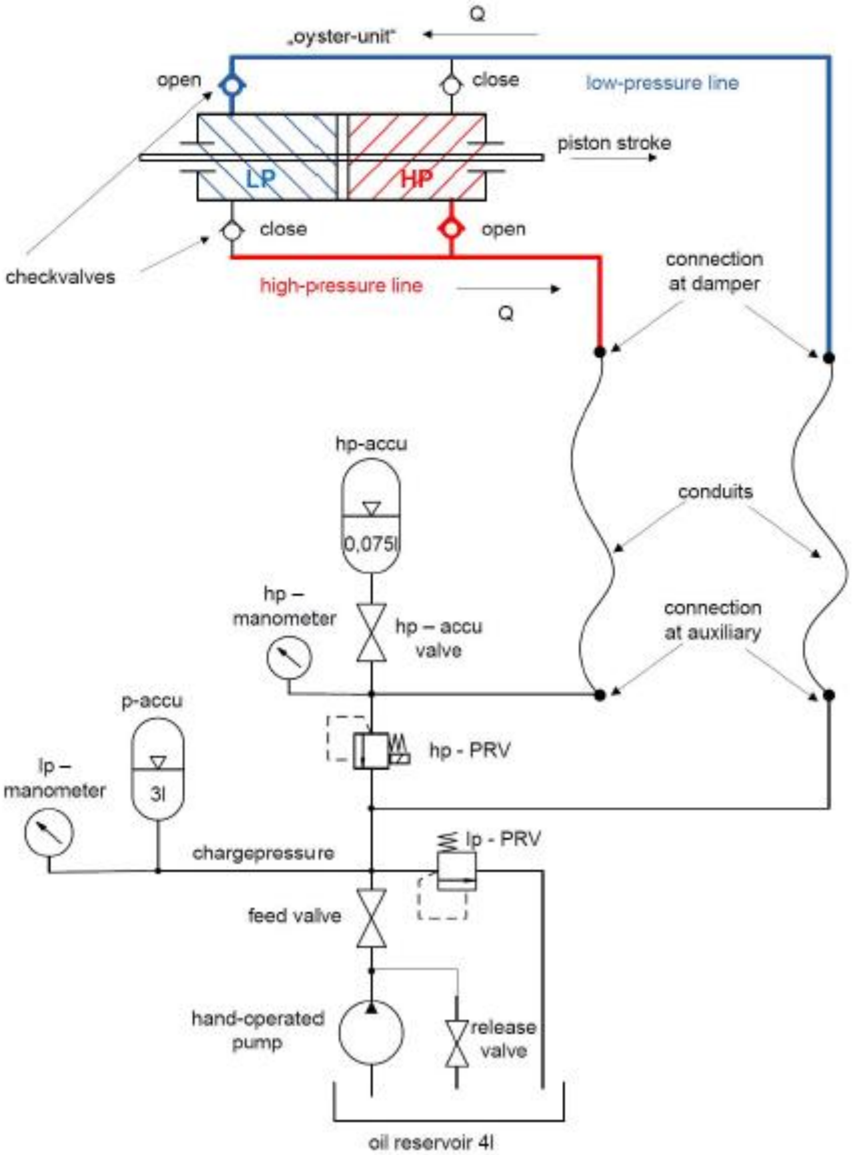


Figure 24. Schematic diagram of the hydraulic circuit of the IFAS damper.

The auxiliary hydraulic circuit of the IFAS damper system is shown in Figure 24. The damping magnitude applied by the system is governed by the internal hydraulic pressure in each chamber either side of the piston head. A series of flow-rectifying check valves ensures that flow in the auxiliary circuit is unidirectional. The fluid pressure in the chambers is controlled using an electrically controlled pressure relief valve (PRV) attached between the high- and low- pressure sides of the hydraulic circuit. The operation of this valve can be controlled by a signal sent from a PC (or independent power supply) and linked to the data acquisition system, thus allowing dynamic control of the damping applied during tests. The other components in the circuit are used primarily for safety, monitoring and commissioning purposes. For brevity, they are not discussed in detail here.

Figure 25 shows the auxiliary hardware of the IFAS damper system which is installed on a single support frame. This equipment is mounted outside of the wave tank for ease of access and connection to the data acquisition/control system. Although this system was built for a single PTO damper unit, IFAS suggests that the auxiliary circuit could be used with multiple damper units (i.e. simultaneously operating flaps (e.g. an array)). Up to 5 units was suggested as the max capacity of the system. Thus, there is some economies of scale within the hardware delivered with this design.

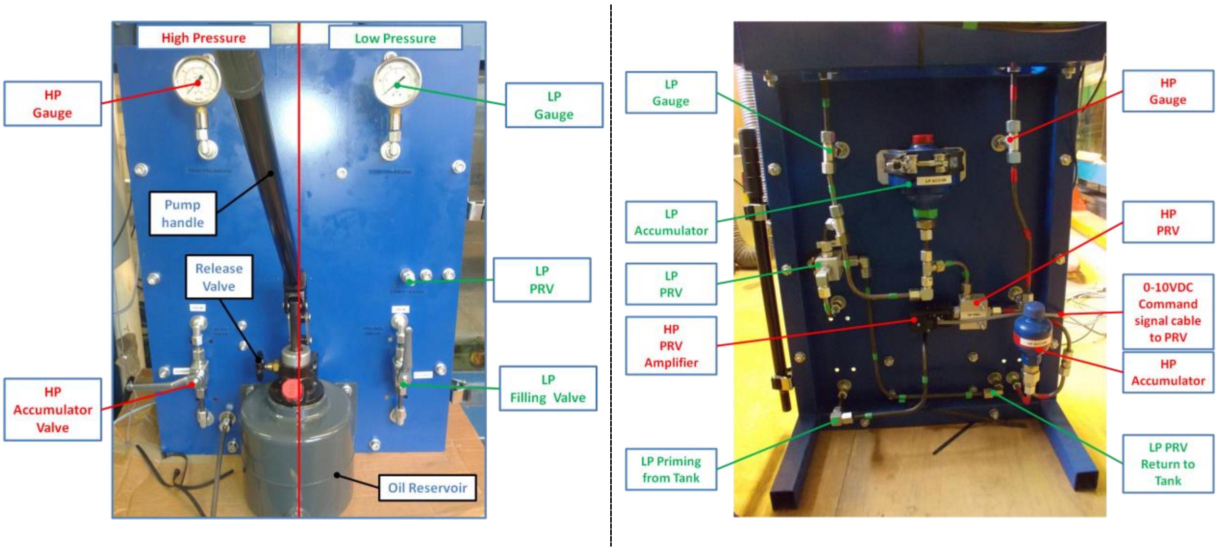


Figure 25. Front (left) and back (right) images of the auxiliary hydraulic circuit hardware. The components of the high- and low- pressure side of the circuit are labelled in red and green respectively.

IFAS conducted a series of 'dry' (not submersed in water) laboratory tests on the system before delivery to APL and showed that the damper could deliver the principles of operation required (see Section 10.2 for more details). In addition to this, APL also conducted a series of commissioning tests which integrated the damper system into a functioning flap model and also used it in preliminary wave tank tests. The focus of APLs efforts was on integrating this damper into 40th scale flap models. Despite the damper system delivering on almost all of its functional requirements, a key issue arose during these trials, which was an excessively high level of minimum damping (for APL's 40th scale model tests requirements). The system as delivered could only achieve a minimum damping of >2Nm, which equates to >5MNm at full scale (over 50% of the capacity of the real Oyster800 PTO system!).

APL and IFAS worked together to improve the model in an attempt to reduce the minimum level of damping. The changes made were: to reduce the threshold pressure on the low-pressure relief valve; use a lower lubrication oil in the system; re-plate the cylinder rod with a higher grade, lower friction material; and replace the rotational bearings with glass ones. However, these modifications combined only reduced the minimum damping to circa 1.7Nm, which was still too high for APLs requirements for 40th scale model tests. That is not to say that the system is not in general fit-for-purpose for flap model testing, but rather it was just out of range for APLs 40th scale model test programme. However, if a slightly larger scale was selected for model testing then this system could deliver a very controllable and repeatable PTO damping strategy. For example, if this was used at 20th scale, then the full-scale minimum torque would be <0.3MNm.

This development work was undertaken in 2013. However, during this time APL's R&D priorities shifted to the analysis and monitoring of the data recorded from the full-scale Oyster800 device, which was in operation at EMEC, Orkney, Scotland. Analysis of the Oyster800 device continued to be a top R&D priority of the company until it ceased trading in 2015 and so, testing and commissioning of the IFAS damper was never fully completed or used by APL to conduct any WEC performance or development wave tank tests.

10.2 Torque Trace Profile

As mentioned previously, IFAS conducted a series of 'dry' laboratory tests on the damper system during its design. Figure 26 shows an example of the damping torque delivered by the system under two different configurations, with and without the high-pressure accumulator in the circuit. The hydraulic piston was artificially driven with a real motion profile recorded from one of APL's historic wave tank tests. Three different target torque magnitudes were demanded from the system (green line) and it can be seen that the IFAS damper delivered on these demand signals with a very well conditions square wave torque profile. Note however that due to the presence of the high-pressure accumulator, there is a lag-time in the system before it delivers on the target signal. The accumulator is only installed for safety and to absorb very high-pressure spikes in the system and so is not necessarily required during normal operation (i.e. typical performance sea state tests). Thus, the system can be tailored to produce the necessarily responsiveness of a given test programme.

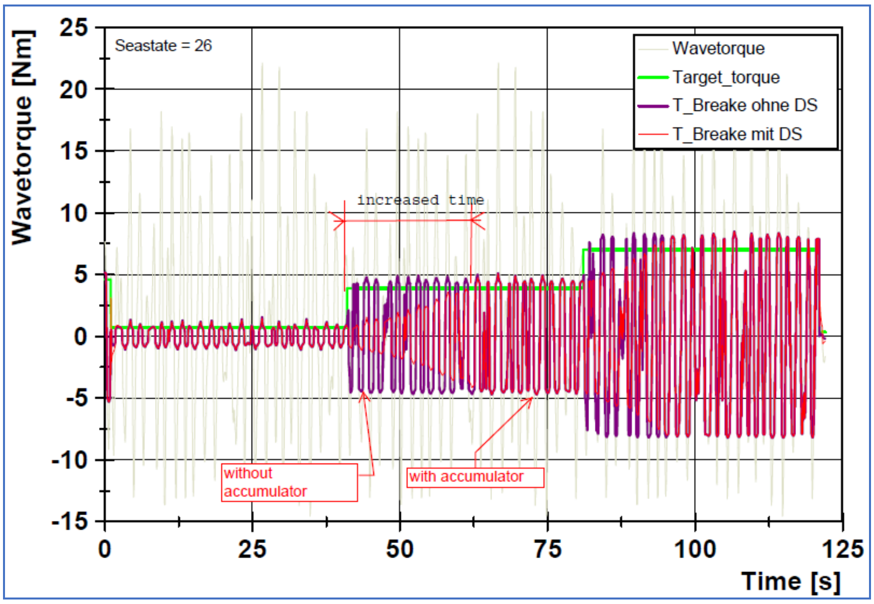


Figure 26. Damping profile delivered by the IFAS damper system during 'dry' forced laboratory tests. The green line denotes the target torque signal demanded from the control system. The purple traces show the system response if no accumulator is present and the red line shows the response with an accumulator in the high-pressure side of the hydraulic circuit.

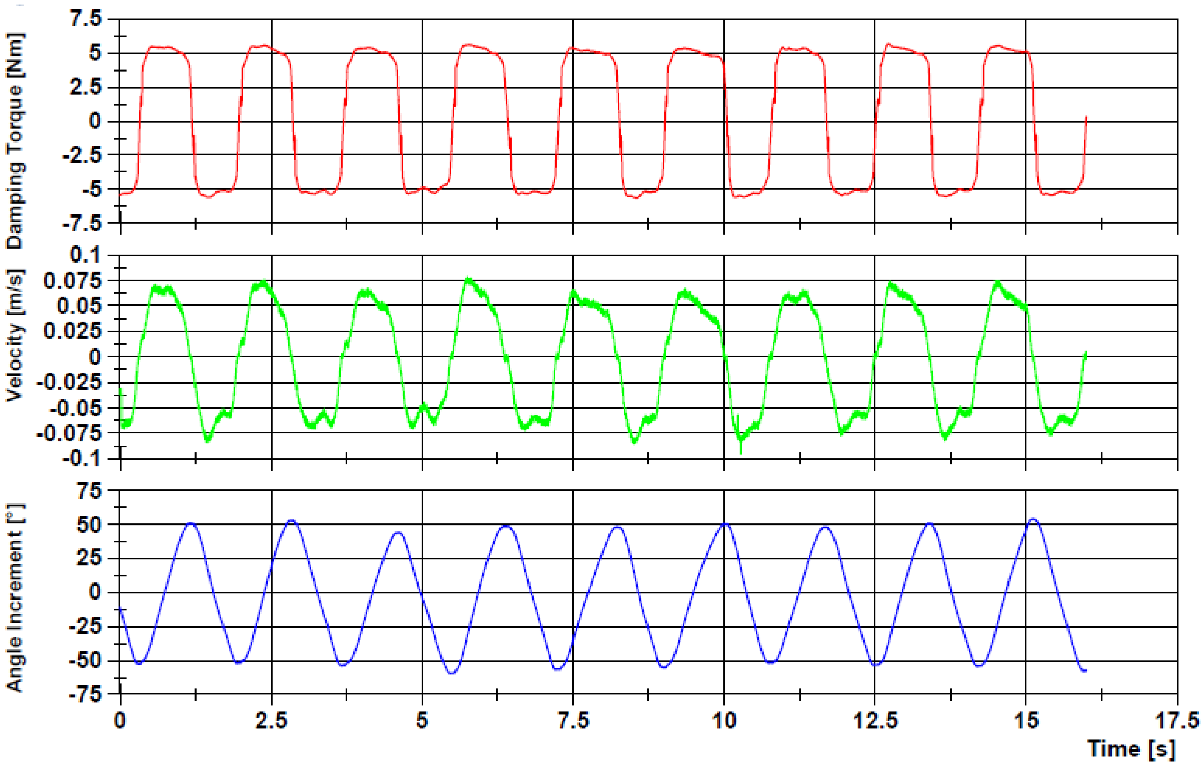


Figure 27. Damping torque profile (top), linear velocity of the horizontal piston rod (middle) and angular rotation of the flap (bottom) under forced sinusoidal motion during the 'dry' laboratory tests at IFAS.

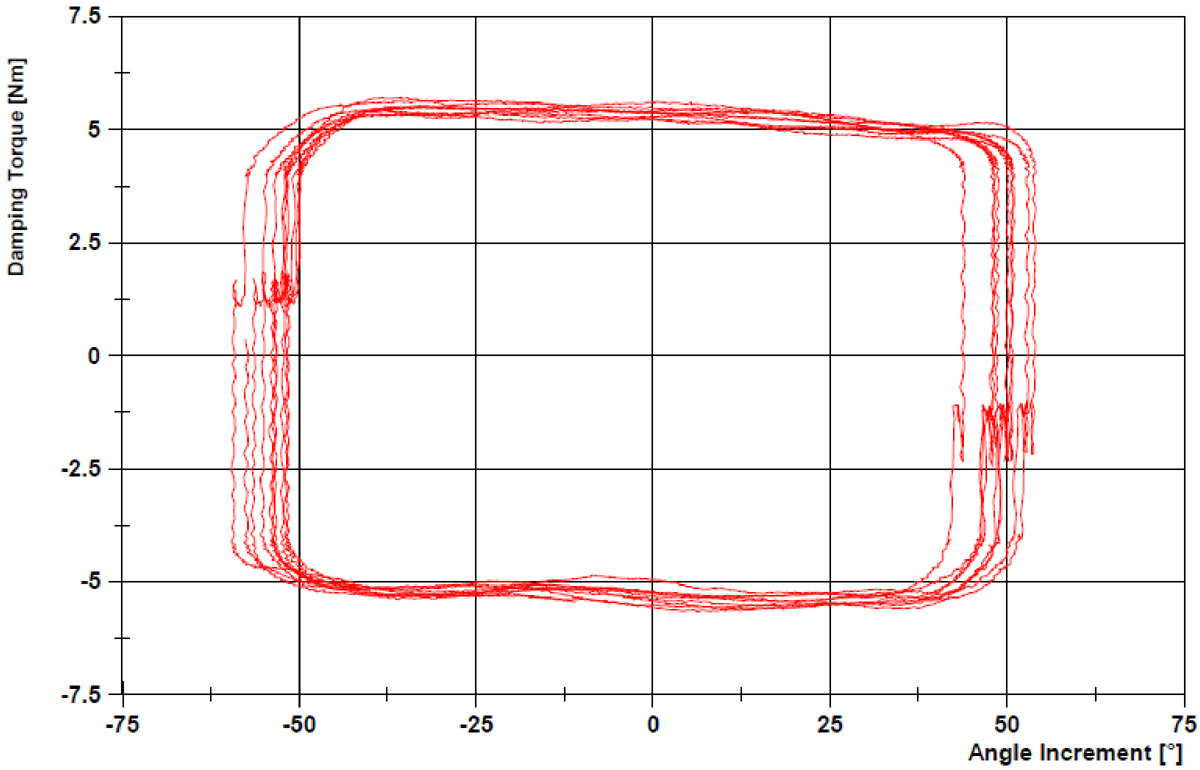


Figure 28. Damping torque versus flap rotation angle showing a very well-conditioned square-wave damping profile during 'dry' laboratory tests at IFAS.

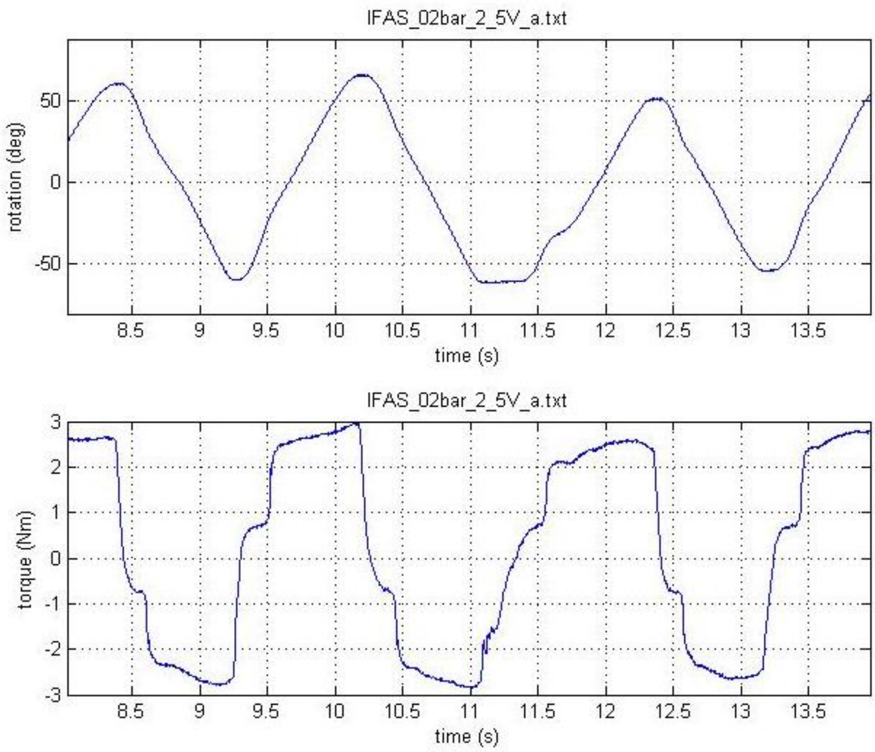


Figure 29. Flap rotation angle (top) and damping torque profile (bottom) recorded during 'wet' wave tank tests conducted by APL at the QUB wave tank facility.

Figure 27 and Figure 28 show the well-conditioned sqaure-wave damping profile delivered by the IFAS damper during 'dry' laboratory tests conducted by IFAS, prior to delivery to APL. It can be seen that this design can deliver an excellent and controllable damping within a given torque magnitude range of circa 2Nm up to 10Nm.

Figure 29 shows the damping profile achieved during APL's commissioning wave tank tests in monochromatic waves. Again, it can be seen that the damper delivers a good approximation to a square-wave torque profile when the RMS magnitude is above ~2Nm. It is interesting to note in this profile though that there is more evidence of 'backlash' in the system when the flap motion changes direction. The fact that this was not present during the 'dry' laboratory tests conducted by IFAS would suggest this 'loosness' may come from the interface between the damper and the flap or some other source of compliance in the flap system, outside the damper hardware itself.

10.3 Advantage and Disadvantages

Advantages

i. Can be used underwater.
ii. Damping can be digitally controlled outside of the wave tank.
iii. Once the initial equipment set-up overhead is overcome, the damping torque is easily modified and very repeatable.
iv. Damping can be controlled dynamically, albeit more on a wave group rather than wave-by-wave basis during tests.
v. Delivers a very well-conditioned square wave damping strategy.
vi. Single hydraulic system could be used to control multiple flaps/dampers.

Disadvantages

i. The minimum damping achievable is quite high at 1.7Nm. This is only seen as a disadvantage for flap model tests ≤40th scale. The scale of the tests could actually be selected around the torque capacity of the damper.
ii. Quite a complex set up and geometrical interface with the flap model.
iii. The auxiliary equipment is heavy and bulky and not easy to manoeuvre.
iv. The initial test set-up and priming of the hydraulic circuit is labour intensive/time consuming.
v. Very expensive.

10.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Hydraulic
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Linear
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Automatic
Max Damping [Nm]	RMS level achieved experimentally	10 Nm
Min Damping [Nm]	RMS level achieved experimentally	1.7 Nm
Assessment Criteria	**Score**	
Repeatability	Good	
Ease of Use/Control	Good	
Control outside tank (Y/N)	Y	
Dynamic Control (Y/N)	Y	
Ease of use under water	Medium	
Lifespan / robustness	Good	
Cost (approximate)	£19k for 1st full system ~£5k for additional units	
Size	126 mm wide ~ 270mm piston length	

Key:

Good
Medium
Poor
N/A

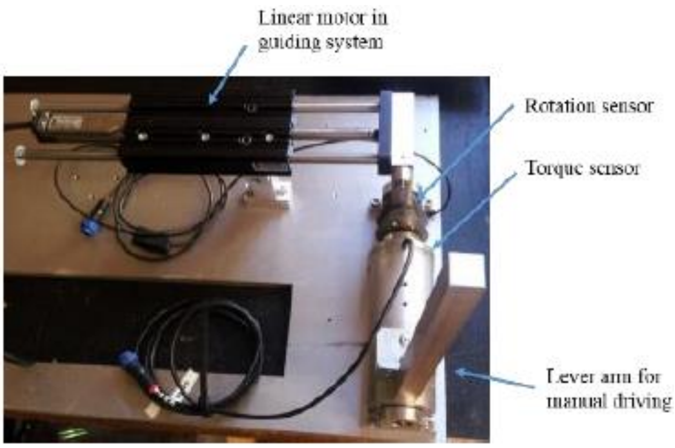
Table 7. Specification and assessment of the oil-based, closed-loop, IFAS hydraulic damper system

11 Linear Motor

11.1 Description & Use

In 2013, an Engineering Doctorate (EngD) project was initiated as part of the IDCORE programme. APL, in partnership with QUB, were the industrial supervisors for the project and the EngD student was based at the wave tank facility in QUB. The project investigated the design of a modular flap-type WEC using physical testing and numerical research techniques, see [R2] for details. As part of the physical scale model design, a detailed assessment of PTO dampers was conducted and a Linear Motor was one such version assessed. This is the only source of information and testing of this type of damper done by the extended APL-QUB research group and the damper system never progressed beyond 'dry' bench testing done in-house at the QUB facility.

The linear motor used was manufactured by LinMot and attention was first drawn to this type of system from experience in the testing of pitching buoys at Aalborg University, [R5]. The motor is made up of a stationary part – the 'stator' and a moving part – the 'slider'. The stator is an embedded coil, to which current is supplied to control the motor. The slider contains permanent magnets. The magnetic force between the coil and magnets provided the driving force. A motor was temporarily provided by LinMot to enable preliminary tests to be done on the system to assess its suitability for bottom hinged, flap-type model tests. The equipment was tested in a dry bench rig (Figure 30), designed and manufactured in-house. The rig allowed the motor to be tested by attaching it to a lever arm, connected to a shaft. At the shaft were torque and rotation sensors.



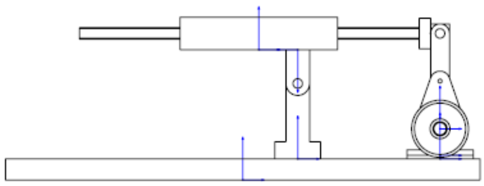


Figure 30. Bench test-rig used to assess the linear motor damper system.

The testing showed that the damping levels were repeatable, with recorded deviations of less than 0.1 Nm for mean torques. The recorded damping torque range was 0.2-5.2 Nm, showing good suitability for APL's 40th scale flap model tests. Although the manufacturers of this motor suggest its use with water with various waterproofing standards indicated on its components, its sustained use immersed under water was not recommended and the EngD project did not reveal any historic literature/evidence where this was achieved by another research group. In addition, the temporary motor provided by LinMot was not of an adequate waterproof grading to attempt to test its immersion capability out. Thus, it was concluded that if such a damper was to be used, it would have to remain outside of the water.

11.2 Torque Trace Profile

As mentioned previously, a plot of damping torque against flap rotation would yield a square shape if a 'constant' (or square-wave) damping strategy is provided. With the linear motor damper in its basic form, it was found that the damping profile was not a square shape. In addition, small oscillations of the damping also occurred in the signal, likely due to the magnet passing the non-continuous stator coil. A graph illustrating these two observations is shown in Figure 31.

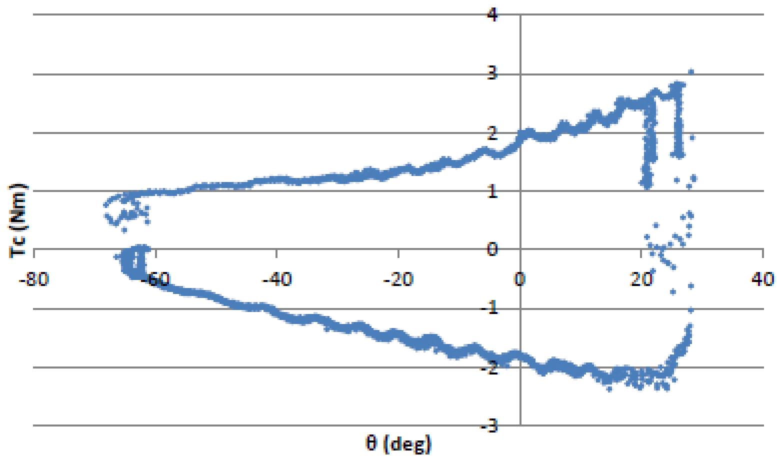


Figure 31. Plot of flap rotation angle versus the damping torque provided, shown a non-constant damping profile. Oscillations due to the behaviour of the magnet can also be seen in the signal. The rotation 'limit' of circa +25° is due to a shortfall in the test-rig configuration design.

For a pitching/rotating WEC system, and unless otherwise controlled, the damping from a linear force/motion applied at a lever arm will inevitably result in a non-constant (non- square-wave) torque profile due to the fact that the effective lever arm will also change to accommodate the geometry between the damper and prime mover connection point. However, it is predicted that the linear motor damper could provide a constant damping strategy, but it would have required an upgrade to the system, using a force control module. This would incorporate a closed-loop measurement and control system for the prime mover velocity and damping force, requiring extra hardware and software. Alternatively, cable-wheel drive system, similar to that employed in the hydraulic-oil IFAS damper (see Section 10), could have been used in conjunction with the motor.

From the preliminary test results and the estimate of the cost and complexity of upgrading the system to provide a more constant-type damping strategy, it was concluded that this system would not be suitable for the testing needs of APL. Thus, the system was not developed any further. However, it should be reiterated that other research groups (e.g. Aalborg) have successfully used this system for wave tank testing of WECs. This highlights again that conclusions iterated in this report are only directly applicable to APLs technology and testing strategy. The reader must take care when interpreting the results and conclusions reported.

11.3 Advantage and Disadvantages

Advantages

i. Damping torque range is very suitable for circa 40th scale flap model tests.
ii. Controllable and repeatable damping profile.
iii. Damping can be digitally controlled outside of the wave tank.
iv. Damping can be controlled dynamically.

Disadvantages

i. Cannot be directly used under water.
ii. Expensive.
iii. A complex set up and geometrical interface with the flap model.
iv. Does not directly provide a constant (square-wave) damping strategy.
v. Initial equipment set up can be time consuming.
vi. Large piece of equipment, results in complex flap-damping geometric design and interface.
vii. Very difficult to integrate on top of a 6 dof foundation load cell arrangement.

11.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Electric
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Linear
Damping Profile	Constant/Linear/Quadratic/Other damping	Other
Type of Control	How the damping level is controlled – Manual/Automatic	Automatic
Max Damping [Nm]	RMS level achieved experimentally	5.2 Nm
Min Damping [Nm]	RMS level achieved experimentally	0.2 Nm

Assessment Criteria	Score
Repeatability	Good
Ease of Use/Control	Medium
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	Y
Ease of use under water	Poor
Lifespan / robustness	Good
Cost (approximate)	~£3,000 per unit (motor & control system)
Size †	300 mm slider length 120mm stroke

Key:

Good
Medium
Poor
N/A

Table 8. Specification and assessment of the LinMot linear motor damper system

† This was the size tested by APL/QUB. The manufacturers can provide a range of liner motor sizes and stroke lengths

12 Electric Hysteresis Brake

12.1 Description & Use

As introduced in the previous section (Section 11), in 2013 an extensive assessment of physical PTO damper systems was conducted as part of an IDCORE engineering doctorate project, coordinated by APL and QUB. During this assessment, a rotary Electric Hysteresis Brake (EHB) was tested and evaluated. The brake examined was manufactured by Mobac GmbH (model HB-250M-2), with a rated torque of 2.3 Nm, diameter of 112.5 mm and length of 91.7 mm. At the time of the assessment, the cost of such a unit/model was circa £650. A photograph of the brake is provided in Figure 32.

Figure 32. Mobac's electric hysteresis brake

The brake produces a damping torque by exerting magnetic drag on an internal rotor connected to the shaft. The magnetic field is generated with a coil and so, the strength of the field (and thus the torque level) is proportional to the electric current supplied to the brake. This mechanism results in the torque being independent of rotational velocity. Thus, it gives the 'constant' (square-wave) damping profile desired by APL.

The brake itself is not inherently waterproof and so the EHB was only 'dry' bench tested using a similar experimental rig to that employed for the linear motor (see

Figure 30). It was speculated by various brake manufacturers that an electrical hysteresis brake may encounter an issue if it operated in a cyclic oscillatory fashion. The perceived problem was that the brake would show an overshooting of torque when the shaft changed direction. However, the initial bench test results were very promising (as discussed in the Section 12.2) and no evidence of this issue was observed.

The brake was evaluated in comparison with a similar damper system, the Magnetic Particle Brake (MPB) (discussed in Section 13), which was ultimately ranked just ahead of the EHB. Thus, the EHB system was not pursued beyond 'dry' bench testing but it is still believed to be a very robust and suitable rotary damper system for certain scale model WEC testing programmes. It should be noted that waterproofing of the system requires careful attention. although as part of the IDCORE project an innovative and very effective waterproofing solution was developed using rotary magnetic couplings. Details of this are presented in Section 15.

12.2 Torque Trace Profile

The dry bench tests showed that the EHB delivers a very well-conditioned constant or square-wave damping torque profile as can be seen in the results displayed in Figure 33 and Figure 34.

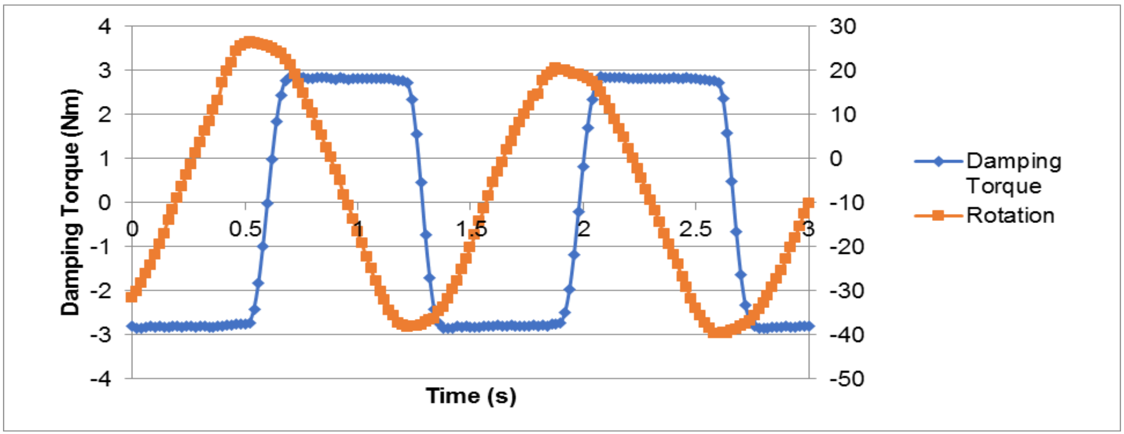


Figure 33. Square-wave damping profile delivered by the EHB.

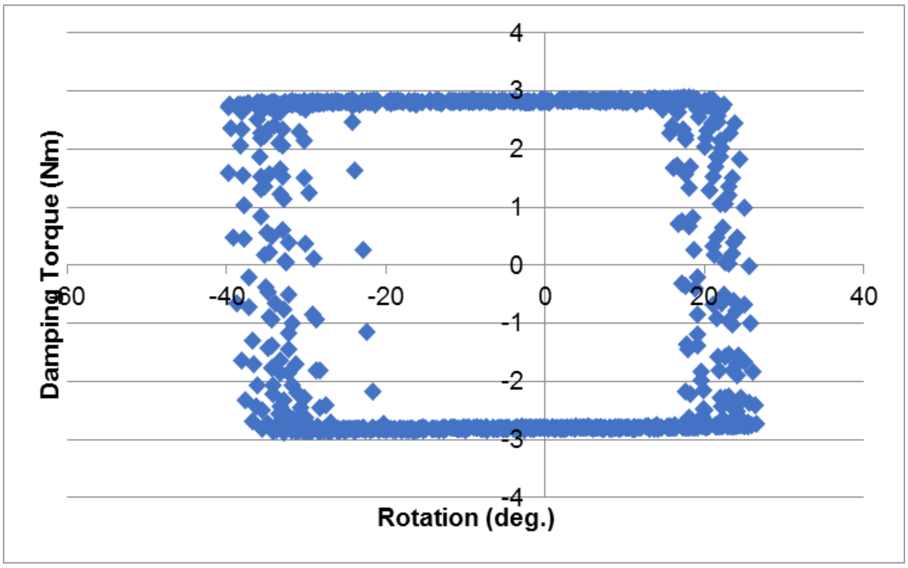


Figure 34. Torque vs Rotation plot illustrating the square-wave damping profile delivered by the EHB.

In addition, the EHB also displayed excellent repeatability of the damping torque levels and a high level of controllability. As the name might suggest, an interesting feature of this brake system was a hysteresis characteristic between the supplied current to the magnetic coil and the damping torque experienced on the rotating shaft. Figure 35 shows the damping recorded during the bench tests as the current was increased from 0 up to its maximum and then subsequently decreased from its maximum back down to 0. Such a feature may have some minor consequences in terms of selecting a specific damping torque magnitude if this system was used in practise in a real wave tank test. However, this hysteresis loop was found to be very repeatable and so, characterisation of the damper in advance of testing would overcome any potential issue associated with this.

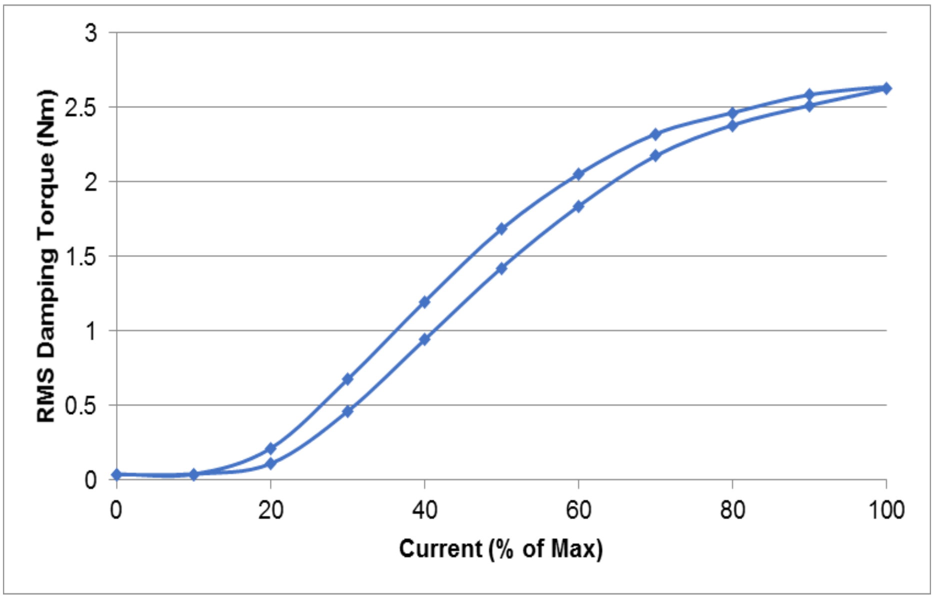


Figure 35. Torque vs Supply Current hysteresis loop of the EHB.

12.3 Advantage and Disadvantages

<u>Advantages</u>

i. Excellent square-wave damping torque profile.
ii. Damping torque range is very suitable for circa 40th scale flap model tests.
iii. Very controllable and repeatable damping profile.
iv. Damping can be digitally controlled outside of the wave tank.
v. Damping can be controlled dynamically.
vi. Direct rotary damping mechanism more compatible with a pitching flap design.

<u>Disadvantages</u>

i. Cannot be directly used under water. Thus, a waterproofing solution must be design in addition
ii. Quite expensive at ~£650 for the brake.
iii. EHB is quite large (see Table 9) and heavy for the torque range it delivers. (i.e. smaller torque ranges are appropriate for geometrically small-scale testing, but as this equipment is physically quite large it makes the model integration more complex).

12.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Electric
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Automatic
Max Damping [Nm]	RMS level achieved experimentally	2.6 Nm
Min Damping [Nm]	RMS level achieved experimentally	0.04 Nm

Assessment Criteria	Score
Repeatability	Good
Ease of Use/Control	Good
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	Y
Ease of use under water	Poor
Lifespan / robustness	Good
Cost (approximate)	£650 (+water proofing)
Size	113 mm diameter 92 mm length

Key:

Good
Medium
Poor
N/A

Table 9. Specification and assessment of the Electric Hysteresis Brake damper system.

13 Magnetic Particle Brake

13.1 Description & Use

A similar damping system was testing in parallel to the Electric Hysteresis Brake (EHB) (discussed previously in Section 12) under the IDCORE project. This was a Magnetic Particle Brake (MPB).

A MPB consists of a hysteresis disk/rotor within a magnetic-particle filled powder cavity. The brake operates by using the magnetically-energised particles to apply friction to the rotor which is connected to an output shaft, ultimately attached to the WEC's oscillating prime mover. The magnetic field is created with an electrical coil inside a stator. Thus, the friction level and so damping torque, is controlled by changing the supply current to the coil. With no current applied, the disk is free to move with only bearing friction acting to oppose relative motion. As a current is applied, low current coils generate an electromagnetic force, causing the particles to bind into a sort of 'particulate fluid'. As the current is increased, the binding force increases and the apparent viscosity of the particle fluid increases, resulting in a drag based damping form being imposed on the motion of the hysteresis disk. Similar to the EHB, this mechanism will result in the damping torque being independent of rotational speed, thus delivering a constant or square-wave damping profile. A cross-section showing the working principle of a MPB is shown in Figure 36 (left). Particle brakes are often used in areas where swift response of varied damping is required. They are also known for their ability to provide a wide range of damping levels with high levels of both accuracy and precision.

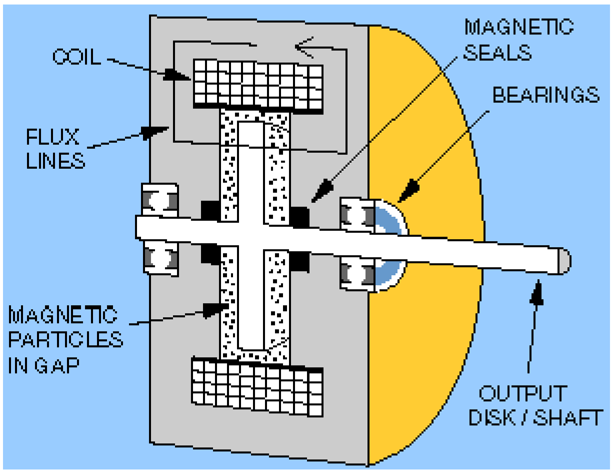


Figure 36. Cross-section of a MPB (left). Example of the range MPB supplied by Placid Industries Ltd. (right)

The unit tested by APL/QUB (via the IDCORE project) was supplied by Placid Industries Ltd. who supply a range of different MPB ratings. The B35 model was purchased which has a torque rating of 0.06 – 3.95 Nm, external diameter of 86mm and at the time of testing cost circa £350.

Similar to the EHB, the MPB is inherently not waterproof. Thus, careful attention must be given to an appropriate water proofing solution. The IDCORE project developed an innovative waterproofing solution using rotary magnetic couplings for this damper, details of which are discussed in Section 15.

The MPB was first dry bench tested similar to the EHB (Section 12) and the Linear Motor (Section 11). The preliminary bench-test results were very promising and so a waterproofing solution was developed and the damper then tested in the QUB wave tank. Ultimately, this damper system was fully deployed and used very successfully on a 30th scale modular flap WEC concept which was the focus of the IDCORE project. Details of this have been published in several sources, the most relevant of which are [R2] and [R6].

13.2 Torque Trace Profile

As this system essentially operates on a magnetic hysteresis based principle, the damper experiences a similar torque-current hysteresis characteristic to the EHB, as shown previous in Figure 35. However, this was found not to affect the usability or performance of the damper.

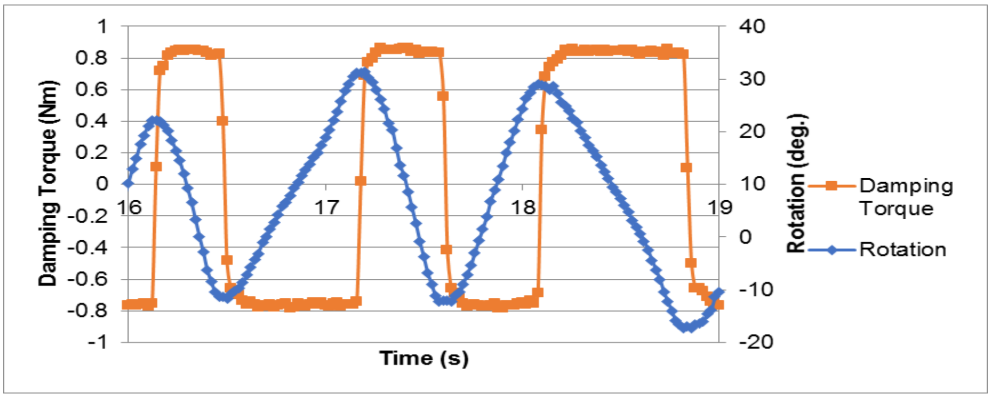


Figure 37. Damping strategy provided by the MPB system.

The dry bench tests showed that the MPB delivers a very well-conditioned constant or square-wave damping torque profile as can be seen in the results displayed in Figure 37. Following extensive dry bench tests and the development of a waterproofing solution (see Section 15); the damper system was integrated into a flap model and tested in the wave tank at QUB.

Figure 38 shows the results from some of the wave tank tests where the flap-damper system was tested in regular (monochromatic) wave (top graph) and irregular (polychromatic) wave conditions (bottom graph). It can be seen in both cases that the MPB delivered a very well-conditioned, repeatable square-wave damping profile, independent of flap velocity.

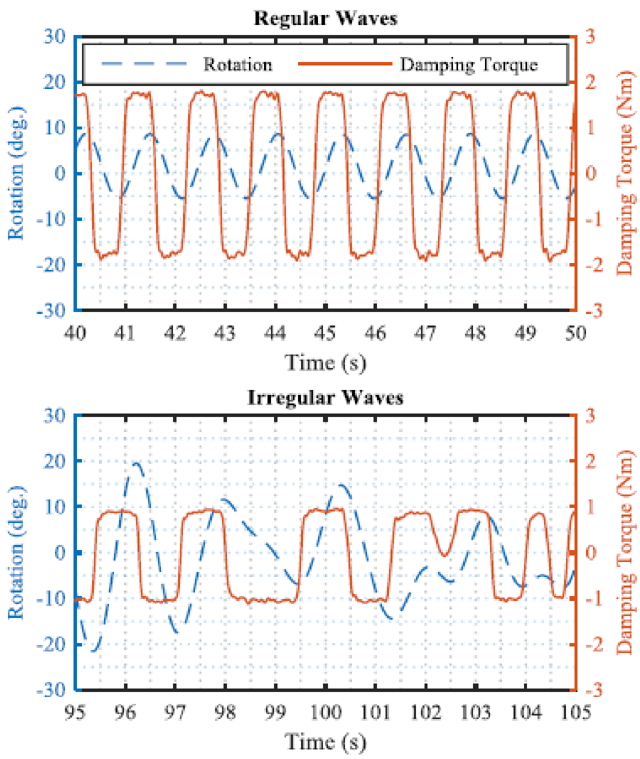


Figure 38. MPB damping during wave tank tests.

Based on discussions with the manufacturer, the applications of magnetic particle brakes are typically not used for oscillatory motions. It was warned that changing the direction of the shaft for over 50 cycles will result in eventually a sudden increase in torque due to uneven distribution of the magnetic particles. However, during extensive bench and wave tank testing of the brake, horizontally mounted with oscillatory motions several orders of magnitude greater than the quoted 50-cycles, showed no signs of degrading the damping profile. This test campaign showed that this system is capable of delivering the required damping characteristics, which is easily controllable and very repeatable.

13.3 Advantage and Disadvantages

<u>Advantages</u>

i. Excellent square-wave damping torque profile.
ii. Very controllable and repeatable damping profile.
iii. Damping can be digitally controlled outside of the wave tank.
iv. Damping can be controlled dynamically.
v. MPB's are available in a wide range of torque ranges.
vi. The torque range to geometric size ratio is very suitable for scale WEC model testing.
vii. Direct rotary damping mechanism more compatible with a pitching flap design.
viii. Good value for money at ~£350 per unit.

<u>Disadvantages</u>

i. Cannot be directly used under water. Thus, a waterproofing solution must be design in addition.
ii. The brake itself is quite heavy at ~1.25kg.

13.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Electric
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Automatic
Max Damping [Nm]	RMS level achieved experimentally	3.95 Nm
Min Damping [Nm]	RMS level achieved experimentally	0.07 Nm
Assessment Criteria	**Score**	
Repeatability	Good	
Ease of Use/Control	Good	
Control outside tank (Y/N)	Y	
Dynamic Control (Y/N)	Y	
Ease of use under water	Poor	
Lifespan / robustness	Good	
Cost (approximate)	£350 (+water proofing)	
Size	86 mm diameter	

Key:

Good
Medium
Poor
N/A

Table 10. Specification and assessment of the Magnetic Particle Brake damper system.

14 Force Feedback Dynamometer

14.1 Description & Use

APL continually aimed to develop, evolve and improve their research techniques. In 2012, they ambitiously set out to develop a Force Feedback Dynamometer (FFD) which could fully control a WEC model in real-time during small scale experimental wave tank testing. Unlike, a passive damper system (like those discussed in other sections of this report), which are only reactive to the WEC prime mover, a dynamometer can also be used to actively drive the motion of the WEC. This system not only unlocks a huge potential for the physical testing of advanced PTO control system research, but it also enables fundamental wave-structure hydrodynamic research to be pursued and/or verified by physical means. This dual purpose was the motivation behind APL's FFD system.

A QUB Ph.D. project (the student of which was also heavily involved in the development and commissioning of the system) utilised the FFD to conduct Forced Oscillation Tests on a flap-type WEC, and determine and verify the fundamental hydrodynamic parameters governing the wave-structure interaction. Details of this pioneering research can be found in several published sources, the most relevant are [R8] and [R9].

APL engaged the services of power and automation specialists, MoTeam Ltd. to help develop the system. The FFD system developed uses a 3-phase permanent magnet motor to indirectly drive the flap. The motor, on a stand above the water, rotates a primary drive shaft. A toothed belt is then used to connect this primary shaft to a secondary drive shaft which is submerged in the wave tank, on which the flap WEC model attaches. A photo and schematic diagram of the FFD system developed in shown in Figure 39. It should be noted however, that during development and commissioning various iterations of this design evolved. For example, the FFD was successfully used with a flap model which had a direct torque tube sensor integrated into the hinge line of the flap, negating the necessity to have a torque sensor on the primary shaft, as shown in Figure 39(b).

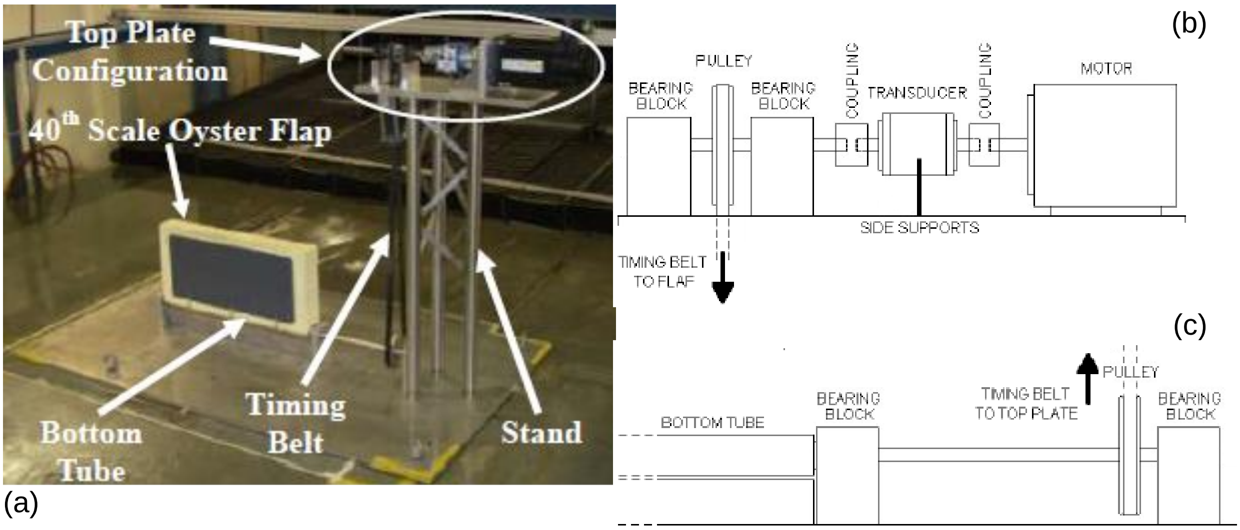


Figure 39. (a) photo of a version of the FFD system developed by APL; (b) schematic of the top plate configuration (primary shaft); (c) schematic of the secondary shaft onto which the flap model connects.

The motor is controlled with a programmable logic controller (PLC) and a PC, with a user-interface. The PLC system acts as an interpreter between a programmable language and a driven mechanical system. The user writes the control program within an interface software (e.g. APL's FFD system uses Bosch Rexroth's IndraWorks). This control is passed via an Ethernet to a SERVO, which regulates the motion of the motor via an integrated motor encoder. The encoder allows position and velocity control, which is important for PTO and control system experiments as it allows damping strategies to be triggered off a velocity threshold; this facilitates investigations into complex damping strategies. Further to the plethora of PTO strategies available for investigation with PLC control, the system enables analogue and digital input/output (I/O). I/O boards are coupled together with a bus coupler which groups signals from the I/O channels and passes them to the SERVO. The SERVO communicates these signals back to the user interface and if desired passes them to a data acquisition (DAQ) system. A schematic of the control architecture is shown in Figure 40. A detailed description of the development of this system was published by [R7].

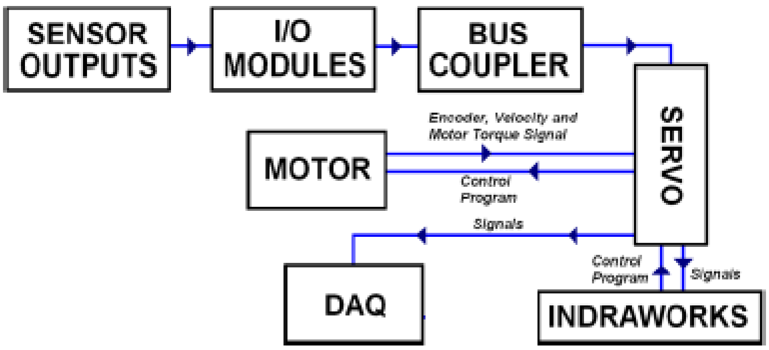


Figure 40. Control architecture of the FFD system.

The FFD system, which was installed at the wave tank facility at QUB, underwent significant in-house commissioning by APL staff. Many mechanical and PLC system refinements were made in order to enhance and optimise the functionality of the system. Much of this work was conducted in 2013. After this however, APL's research activities focused on analysing the behaviour of the full scale Oyster800 device which was installed and operating at EMEC, Orkney, Scotland. This became the priority for the company and continued until the company ceased trading in 2015. Thus, the FFD system was never fully used by APL to conduct detailed WEC PTO damping and control system research.

14.2 Torque Trace Profile

As with such a sophisticated and intelligent piece of equipment it is no surprise that the FFD could deliver the required square-wave damping torque profile required by APL. In fact, this system could deliver a wide variety of damping strategies which could be controlled and reactive on a wave-by-wave basis. Figure 41 shows an example of two simple damping strategies which are easily achieved with the FFD system. The graph on the left shows a regular square-wave damping strategy implemented in monochromatic wave tank tests. The graph on the right is a directionally biased square wave damping in which the flap motion is only resisted when moving in one direction (in this case moving towards the land).

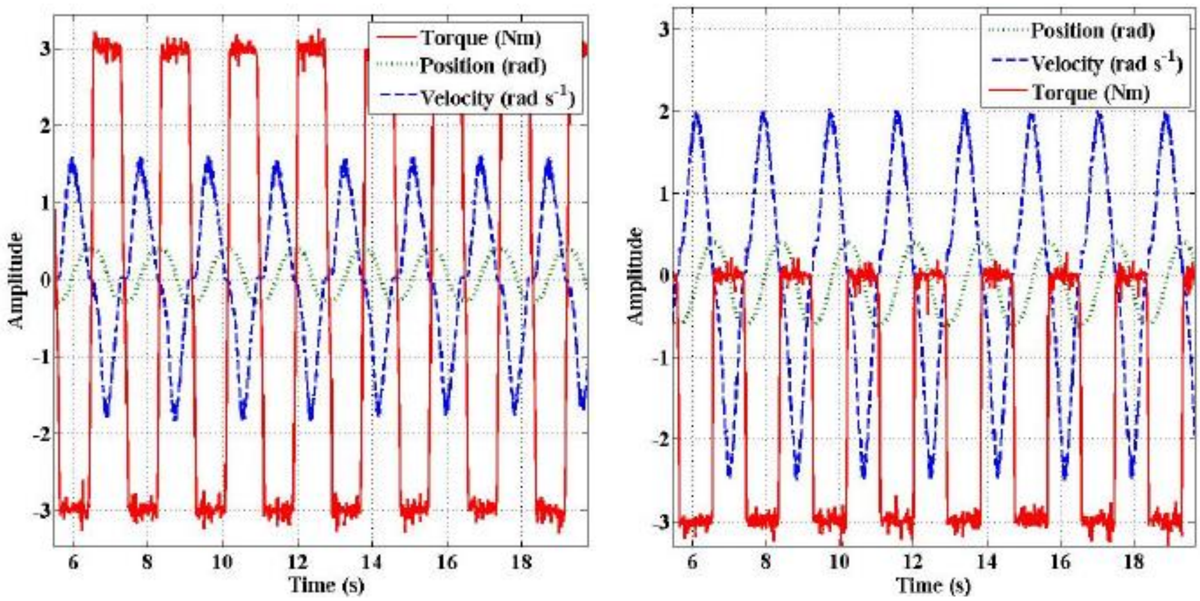


Figure 41. (Left) A square-wave damping strategy implemented by the FFD system. (Right) A directionally biased square wave damping strategy which is only activated when the flap moves in a certain direction.

14.3 Advantage and Disadvantages

<u>Advantages</u>

i. Excellent square-wave damping torque profile.
ii. Very controllable and repeatable damping profile.
iii. Damping can be digitally controlled outside of the wave tank.
iv. Damping can be controlled dynamically and reactive on a wave-by-wave basis.
v. Advanced and complex PTO control system strategies can be implemented, including those that require reactive power.
vi. The damping torque of the equipment range spans a very wide range of scale tests.
vii. Can be used for Forced Oscillation Tests and enable other fundamental research to be conducted.

<u>Disadvantages</u>

i. Very expensive, > £20,000.
ii. The main 3-phase driving motor cannot be directly used under water. Thus, an external support frame is required.
iii. Due to the sensitive nature of the control feedback loop, the system has a very low tolerance to mechanical misalignment anywhere in the system. This makes model set-up and commissioning very time-consuming and could destabilise implemented control algorithms.
iv. User requires good working knowledge of PLC programming.
v. Large and heavy equipment results in robust lifting and handling procedures.
vi. System cannot be used in conjunction with a foundation load measuring arrangement.
vii. Is not adaptable to use on multiple flaps and so no economies of scale exist in the equipment.

14.4 System Evaluation

Category	Description	Feature
Damping Type	e.g. Friction, Hydraulic actuation, Electric actuation	Electric
Geometric Configuration	How the damping is applied - Linear or Rotary resistance	Rotary
Damping Profile	Constant/Linear/Quadratic/Other damping	Constant
Type of Control	How the damping level is controlled – Manual/Automatic	Automatic
Max Damping [Nm]	Rating of the motor, belt gearing can double this to 66Nm	33 Nm
Min Damping [Nm]	Dictated mostly by bearing friction	<0.1 Nm

Assessment Criteria	Score
Repeatability	Good
Ease of Use/Control	Good
Control outside tank (Y/N)	Y
Dynamic Control (Y/N)	Y
Ease of use under water	Poor
Lifespan / robustness	Good
Cost (approximate)	£20,000
Size	250 mm Motor Size 1000 mm support stand

Key:

Good
Medium
Poor
N/A

Table 11. Specification and assessment of the Force Feedback Dynamometer system.

15 Magnetic Couplings

As presented in Sections 12 and 13, the Electrical Hysteresis Brake (EHB) and Magnetic Particle Brake (MPB) deliver very controllable and repeatable damping and satisfied almost all of APL's wave tank damper requirements, except that they are not inherently waterproof, unlike a solution such as the Disc Brakes for example (Section 6). So, as part of the IDCORE project (which assessed many PTO damper systems) an innovative and novel waterproofing solution was developed which is compatible with a rotary damping mechanism. Further details of this are also presented and discussed in [R2] and [R6].

It was decided to avoid radial shaft seals as a waterproofing solution, due to the added friction and limited life associated with them. An alternative was to use magnetic couplings. These are a pair of magnetic hubs that transmit torque across an air gap. These can be separated by a non-magnetic 'containment barrier', allowing isolation from liquids. One hub would be connected to the MPB shaft and with both components then housed in a waterproof box. The other hub would be connected to the WEC prime mover (i.e. a flap model) and immersed in the wave tank. Two types of magnetic coupling were considered: 'disc' and 'co-axial' and Figure 42 shows an example of the disc coupling arrangement to give and idea of the working principle of the system.

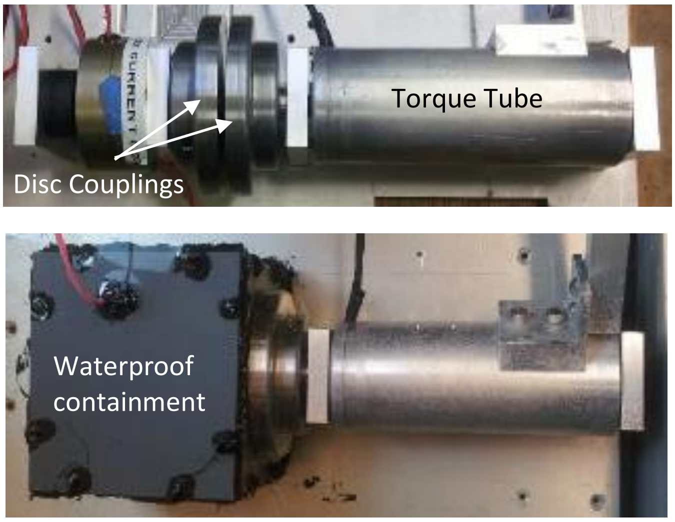


Figure 42. Magnetic Disc couplings attached to torque tube bench-test rig (top). The test rig with one of the magnetic hubs sealed inside a waterproof containment box (bottom).

Disc couplings were extensively tested first under the IDCORE project. MTD-5 disc couplings with a diameter of 97.5 mm were purchased from Magnetic Technologies Ltd. with a torque rating of 5 Nm. It was found that although the waterproofing principle of the system was well demonstrated, this system suffered from 'backlash', i.e. a 'looseness' or compliance in the magnetic system when the flap changes direction. During testing, this was measured to be circa 4^{o} and was observed even when the torque was a low as 1.7 Nm. A possible solution to this would be to increase the rating of the couplings to say 10 Nm. However, this would also increase the diameter of the couplings, making it more difficult to integrate into scale models and would exert a very high axial thrust load on the shaft of the MPB.

Magnetic Technologies also sell a magnetic coupling set called 'co-axial magnetic couplings, which overlap along the axis of the shaft. A 'containment barrier' can also be purchased, to separate the inner and outer magnetics couplings. An illustration of these is shown in Figure 43. The model 'MTC-10' of co-axial couplings has a nominal torque of 10 Nm, double that of the tested MTD-5 disc couplings. Even with this greater rating, the diameter of the outer magnetic hub for the MTC-10 option is smaller at 90 mm.

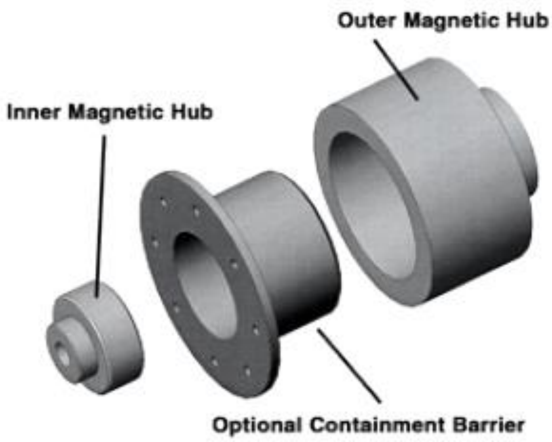


Figure 43. Co-axial magnetic couplings

These co-axial couplings were the preferred solution due to their higher torque rating and reduced geometric size. They were used and very successfully tested in the physical scale WEC model developed as part of the IDCORE project. The inner magnetic hub was connected to the MPB damper shaft, using an adaptor plate and bolts. This sub-assembly was then housed in a waterproof box with a plastic containment barrier as the lid; the outer magnetic hub rotated around the inner hub and was connected to the torque sensor shaft (see [R2] and [R6] for more details).

Thus, it was found that for a rotating damper system, co-axial magnetic couplings provide a very robust solution to waterproofing non-waterproof equipment. This opens up the possibilities of integrating more sophisticated damper mechanisms into small scale WEC models.

16 Appendix – Damping Torque Formulation

16.1 Constant (Coulomb) Damping

An ideal constant damping torque (T) can be expressed mathematically as:

$$T = -sgn(\dot{\theta})A \tag{1}$$

where *sgn* is the sign-function, $\dot{\theta}$ is the flap's angular velocity and *A* is a constant which denotes the amplitude of the torque signal. This expression will result in a perfect square-wave time trace whose periodicity depends on the WEC's velocity. Such a binary (on/off) type signal often occurs in classical friction systems and so this type of damping strategy is sometime referred to as Coulomb damping.

16.2 Linear Damping

Linear damping is the term given to a damping strategy where the magnitude of the torque/force depends linearly on the velocity of the WEC/PTO prime-mover

$$T \propto \dot{\theta} \tag{2}$$

16.3 Quadratic Damping

Similar to the definition of linear damping, quadratic damping is where the magnitude of the damping torque is proportional to the square of the WEC/PTO prime-mover velocity

$$T \propto \dot{\theta}|\dot{\theta}| \tag{3}$$

17 References

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