Mooring Solutions for Large Wave Energy Converters

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DOI (link to publication from Publisher):
10.5278/vbn.phd.eng.00045

Publication date:
2017

Document Version
Publisher's PDF, also known as Version of record

MOORING SOLUTIONS FOR LARGE WAVE ENERGY CONVERTERS

BY
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DISSERTATION SUBMITTED 2017

AALBORG UNIVERSITY
DENMARK

Mooring Solutions for Large Wave Energy Converters

Ph.D. Dissertation

Jonas Bjerg Thomsen

Dissertation submitted December 2017

Dissertation submitted:	December, 2017
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ISSN (online): 2446-1636
ISBN (online): 978-87-7210-112-5

Published by:
Aalborg University Press
Skjernvej 4A, 2nd floor
DK – 9220 Aalborg Ø
Phone: +45 99407140
aauf@forlag.aau.dk
forlag.aau.dk

Printed in Denmark by Rosendahls, 2017

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Preface

The work in this thesis is the outcome of the past three years since September 2014, when I was employed as PhD student. During this time, I have had some of my most exciting, interesting, busy and fun years. Being surrounded by supporting people and having countless experiences have not only helped me develop as a researcher, but also as a person. For that, I am truly grateful.

Firstly, I would like to thank my two supervisors, Jens Peter and Francesco. Jens Peter gave me the opportunity of becoming PhD student and has been providing valuable guidance in my work. I appreciate his trust in my efforts, for letting me be part of other exciting projects and for listening to me when I needed it. Francesco has been my closest collaborator in this project, and I have had tremendous benefit of his guidance, knowledge and feedback. I am genuinely thankful for his supervision and involvement in my work.

My greatest appreciation is aimed at my colleagues at the Department of Civil Engineering. I am thankful to be surrounded by people who are not only my co-workers, but also my friends. I appreciate every constructive, work-related discussion and every social experience we have had together. Without their support and feedback, the outcome of the past years would have been much different. A great thank to the technicians and secretaries for aiding in the experimental and administrative work, with a special thank to Vivi Søndergaard for her help on correcting all my past and present work. I would like to extend my acknowledgement to all partners in the MSLWEC project for feedback and for taking part in many fruitful discussions. Thanks to everybody I have met during the past years at seminars, conferences, and INORE. Most of you have had some kind of impact on my work.

Finally, I would like to thank my family and friends outside the university. The last three years have been tough and greatly focussed on research, but I have felt the support from everybody. Without you, this thesis would not have been possible.

Jonas Bjerg Thomsen
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Jonas Bjerg Thomsen has a M.Sc. degree in Coastal Engineering from Aalborg University and has since September 2014 been employed as a Ph.D. student. His current research revolves around mooring systems and modelling of hydrodynamic behaviour of floating structures. His research has been aimed at optimizing mooring layouts with respect to cost and improved reliability by investigating and incorporating alternative materials and standardized design procedures. His research covers both experimental work and the corresponding analysis of data, together with validation, incorporation and use of numerical models. In addition to his research, he has also taken part in teaching and supervision of both bachelor and master degree students.

Thesis Details

Thesis Title: Mooring Solutions for Large Wave Energy Converters
Ph.D. Student: Jonas Bjerg Thomsen
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Project Mooring Solutions for Large Wave Energy Converters (Forankringsløsninger for store bølgekraftanlæg), funded by the Energy Technology Development and Demonstration Program (EUDP) (Grant number 64014-0139)

The main body of this thesis is composed of a collection of the following eight papers.

[A] Thomsen, J. B., Ferri, F., and Kofoed, J. P. (2015). Assessment of Current State of Mooring Design in the Danish Wave Energy Sector. In *Proceedings of the 11th European Wave and Tidal Energy Conference EWTEC2015*. Technical Committee of the European Wave and Tidal Energy Conference

[B] Thomsen, J. B., Kofoed, J. P., Delaney, M., and Banfield, S. (2016c). Initial Assessment of Mooring Solutions for Floating Wave Energy Converters. In *The 26th International Ocean and Polar Engineering Conference ISOPE2016*. International Society of Offshore and Polar Engineers

[C] Thomsen, J. B., Ferri, F., and Kofoed, J. P. (2016b). Experimental Testing of Moorings for Large Floating Wave Energy Converters. *Progress in Renewable Energies Offshore RENEW2016*

[D] Thomsen, J. B., Ferri, F., and Kofoed, J. P. (2017c). Screening of Available Tools for Dynamic Mooring Analysis of Wave Energy Converters. *Energies*, 10(7)

[E] Thomsen, J. B., Ferri, F., and Kofoed, J. (2017b). Validation of a Tool for the Initial Dynamic Design of Mooring Systems for Large Floating Wave Energy Converters. *Journal of Marine Science and Engineering*, 5(4)

[F] Thomsen, J. B., Kofoed, J. P., Ferri, F., Eskilsson, C., Bergdahl, L., Delaney, M., Thomas, S., Nielsen, K., Rasmussen, K. D., and Friis-Madsen, E. (2017f). On Mooring Solutions for Large Wave Energy Converters. In *Proceedings of the 12th European Wave and Tidal Energy Conference EWTEC2017*. Technical Committee of the European Wave and Tidal Energy Conference

[G] Thomsen, J. B., Ferri, F., Kofoed, J. P., and Black, K. (2017d). Cost Optimization of Mooring Solutions for Large Floating Wave Energy Converters. *Under review for publication in Energies*

[H] Thomsen, J. B. and Andersen, M. T. (2018). Sensitivity and Cost Analysis of Mooring Solutions for Large Renewable Energy Structures. *Submitted to the 37th International Conference on Ocean, Offshore & Arctic Engineering OMAE2018*

In addition to the main papers, a number of papers and reports have been produced and are listed below, but not appended to this thesis. The publications are an extension of the work presented in the main body and focuses on experimental and offshore testing, mooring analysis and design.

[1] Andersen, M. T. and Thomsen, J. B. (2014). *The Inter Facility Testing of a Standard Oscillating Water Column (OWC) Type Wave Energy Converter (WEC)*. Department of Civil Engineering, Aalborg University

[2] Thomsen, J. B. (2015). *Scale 1:10 Test of the OWC WEC LEANCON at Nissum Bredning*. Department of Civil Engineering, Aalborg University, Denmark

[3] Thomsen, J. B., Ferri, F., and Kofoed, J. (2016a). *Current Mooring Design in Partner WECs and Candidates for Preliminary Analysis: CM1 & M3*. Aalborg University, Department of Civil Engineering. Confidential report

[4] Thomsen, J. B. (2017). *Validation of Mean Drift Forces Computed with the BEM Code NEMOH*. Department of Civil Engineering, Aalborg University

[5] Thomsen, J. B. and Ferri, F. (2017). *Full Dynamic Analysis of Mooring Solution Candidates - First Iteration: T4.3 & M6*. Department of Civil Engineering, Aalborg University

[6] Thomsen, J. B., Eskilsson, C., and Ferri, F. (2017a). *Assessment of Available Numerical Tools for Dynamic Mooring Analysis: WP1.2 & M1*. Department of Civil Engineering, Aalborg University

[7] Nielsen, K., Bingham, H., and Thomsen, J. B. (2018). On the Absorption of Wave Power Using Ship-Like Structures. In *The 28th International Ocean and Polar Engineering Conference ISOPE2018*. International Society of Offshore and Polar Engineers. (Submitted)

Abstract

The ever-increasing population and living standard have put a rising demand on the global energy production. Today, the energy mix is primarily dominated by fossil fuels, but as the demand increases and the finite fossil resources are exhausted, it becomes critical to implement renewable energy supplies into the mix. With the increasing focus on environmental sustainability, security of jobs and energy security and equity, it is beneficial not only to improve the established renewable energy sources, but also to develop and introduce those resources that are not yet commercially mature. The wave energy sector provides a significant global energy potential, but is not yet providing any contribution to the energy mix.

Since the first patented wave energy device, many different concepts have been developed, but none have yet managed to reach a level of maturity, which allows for feasible offshore deployment for energy production. This is both resulting from a cost of energy, which is not yet competitive to other energy resources, but also from failures of many of the early-stage deployments. To some extent, both of these factors can be linked to the topic of moorings as the cost of these has been shown to take up a considerable part of the total cost and as several mooring failures have been experienced so far.

The present work focusses on mooring solutions for large floating wave energy converters and aims at developing guidelines and procedures used for designing cost-efficient and reliable solutions. The work takes its basis in four large wave energy converters and their initially proposed mooring solutions in order to clarify how moorings have been designed, and uses it to discuss promising layouts and materials. The applicability of different types of numerical models for initial mooring design is investigated by validation against experimental data, with quantification of the accuracy of the capability to model line tensions. The numerical model is further implemented in an optimization routine, which allows for fast evaluation of an optimized and reliable mooring layout. This is used to design three different types of mooring systems and compare cost and sensitivity in order to find pros and cons from each system. Consequently, the presented work aids in early stage design of moorings and provides a methodology, which finds durable and cost-efficient solutions.

Resumé

Det voksende globale befolkningstal, og en levestandard i konstant udvikling, har medført et stigende krav til verdens energiproduktion. I dag er fordelingen i energiforsyningen primært domineret af fossile brændstoffer, men i takt med at efterspørgslen stiger, og de begrænsede fossile brændstoffer opbruges, er det kritisk, at der implementeres vedvarende energikilder i forsyningen. Med et stigende fokus på miljømæssig bæredygtighed, fastholdelse af arbejdspladser, energisikkerhed og ligelighed i adgangen til energi, er det gunstigt både at forbedre de etablerede energiproduktioner, men også at udvikle og implementere de ressourcer, som endnu ikke er kommercielt udviklede. Bølgeenergisektoren udgør her et betydeligt globalt energipotentiale, men er stadig ikke aktiv i verdens energiproduktion.

Siden det første patenterede bølgeenergikoncept er der udviklet mange forskellige koncepter, men ingen er endnu lykkedes i at nå et niveau, der tillader rentabel offshore installation og energiproduktion. Dette er både et resultat af en energipris, der aktuelt ikke kan konkurrere med de øvrige energikilder, men også pga. svigt af flere koncepter i tidelige udviklingsfaser. Begge faktorer kan til en vis grad knyttes til koncepternes forankringsløsninger, da prisen for disse har vist sig at optage en betydelig del af den samlede pris, og da adskillige svigt er forekommet i sektoren som følge af brud i forankringen.

Det forelagte arbejde fokuserer på forankringsløsninger til store bølgeenergianlæg og søger at udvikle retningslinjer og procedurer for design af priseffektive og pålidelige løsninger. Arbejdet tager sit udgangspunkt i fire store bølgeenergianlæg, og de løsninger som de oprindeligt har foreslået, for at klarlægge hvordan forankring tidligere er blevet designet, og for at diskutere potentielle løsninger og materialer. Anvendeligheden af forskellige typer numeriske modeller til initierende forankringsdesign bliver undersøgt og valideret mod eksperimentelt data for at kvantificere præcisionen i at modellere linekræfter. Den numeriske model bliver ydermere implementeret i en optimeringsrutine, der tillader identificering af optimale og pålidelige systemer.

Denne rutine anvendes til design af tre forskellige forankringssystemer og sammenligner pris og sensitivitet for at identificere fordele og ulemper ved hvert system. I konsekvens heraf, understøtter det forelagte arbejde i initierende design af forankringssystemer og bidrager med en metode til at finde pålidelige og priseffektive løsninger.

Nomenclature

Abbreviations

ALS	Accidental Limit State
BEM	Boundary Element Method
CAPEX	Capital Expenditure
CFD	Computational Fluid Dynamics
CO_2	Carbon Dioxide
DNS	Direct Numerical Simulation
DoE	Design of Experiment
DoF	Degree of Freedom
FE	Finite Element
FD	Finite Difference
FLS	Fatigue Limit State
FOWT	Floating Offshore Wind Turbine
LCOE	Levelised Cost of Energy
LES	Large Eddy Simulation
LF	Low Frequency
MEC	Marine Energy Converter
MoI	Moment of inertia
O&G	Oil & Gas
O&M	Operation & Maintenance
OPEX	Operational Expenditure
OT	Overtopping
OWC	Oscillating Water Column
PTO	Power Take Off

QTF	Quadratic Transfer Function
RANSE	Reynold-averaged Navier-Stokes Equation
RAO	Response Amplitude Operator
SPH	Smoothed Particle Hydrodynamics
SPM	Single Point Mooring
SWL	Still Water Level
TPL	Technology Performance Level
TRL	Technology Readiness Level
ULS	Ultimate Limit State
WAB	Wave Activated Body
WEC	Wave Energy Converter
WEU	World Energy Use
WF	Wave Frequency

Latin Nomenclature

A_b	Body area
$\boldsymbol{C_{A\infty}}$	Added mass at infinity frequency
$\boldsymbol{C_A}$	Frequency-dependent added mass matrix
C_a	Added mass coefficient
$\boldsymbol{C_B}$	Frequency-dependent radiation damping matrix
C_D	Drag coefficient
C_{drift}	Drift coefficient
D	Characteristic body length
F_B	Net buoyancy loads
F_c	Current load
f_D	Time-dependent viscous drag force
$\overline{F}_d$	Frequency-dependent diffraction force vector
f_{exc}	Time-dependent 1$^{\text{st}}$ order wave excitation force
$\overline{F}_{exc}$	Frequency-dependent excitation force vector
F_{ext}	External environmental loads
f_{FK}	Time-dependent Froude-Krylov force
f_I	Time-dependent inertia force
F_{int}	Internal loads

f_m	Time-dependent mooring load
f_n	Natural frequency
f_o	Objective function
f_r	Time-dependent radiation load
$\overline{F}_r$	Frequency-dependent radiation force matrix
F_w	Wind load
H_k	Kochin function
H	Wave height
I_{ii}	Mass moment of inertia in *ii* DoF
IRF_r	Impulse response function of radiation force
$\boldsymbol{K_{hyd}}$	Hydrostatic stiffness matrix
K_m	Linearized horizontal mooring stiffness
k_{moor}	Axial mooring line stiffness
L	Wave length
$\boldsymbol{M}$	Body mass matrix
Q_d	Frequency-dependent 2$^{\text{nd}}$ order wave load
s	Surrogate model
u	Fluid displacement
v	Body displacement
V_b	Body volume
$\boldsymbol{x}$	Variable vector

Greek Nomenclature

ϵ	Surrogate model error
γ	Partial safety factor
ω	Angular frequency
Φ	Fluid velocity potential
Φ_d	Diffracted fluid velocity potential
Φ_i	Incident fluid velocity potential
Φ_r	Radiated fluid velocity potential
Φ_s	Scattered fluid velocity potential
ρ	Fluid density

1 Introduction

This thesis handles the topic of mooring solutions for large Wave Energy Converters (WECs) and is greatly motivated by the desire to bring wave energy closer to commercialization, to facilitate the use of sustainable energy resources and to satisfy the increasing need of energy throughout the world.

The population of the world has been in constant development for many years, both in terms of population size and living standards, which has now resulted in an energy consumption that has increased with a yearly average of 1.9% for the past 10 years. This creates a growing demand on the exploration of available energy sources, which so far, fossil fuels have been dominating. In most of the world, however, the energy supply is now transforming from the well-known and used fossil resource and into renewable energy sources, partly because of a desire for more sustainable energy. As stated by IRENA (2017), by the end of 2015, 173 of the 195 countries in the world had formulated renewable energy targets at national level and 146 had agreed on renewable support policies. More importantly, 193 countries signed the Paris Agreement in 2016 and the remaining two countries signed in 2017, establishing a global attempt to decrease global warming in large part by focussing on renewable energy. Sadly, in 2016 USA announced its intention to withdraw from the agreement and, thereby, be the only country in the world who is

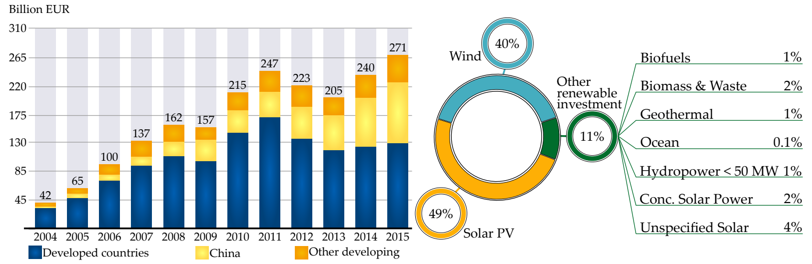


Fig. 1.1: Investment in the renewable energy sector over time (left) and by technology (right). Based on values presented in IRENA (2017)

not a signatory (United Nations, 2017). Globally, this focus has resulted in an investment in renewable energies as presented in Fig. 1.1. As seen, there has been an increasing trend in the investments up until 2011, which was followed by a few years of decrease. This was partly caused by a decrease in the investments from the developed countries, but nevertheless, the investments afterwards seem to be increasing.

The motivation for the transformation towards renewable energy has been listed by e.g. MacKay (2008) as three focus points, *Environmental Sustainability*, *Energy Equity* and *Energy Security*, and has been named by World Energy Council (2016) as the *Energy Trilemma*. As a measure of each country's efforts towards sustainable energy solutions, World Energy Council (2016) introduced the Energy Trilemma Index, which has been used to rank countries from the most active in achieving sustainable solutions (Denmark, Switzerland and Sweden) to the countries with largest problems (Niger, Nepal).

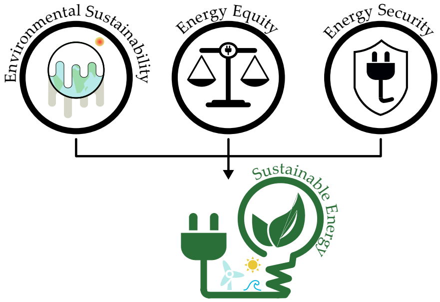


Fig. 1.2: Illustration of the energy trilemma.

In the last couple of decades, the impact on the environment and climate from fossil fuels has become the attention of much discussion and controversy. Today, this forms one of the greatest arguments for the transformation towards use of renewable energy resources, as these are generally considered

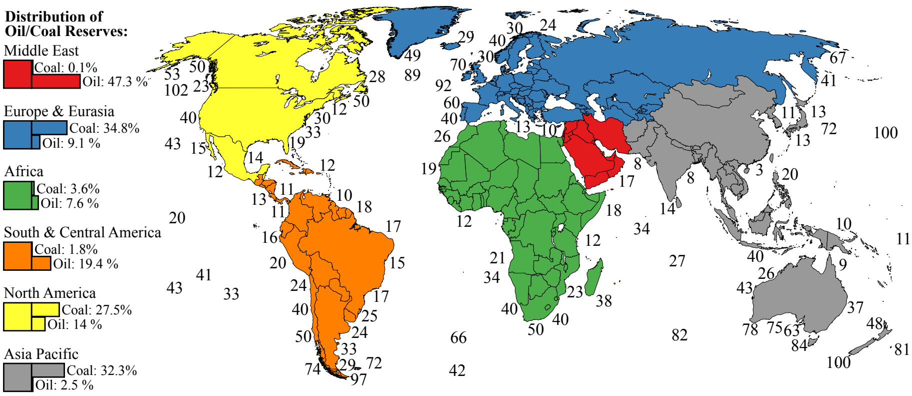


Fig. 1.3: Worldwide distribution of oil and coal reserves based on key numbers from BP (2016). The numbers on the map indicate the average available wave power per meter (kW/m). Based on Sea Power (2017) and Mørk et al. (2010).

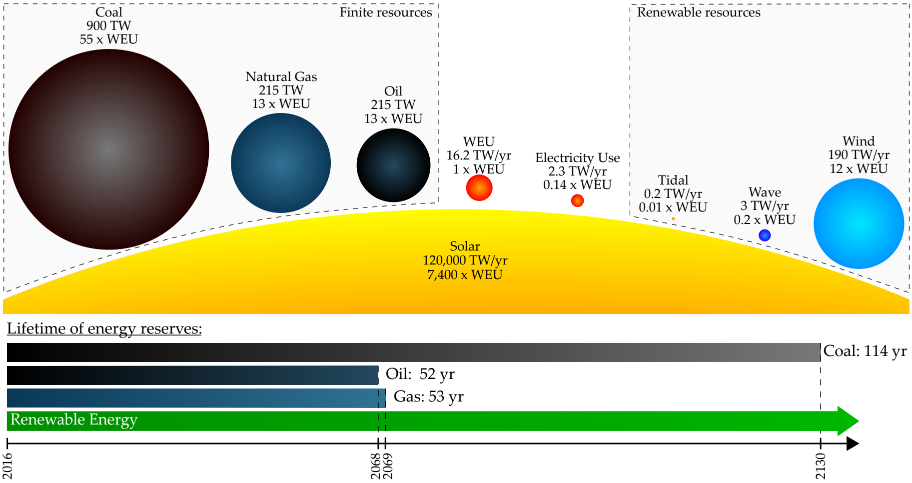


Fig. 1.4: Potential of selected energy resources based on values given in Johansson et al. (2004), Perez and Perez (2009) and Pecher and Kofoed (2017). The figure also illustrates the lifetime of energy reserves as given in BP (2016). The renewable energy potentials are defined as theoretical potentials, while the finite resources are based on proven reserves.

to ensure a higher level of *environmental sustainability*. Use of fossil fuels has a great correlation with the emission of greenhouse gasses such as Carbon Dioxide (CO_2), which has been identified as a cause of climate changes. Considering just the last 12 years, NASA (2017) states that the level of CO_2 has increased from 379 ppm in the atmosphere to 406 ppm in early 2017. In the same period, the average annual temperature increased by 0.6°C and resulted in an average sea level rise of 43 mm. If this tendency continues over the next period of time, the predictions estimate an average sea level rise of 0.5-1.3 m by the time of the year 2100 with crucial consequences. In addition to the climate impact, the extraction of fossil fuels has a very unfortunate effect on the environment through heavy pollution of the air, water and soil.

A paramount problem with an energy production mainly conformed by fossil fuels, is a very poor level of *energy equity*. By 2016, up to 17% of the world population is estimated to be without access to electricity (IRENA, 2017), and the problem is highlighted by the fact that three countries possess close to 60% of the world coal reserves, while three others own 50% of the oil reserves (BP, 2016). Considering Fig. 1.3, which illustrates the worldwide distribution of oil and coal reserves, it is seen how the fossil fuels are unevenly distributed and, consequently, allows a few developed areas access to a large part of the reserves, while others have very limited access. Not surprisingly, this highly affects the politics and price for energy, not to say the risk of conflicts as some countries are dependent on supply from others.

A beneficial factor in introducing new renewable energy sectors, particularly in parts of the world where the energy sector previously was limited, is the production of a large amount of new jobs. According to REN21 (2016) there is a significant development in jobs in the renewable sector at present, with an increase of 5% from 2014 to 2015 resulting approximately in a total of 8.1 million direct and indirect jobs worldwide.

Energy security is a great motivation for renewable energy as it treats the problem of ensuring that the energy resources can meet the current and future demands. Furthermore, it helps decrease most countries' dependency on energy supply from other countries and eliminate the risk for energy conflicts. The finite fossil resource will eventually result in a change into other energy sources. Despite a refinement in the extraction methods due to technology development and higher efficiency, the fossil fuels are limited and will completely run out within a relatively short time, cf. Fig. 1.4. By continuing the consumption of oil and gas resources as it is today, BP (2016) estimate that the reserves are completely consumed by the year 2069, cf. Fig. 1.4. The coal reserves will be consumed within 114 years at the current consumption rate. In fact, it might not even be cost beneficial to extract all resources, due to the complexity of such, and making a gradual change from the fossil fuels into alternative solutions will eventually ease the transition into renewable sources.

The benefits of switching to renewable energies are many. Considering the Energy Trilemma, renewable energy secures a high level of security due to the infinite resource, the equality is high as the availability and distribution of resources are more even, and finally switching from fossil fuels usage to renewable energies has a very positive influence on the emission of greenhouse gasses and the environmental impact.

Renewable energy covers a large variety of different energy sources – some more developed and implemented in the energy production than others. In 2015, renewable energy accounted for approximately 18.3% of the

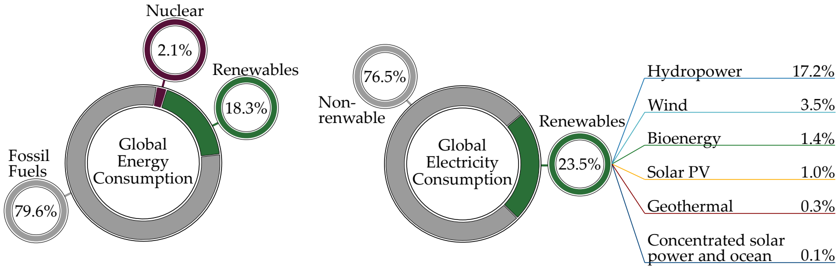


Fig. 1.5: Distribution of energy and electricity consumption by energy type (IRENA, 2017).

total energy consumption (REN21, 2016), with the largest yearly increment of growth in renewable power generation on record (BP, 2016). Considering only the worldwide electricity consumption, 23.5% was generated by renewable resources, with a distribution as illustrated in Fig. 1.5.

When observing the energy mix in Fig. 1.5, it is clear that the renewable sector needs further development before the fossil fuel consumption will decrease. Further development on the established resources is needed, but also inclusion of resources that are not as developed by now. A sector, which currently does not contribute to the energy mix, is the wave energy sector also seen from Fig. 1.5.

1.1 Wave Energy

As shown in Fig. 1.4, the wave energy is not the most dominant resource, but it has the potential to be a considerable contributor, especially against the finite resources of oil, coal and natural gas. The total available wave energy potential is approximated to 3 TW/yr, which is nearly 20% of the World Energy Use (WEU) (Mørk et al., 2010; Pecher and Kofoed, 2017). Considering only the electricity consumption (2.3 TW/yr), wave energy could potentially cover this completely. However, this implies 100% utilization which will not be achieved, but even 10-20% could pay a considerable contribution and forms a strong argument for investigation and improvement of the resource.

The average available wave energy along the coasts of the world is illustrated in Fig. 1.3, and this presents another benefit from wave energy. Compared to the concentrated fossil resources, wave energy is much more evenly distributed throughout the countries of the world, and the energy from the oceans is transported directly to the coasts at which a large part of the population is concentrated. This also ensures a high level of predictability and provides a great supplement to wind energy as waves occur even when no significant wind resource is present. Finally, waves have a large power density, which is good from an energy point of view, but it puts more demand on the structures.

It is obvious that the implementation of wave energy into the energy mix can help improve the Energy Trilemma. The resource not only ensures energy security through its renewable nature, but also helps provide energy equity and sustainability. In spite of this, the wave energy sector is still at an early stage, where no commercial devices have yet been deployed. Nevertheless, many different developers exist today (more than 200 by 2017 (EMEC, 2017)), and much research is conducted throughout the world.

WECs differ significantly by shape, size and working principle, and traditionally they are characterized by their working principle as listed below and illustrated in Fig. 1.6.

- Overtopping (OT)
- Oscillating Water Column (OWC)
- Wave Activated Body (WAB)

This characterization is described by e.g. Falcão (2010), Karimirad (2014) and Pecher and Kofoed (2017), which also introduce characterizations based on location (offshore, near-shore and onshore) and the extension of the structure according to the wave propagation (attenuator, terminator and point absorber). As suggested from the title of this thesis, only devices that can be considered as large floating WECs are considered. Nevertheless, they can all be characterized differently according to the previous definitions.

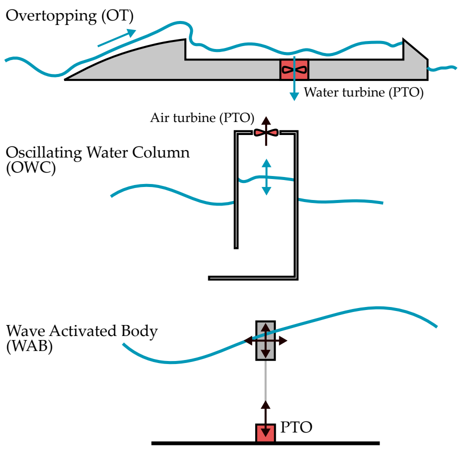


Fig. 1.6: Simple sketch of three different working principles of WECs. The layout and size can differ significantly within each principle.

1.1.1 Large Floating WECs

Large floating WECs are the type of devices closest to the well-known floating Oil & Gas (O&G) structure due to their large masses and requirement to response. The structure, in most cases, consists of a floating foundation at which the wave energy Power Take Off (PTO) is installed. This means that the floating foundation in itself is not supposed to be in resonance with the wave,

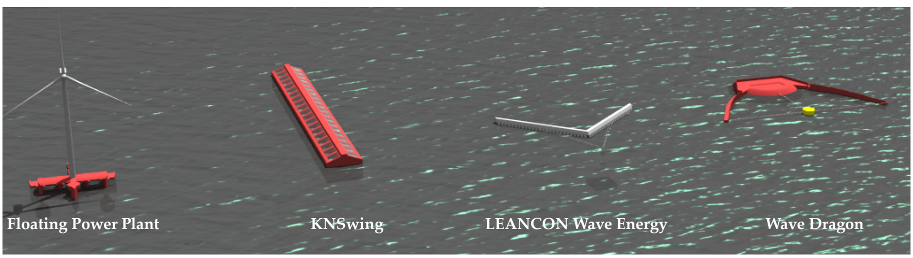


Fig. 1.7: Examples of large floating WECs, which is used for case studies in this thesis.

but rather provide a stable foundation for the PTO which on the contrary can be resonating for optimal power production.

Several large floating WECs are being developed like e.g. the Danish devices Floating Power Plant (Floating Power Plant, 2017), KNSwing (Nielsen and Bingham, 2015), LEANCON Wave Energy (LEANCON, 2017) and Wave Dragon (Wave Dragon, 2017), cf. Fig. 1.7. These form the primary focus of this thesis and act as case studies for the research with the expectation that the findings can be applied to similar types of WECs. Examples of such could be the WEPTOS (WEPTOS, 2017) and WavePlane (WavePlane, 2017).

Obviously, the four devices use different principles for wave energy absorption ranging from the use of OWC in LEANCON and KNSwing, to WAB in Floating Power Plant and to OT in Wave Dragon. Despite this significant difference between them, they all have in common that they are equipped with a safety system, hence the PTO is disabled during extreme events and, therefore, without influence on the response. Furthermore, they are planned for deployment under similar environmental conditions as explained in Chapter 3, for which reason the same design approach expectedly could be used for all.

1.2 Current Status of Wave Energy

Different terminologies are used to describe the maturity of the wave energy concepts. Weber (2012) introduced the matrix illustrated in Fig. 1.8 where the WECs are defined by their Technology Readiness Level (TRL) and Technology Performance Level (TPL). This allows for a measure of the current stage of development, by means of the amount of funding needed in order to obtain a commercial device and by means of the current Levelised Cost of Energy (LCOE).

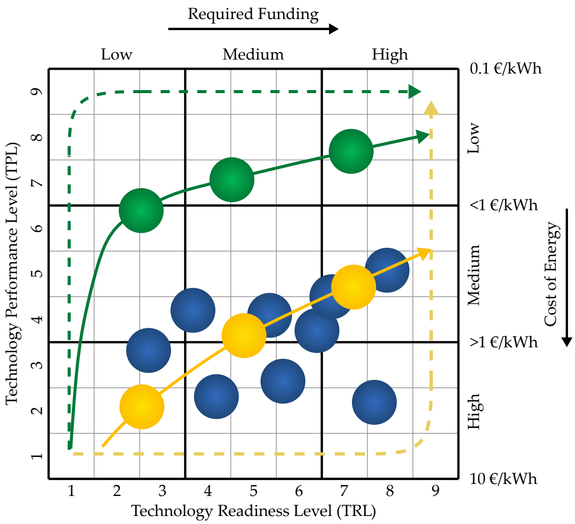


Fig. 1.8: The TRL/TPL matrix as defined by Weber (2012). Green dots represent a viable trajectory, while the yellow represent the current development. The blue dots present patented WECs and the dashed lines provide examples of extreme cases.

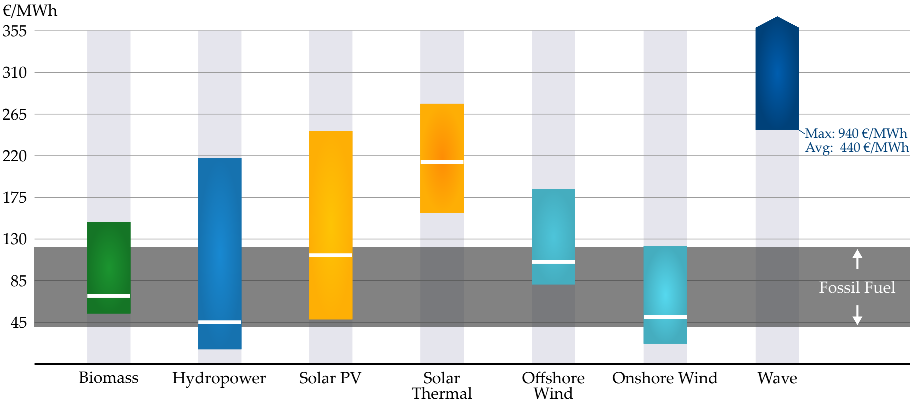


Fig. 1.9: Estimated LCOE ranges for different energy sources, based on World Energy Council and Bloomberg New Energy Finance (2013), Carbon Trust (2011), IRENA (2017) and OES & IEA (2017). The white lines indicate the weighted average.

The first patent submitted for a wave energy concept dates back to 1799 and since then much research and development has been conducted, starting intensely in the 1970s during the oil crisis. Despite of this, there are still no commercial WECs, and according to Magagna et al. (2016) and Pecher and Kofoed (2017), the maximum achieved TRL for a wave energy converter by 2016 is level 8, with no considerable development in the TRL since 2014. The dashed lines in Fig. 1.8 resemble extreme trajectories, where the green indicates focus on performance before readiness, while the yellow indicates the opposite. The blue dots represent a number of patented WECs and the yellow dots present the development trajectory applied in the wave energy sector so far. According to Weber (2012), the green dots represent a realistic trajectory, which secures the goal of reaching a commercial state. It is observed how the wave energy sector has put most focus on TRL, resulting in TPLs that are not yet commercially mature.

A major problem for the development of wave energy is clearly the current LCOE (TPL) as it is still excessively high. Considering Fig. 1.9, the cost of wave energy is indeed noticeably higher than for fossil fuels and even other renewables.

There is a clear need for lowering the LCOE of wave energy and, therefore, several authors have tried to identify the drivers for such. In Carbon Trust (2011), six cost reduction drivers for wave energy farms were investigated:

- Structure
- Operation & Maintenance (O&M)
- PTO

- Station keeping
- Grid connection
- Installation.

From the list, optimization of structure, O&M and PTO were expected to provide the largest cost reduction, while it was stated that overall innovation in all areas by changing focus from well-known concepts from the O&G sector was needed in order to lower the LCOE.

Another study by Low Carbon Innovation Coordination Group (2012), found that improvement of moorings can play a significant role in the reduction of the LCOE, as it was predicted that the cost of moorings will be reduced by up to 50% by 2025 and, thereby, be a more dominant cost reduction factor than e.g. PTOs. This reduction is expected to arise from improved deployability and by discarding the use of conventional mooring techniques and by considering more innovative solutions. Carbon Trust (2011) and Weller et al. (2013) estimate potential LCOE reductions of 5-10% by optimization of moorings.

1.2.1 Cost of WECs

Fitzgerald (2009) presented a Capital Expenditure (CAPEX) and Operational Expenditure (OPEX) cost breakdown for a single buoy-type WEC as shown in Fig. 1.10. Clearly, cost of the structure and O&M are the most dominating parameters, but it is critical to notice how large a part of the cost that the mooring takes up. For large WECs, this might look different, but it evidently forms a focus point. In other publications such as Carbon Trust (2011), Low Carbon Innovation Coordination Group (2012), Neary et al. (2014) and Jenne et al. (2015), the percentage of the CAPEX from mooring has been estimated to be around 5-10%, with e.g. Martinelli et al. (2012) estimating mooring cost to take up to 30% of the total cost of a device.

Clearly, mooring is now a considerable contributor to the total cost of WECs with a strong argument for optimization. This, however, is not the only reason for research into moorings, as stated by Low Carbon Innovation Coordination Group (2012), since it is also crucial to improve durability and reliability. This will be illustrated further in the following section.

1.2.2 Deployed Moorings

Several floating WECs have reached a development stage that allows for larger scale testing in real offshore environment. Obviously, the mooring systems and its performance formed a topic for investigation and, therefore,

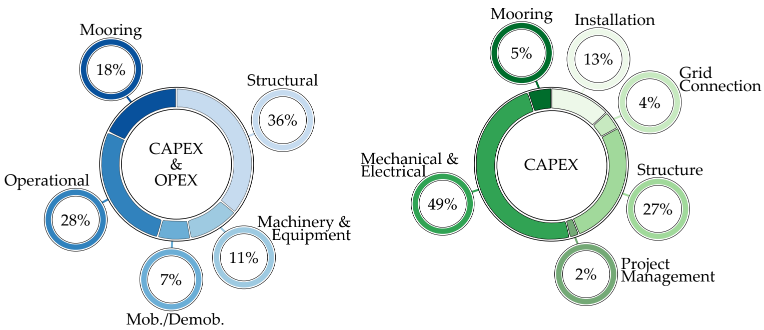


Fig. 1.10: Left: Cost breakdown of combined CAPEX and OPEX for a point absorber buoy, as defined in Fitzgerald (2009). Right: Cost breakdown of CAPEX for point absorber based on values found in Carbon Trust (2011); Low Carbon Innovation Coordination Group (2012); Neary et al. (2014); Jenne et al. (2015).

a realistic solution had to be applied and designed for survivability. Unfortunately, many devices failed due to insufficient mooring (Martinelli et al., 2012), some of which are pictured in Fig. 1.11.

Failure of moorings have caused significant damage to numerous devices and obstructed their further development. The failures seem to be caused by several factors from inadequate assessment of the environmental conditions, to underestimation of tensions and unfortunate exposure to unexpected

Fig. 1.11: Examples of strandings of WECs due to mooring failures. Pictures adapted from Nordyske (2011); Illawarra Mercury (2015)

loads, e.g. from ice. Furthermore, more than one device failed due to imperfect installation and lack of duly inspection. There is a critical need to assess the applied design procedures in order to evaluate more detailed why these failures occur and how they can be avoided. In some cases, the damages have resulted in a need to decommission the entire structure, resulting in a loss of finances and an unfortunate perception of wave energy in the society. As mentioned in the previous section, the desire to lower LCOE by optimizing cost of mooring is a great motivation, but the main challenge really is to ensure that the deployed systems are capable of surviving at the desired deployment sites, while keeping the cost as low as possible.

1.3 Objective and Outline of Thesis

There is a need to improve the wave energy sector and drive it closer to commercialization, for which reason it is important to address and to lower the LCOE. One parameter that can benefit on this matter is the mooring of floating WECs.

The previous sections presented the fact that the cost of mooring is undesirable high and, furthermore, that several failures have been experienced. Therefore, it is relevant to assess the current state of mooring design and to identify the areas, which can be improved, with regard to ensuring both a satisfactory design and a cost-optimized solution. The present thesis identifies the mooring status for four different wave energy converters and attempts to find optimized mooring solutions for each of these.

As a result of this, four points are considered important to address in the process towards optimized moorings for large, floating WECs:

- Assessment of the current state of mooring design in order to identify focus areas and cost drivers.
- Identification of potential cost and durability efficient mooring solutions.
- Identification of mooring design and analysis procedures, in terms of tools, methods and validation against experimental results.
- Economical optimization of the selected mooring solutions and assessment of the sensitivity of different mooring layouts.

1.4 Thesis Outline

This thesis is structured as a collection of eight papers and an extended summary consisting of the present six chapters, which describe the overall work and outcome. The thesis follows a structure as presented in Fig. 1.12.

The current chapter presents the overall topic of wave energy, with a description of present status and focus areas. The chapter highlights mooring as a factor to consider for increasing of durability, reliability and for decreasing LCOE, thereby presenting the motivation for this thesis.

In Chapter 2, the current and general state of the art for mooring design is considered. The chapter sheds light on the methods, solutions etc. considered by other authors.

Chapter 3 puts specific focus on the wave energy sector and the current state of mooring design. The work investigates the solutions and design procedures applied by a number of Danish WEC developers. The work is finally centred on identification of potential mooring solutions for further consideration.

Chapter 4 investigates the method of analysing moorings. The work on physical testing is presented, and the data is used to validate and investigate numerical models for initial mooring design.

Finally, Chapter 5 uses the outcome of the previous chapters to optimize mooring systems by means of cost of the systems. An optimization routine is presented, and the importance of different cost drivers are discussed. Coarse estimations on the potential impact on mooring cost and LCOE is presented.

Chapter 6 summarizes and concludes the work of the thesis and focuses on perspectives for future work.

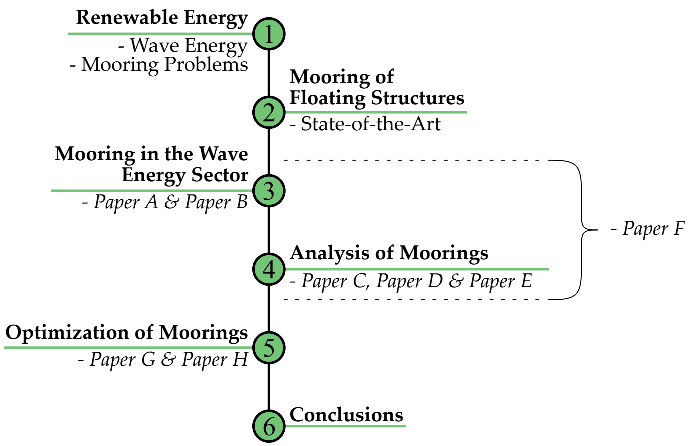


Fig. 1.12: Structure of the thesis.

2 Mooring of Floating Structures

All offshore structures are exposed to a range of environmental loads, primarily dominated by those arising from wind, wave and current, cf. Fig. 2.1. In order to ensure that the structure stays on station, it is necessary to apply a system that restrains motions. This can consist of several solutions, e.g. a dynamic positioning system, but often a mooring is applied.

Naturally, all previously deployed structures have required station keeping systems, and hence, the mooring technology is well-established and used in other offshore sectors like the naval and O&G sectors. As a consequence, the wave energy sector has considered, to a great extent, this existing knowledge and experience when designing mooring systems for WECs (Harris

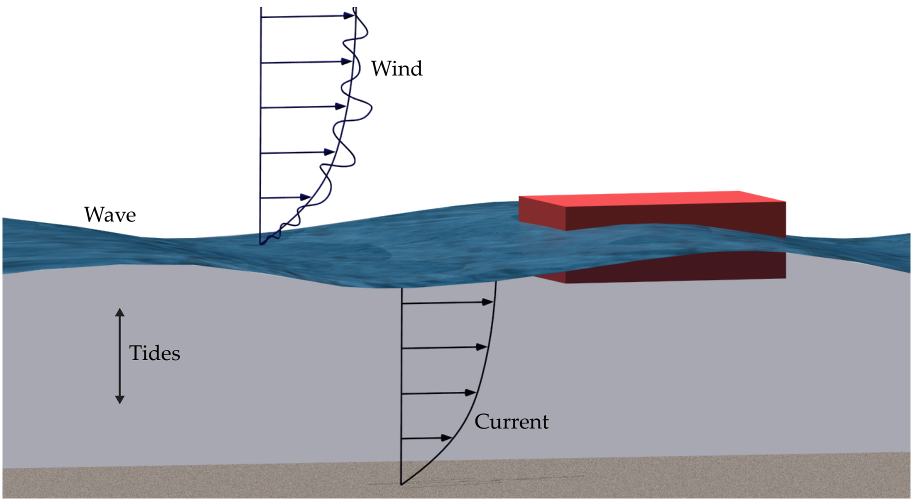


Fig. 2.1: Environmental loads to consider for floating structures.

et al., 2004). However, there are many considerable differences between these sectors, which will be highlighted in the following sections, and applying these to the wave energy sector has proved to be expensive and a driver towards mooring failure. Considering e.g. the cost breakdown in Fig. 1.10, the cost of moorings for a WEC is listed to 18% while for a floating oil platform, it is only 3% (Fitzgerald, 2009). Therefore, the incitement for optimizing mooring in the other sectors is not as significant, and with their available finances, it is more beneficial to ensure that the mooring is oversized (thereby introducing high security) than it is to ensure cost optimization.

The following sections will present the state-of-the-art of mooring design, by describing the applied systems and research performed by other authors. In the next chapter, a more specific description of the current state of mooring design will be presented.

2.1 Mooring System Definition

When exposed to loads from wind (described as a steady wind with fluctuating gusts), current (steady load) and wave, the structure will experience motions. Often, the wave loads are composed of a Wave Frequency (WF), Low Frequency (LF) and mean drift load (Bergdahl, 2017).

In order to accommodate these loads, a mooring system is applied, which by definition comprises a number of lines, connections, buoys, anchors etc., connecting the floating structure to the seabed, and hence, provides a restoring force on the structure. Depending on the system, this structure can provide stiffness in one or more of the Degrees of Freedom (DoFs) defined in Fig. 2.2.

Considering the hydrostatic properties of any floating structure, a mooring is considerably vital in particularly the horizontal DoFs in order to prevent the structure from drifting, but depending on the WEC and its requirements, the system can also be made to influence any other DoF.

As defined by several authors like e.g. Fitzgerald (2009), a well-designed mooring system is capable of reacting to all environmental forces and pre-

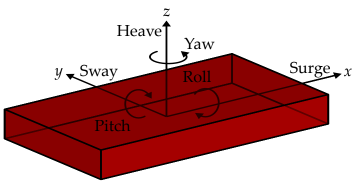


Fig. 2.2: Definition of the DoFs.

venting a large offset of the floating device, but at the same time comply with the offset without inducing large loads on the components and device. Harris et al. (2004); Martinelli et al. (2012); Davidson and Ringwood (2017) list several requirements for WEC mooring systems that can be summarized into the following:

- The mooring system must keep the WEC on station under both operational and extreme sea states. The system must ensure that all specified tolerances regarding motions are respected, e.g. in order to avoid interference with surrounding structures and to avoid undesirable tension in electrical umbilical.
- The system must ensure adequate strength and fatigue endurance in all mooring components to ensure the mooring survivability in the entire lifetime.
- The system must be compliant enough to reduce loads on anchors, in lines and on the WEC, but also be stiff enough to limit the motions into the defined tolerances.
- The mooring must be designed to avoid line-line and line-WEC contact.
- For some types of WECs, the mooring system must ensure that the device can weathervane in order to face the incoming waves.
- The mooring must not affect the PTO performance, but if it is a part of the PTO, it must be dynamically designed as part of the system.
- The mooring is preferably providing some redundancy in order to increase survivability and to allow for removal of a line for inspection and maintenance.
- The system must minimize environmental impact and, therefore, reduce the seabed footprint as much as possible.

Depending on the specific device and site, the list of requirements can naturally vary. When considering the above requirements, it must also be attempted to find the most economical solution by means of limited material use, easy installation, monitoring, maintenance and decommissioning. In later stages, it must also be expected that WEC will be installed in arrays and there will be a greater need for reduction of seabed footprint so that the devices can be installed close to each other.

There are many different types of WECs and, consequently, the requirements to the moorings are specific for each device. For wave energy application, mooring systems are characterized, by e.g. Martinelli et al. (2012), as:

- Reactive moorings when the mooring provides reactive forces and, hence, is a part of the PTO,
- Active moorings when the mooring influences the dynamic response of the WEC and its energy production and
- Passive moorings when the mooring only secures station keeping and has no influence on the PTO.

The list clearly presents moorings with a decreasing influence on the PTO, with the passive moorings being closest to what is known from the naval and O&G sectors. This type of mooring is the focus on the present thesis.

2.2 Near-Shore Deployment

In the first phase of development, most large floating WECs are planned for near-shore deployment. This brings a number of advantages related to O&M, installation etc. due to the short distance. Furthermore, the decreasing water depth results in a narrower spreading of the wave direction because of refraction, thereby putting fewer demands on the weathervaning capabilities of the devices and mooring. All combined, this tends to makes it a cheaper and more attractive solution in the early stages.

Unfortunately, near-shore deployment often implies shallow or intermediate water depths, which give rise to some problem areas when considering moorings. Due to the interaction between seabed and waves in these conditions, the waves will become non-linear and steep, and hence, provide a larger excursion of water particles, which induces a larger drift force on the structure and more outspoken impact from the waves (Fitzgerald, 2009).

Pecher et al. (2014) found that the stiffness of a mooring system increases with decreasing water depths, and thereby, provides larger loads in lines and on the structure. This might result in difficulty in designing a mooring system, since the limited water depth also reduces the possible span that can be used to accommodate these large loads. Furthermore, it is also noticeable that in shallow water depths there is water particle movement in the entire vertical span. This puts a larger load in the lines and anchors and increases the required strength.

The devices in Fig. 1.7 are all planned for deployment in intermediate conditions ($0.05 < h/L < 0.5$, where h is the water depth and L is the wavelength in extreme conditions).

2.3 Mooring Layouts

Researcher have proposed many different types of mooring layouts throughout the years, mainly coming from the O&G and naval sectors (Harris et al., 2004; API, 2005). The most common are illustrated in Fig. 2.3. Overall, moorings can be divided into being either Single Point Moorings (SPMs) or spread moorings, which provide different characteristics to the system. Additionally, the systems can be classified as either slacked or taut, where b) and d) are the former and a), c) and e) are the latter. In addition to the systems illustrated in Fig. 2.3, mooring layouts can also have intermediate elements in the lines and e.g. have a lazy-S shape as shown in Fig. 2.4. The type of system highly influences the type of material used for the mooring lines.

A major difference between the spread mooring and SPM systems is the capability of the latter to allow for weathervaning as it does not provide any restoring force in yaw DoF. For WEC application, the SPM is, therefore, often desirable. Fig. 2.3 c) and d) illustrate respectively a SALM and CALM system, which influences the WEC in a similar way as the device is connected to a buoy that is moored to the seabed. e) illustrates a taut turret system, meaning that the mooring lines are connected directly to the WEC in one connection point. Naturally, this means that the mooring provides a higher influence on the device response in both translational and rotational DoFs. The turret system can also be made of slacked lines. The applicability of different systems are treated in e.g. Harris et al. (2004); Fitzgerald and Bergdahl (2007); Sound and Sea Technologies (2009); Karimirad et al. (2015)

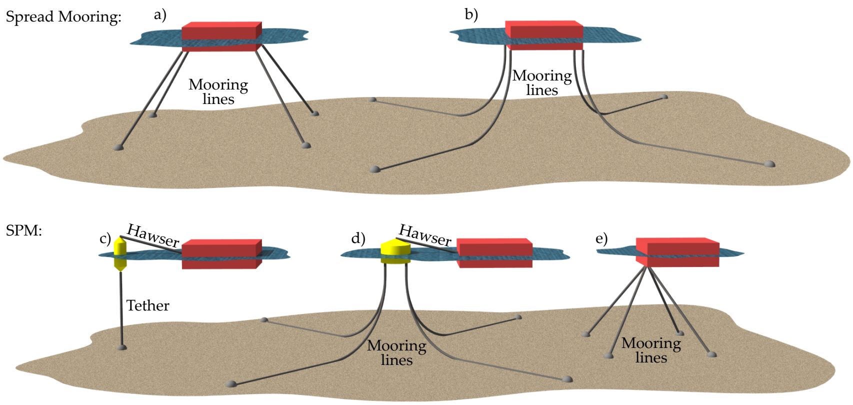


Fig. 2.3: Examples of spread moorings (top) and SPMs (bottom). a) Taut spread; b) Multi-catenary; c) SALM; d) CALM (could also be taut); and e) Taut turret (can also be with catenary lines)

2.3.1 Components

The mooring system is primarily composed of lines, anchors and connectors and can be additionally equipped with buoys and clump weights (risers and sinkers).

Fig. 2.4 illustrates a generic mooring line and lists some of the different materials that can be used for each part. Lines have traditionally been composed of chain and steel wire rope in the O&G sector and with some use of synthetic lines in terms of polyester (Ridge et al., 2010). The choice of line material affects the working principle of the lines as it influences axial and bending stiffness, inertia and damping (Harris et al., 2004; Sound and Sea Technologies, 2009; Harnois, 2014; Davidson and Ringwood, 2017).

Anchors are applied to ensure a fixed connection between mooring lines and seabed. Choice of anchors is largely dependent on the seabed condition and the mooring type. For slacked mooring systems, the anchor is primarily required to provide strength in the horizontal direction, while taut systems also require vertical strength. Anchor types include e.g. gravity based, pile, plate, suction bucket and drag embedded anchors (Sound and Sea Technologies, 2009; Karimirad et al., 2015) and more novel types such as bag anchors (Flory et al., 2016).

Buoys can either be surface piercing or submerged and can act as a connection point between lines (in SPM and CALM). Buoys can be applied in the mooring system together with clump weight to tune the stiffness of the mooring.

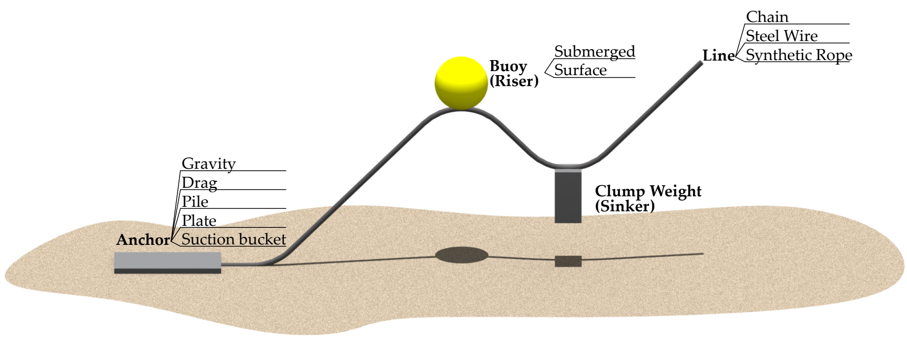


Fig. 2.4: Generic mooring line (lazy-S shape) with indication of different material that can be considering in the lines.

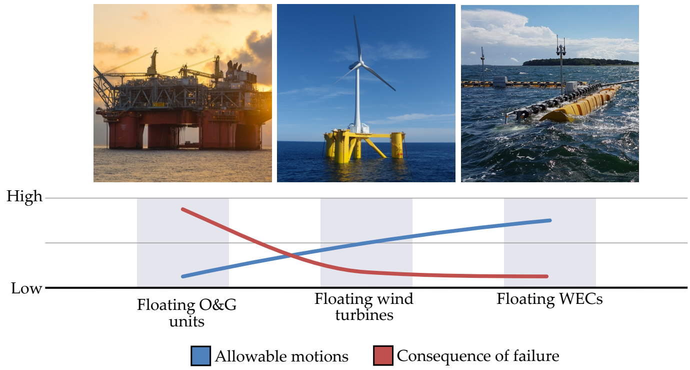


Fig. 2.5: Conceptual illustration of the variation of allowable motions and concequences of failure in offshore energy structures (BP, 2017; Fukushima OWC, 2016).

2.4 Mooring Regulations

On a worldwide basis, numerous certification companies are providing guidelines and regulations on design and construction of offshore structures. These are compiled into a variety of different design standards (cf. Table 2.1) with requirements on applied safety factors, environmental conditions and analysis methodology. In order to obtain a certified structure by a certification company, e.g. crucial for insurance purposes, it is required that these standards are followed whereby a satisfying level of safety and reliability is achieved (Harnois, 2014).

For floating structures, it is necessary to follow design standards for mooring design. Since the wave energy sector is still relatively immature without any long-term deployments, the development and knowledge required to make matured standards are not yet available. Consequently, there has been a tendency to adopt the existing standards from the O&G and naval sectors and apply the traditional layouts and materials. Table 2.1 lists a number of selected and relevant standards for overall mooring design including standards for chain, ropes and anchors.

Applying O&G design standards on WECs is not necessarily the optimal choice because of the considerable difference between the structures and requirements. In most cases, the O&G structures are large by size compared to wave energy structures and, hence, experience a much lower response and corresponding velocities. This is desirable for this type of structure as it is often manned and has very strict limitation on the allowable motions in all DoFs, cf. Fig. 2.5. These limitations are a natural consequence of the risers and their incapability to handle any large displacement.

Standard no.	Title
Offshore Structures (Naval and O&G):	
DNVGL-OS-E301 (DNV-GL, 2015d)	Position Mooring
API-RP-2SK (API, 2005)	Design and Analysis of Stationkeeping Systems for Floating Structures
ISO 19901-7:2013 (ISO, 2013)	Stationkeeping systems for floating offshore structures and mobile offshore units
Floating Wind Turbines:	
DNV-OS-J103 (DNV, 2013)	Design of Floating Wind Turbine Structures
Marine Energy Converters	
IEC TS 62600-10:2015 (IEC, 2015)	Assessment of Mooring System for Marine Energy Converters (MECs)
Mooring Chain:	
DNVGL-OS-E302 (DNV-GL, 2015b)	Offshore Mooring Chain
API SPEC 2F (API, 1997)	Specification for Mooring Chain
ISO 20438:2017 (ISO, 2017)	Offshore mooring chains
Steel Wire Rope:	
DNVGL-OS-E304 (DNV-GL, 2015c)	Offshore Mooring Steel Wire Ropes
Offshore Fibre Ropes:	
DNVGL-OS-E303 (DNV-GL, 2015a)	Offshore Mooring Fibre Ropes
API-RP-2SM (API, 2014)	Design, Manufacturing,and Maintenance of Synthetic Fibre Ropes for Offshore Mooring
ISO 9554:2010 (ISO, 2010)	Fibre ropes - General specifications
Anchors:	
DNV-RP-E301 (DNV-GL, 2017)	Design and installation of Fluke Anchors

Table 2.1: List of relevant standards for mooring design.

The limitations on motions of WECs are much different, as illustrated in Fig. 2.5. This is naturally a consequence of the desire to maximize the interaction between waves and device for optimal wave energy absorption. Contrary to O&G platforms, significant parts of the WECs are desired to be in resonance with the waves, but large floating WECs are not necessarily resonating. Even if resonance is not desired, the structure can often be allowed to experience significantly larger excursions in order to decrease loads. This is a clear benefit as WECs, naturally, are deployed in high-energy areas.

There are some clear differences between the two sectors, which cause challenges in applying the known layouts and design procedures. One sector, which is closer to the wave energy sector than the O&G, is the floating wind sector, which has gained much focus in the last decades and experienced significant improvement. As presented in Fig. 2.5, the allowable motions for this type of structure are larger as there are no limitations from risers. An umbilical must be applied for which reason the excursions must be small enough in extreme seas to limit or prevent tensions in the cable; this also stands for WECs. The motions of a Floating Offshore Wind Turbine (FOWT) is nevertheless more limited than a WEC in operational conditions because of the wind turbine. In order to ensure power production, the acceleration of the hub must be limited to $0.5g$ and the pitch motions typically to $\pm$ 10°. The limitation on motion is a relatively soft criterion and can be made as a compromise between energy production and cost of platform. Large pitch causes less energy production, but is so far often a result of a smaller and cheaper foundation.

Since the floating wind sector is more mature than the wave energy sector and more relatable than the O&G, it is valuable to consider design standards like DNV-OS-J103 for FOWT design in the design of WECs and their moorings.

In the wave energy sector, reports such as EMEC (2009) and DNV (2005) provide guidelines and recommendations on which of the existing design standards and methodologies that should be applied. These guidelines formed the baseline for the IEC TS 62600-10 standard which is aimed at Marine Energy Converters (MECs), but still largely based on the existing standards. It is stated in the design standard that as the wave energy sector matures and as more data and knowledge become available, it should be updated and particularly the safety levels should be addressed.

It is obvious that the safety level and consequence are different for the three sectors. As depicted in Fig. 2.5, the consequence of failure of a O&G platform is massive compared to a FOWT or WEC. The environmental damage will be significant, but the most critical consequence of failure is the loss of human life, as many O&G platforms are manned. Still, the wave energy sector has adopted the same safety factors, even though the devices are unmanned and smaller, thereby resulting in much less damage. Consequently, authors like e.g. Paredes et al. (2013); Johanning et al. (2005) suggest a relaxation of the safety levels. This requires a comprehensive assessment of the reliability and a calibration of safety factors as treated in Ambühl et al. (2014a,b, 2015a,b). However, it is relevant to acknowledge that many WECs have failed and caused financial and political damage as well as induced a negative public opinion on wave energy. In the early stages of development and until more experience is gained, it is, therefore, debatable if not safety levels should be kept high. An opinion which is shared in e.g. Harnois (2014).

2.5 Designing Mooring Systems

Several authors like Martinelli et al. (2012), Bergdahl (2017) and Johanning et al. (2005) describe the necessary design process for a final mooring system. This is naturally also treated in API (2005) and IEC (2015), and a general design flow can be seen in Fig. 2.6. The initial step in the design process is to characterize the deployment site according to all relevant environmental conditions and site-specific restraints. Also, the characteristics of the WEC must be defined. This step is followed by a first mooring system layout, where all components and layouts are defined. From this, it is possible to assess the performance of that specific system, based on different methodologies ranging from experimental tests to numerical modelling. The latter can either be quasi-static or dynamic, where the quasi-static approach has been applied in some O&G applications due to the slow structure response. The same approach has been adapted by several large floating WECs, as will be presented in the next chapter. The applicability of the different analysis methods is a significant uncertainty due to the differences between the sectors. Consequently, this topic will be treated in details later in the thesis. According to the design standards, the response can be assessed by simulating a number of extreme sea states with a duration of 3 hours. The extremes tensions and motions are then extracted and design values found.

Until this step, the design process is independent of the chosen design standard. The difference occurs in the verification phase, where different

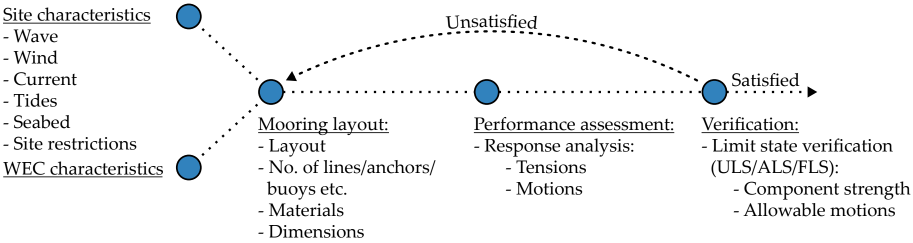


Fig. 2.6: Flow-chart of design process. Inspired by Martinelli et al. (2012); API (2005); IEC (2015)

standards have specific requirements, safety factors, level of safety etc. Generally, it is necessary to verify the system in different limit states: Ultimate Limit State (ULS), Accidental Limit State (ALS) and Fatigue Limit State (FLS), and it must be proved that all components provide sufficient strength and that the motions are within the specified margin. If not, a new layout must be tested, indicating an iterative design process.

All limit states must be considered, but according to Zanuttigh et al. (2012), the ULS often determines the overall cost of the system. For WECs, which experience many cycles of higher loads, the FLS is also expected to be a significant driver of the cost. In order to secure survivability and also find cost-efficient systems, this thesis focuses only on the ULS and considers only aligned environmental loads.

3 Mooring Design in the Wave Energy Sector

The previous chapter showed that the O&G and naval sectors have much experience on mooring design when considering design procedures, layouts and materials. However, there are some significant differences between these and the wave energy sector. Chapter 1 illustrated that many of the deployed WECs have failed due to failure of mooring, and hence, it is critical to investigate and understand the reasons behind this in order to allow for improvement and optimization before new devices are deployed. This chapter will investigate the state of mooring design in the wave energy sector and highlight problem areas and feasible solutions.

3.1 Initial Mooring Design for Large WECs

When considering the four large WECs presented in Thomsen et al. (2015) (Paper [A]) and Fig. 1.7, it is possible to achieve an understanding of the state of the mooring design for such types of WECs. These devices are planned for deployment at the sites presented in Fig. 3.1, covering the Danish test facility DanWEC (1), the Danish part of the North Sea (2) and the Belgian cost (3). These deployments are all in intermediate water depths, which mean that the developers will deal with similar problems and expectedly could use similar considerations on layouts and design approach. In spite of this, different mooring solutions have by now been designed for each of the WECs covering the solutions listed below and illustrated in Fig. 3.2.

- Catenary turret system composed of chains and drag embedded anchors.
- SALM system composed of a deformable steel tether and synthetic hawser.

- SPM system with composite lines of nylon and chain, and with drag embedded anchors.
- Turret system composed of nylon lines.

While some of these systems are based on traditional systems from O&G and consists of chains, others are more novel like the SALM systems. SALM systems are known in the offshore sectors (API, 2005), but are traditionally composed of a surface piercing buoy connected to the seabed by a chain tether and used for non-permanent moorings. The described system consists of two submerged buoys connected to the seabed by three steel rods joined by universal joints, allowing the tether to deform. The WEC is connected to the tether by a nylon hawser. The use of nylon is seen in two of the other systems as well and is considered a more novel solution, since synthetic lines more traditionally have been composed of polyester.

Fig. 3.1: Map of the deployment site for the four WECs illustrated in Fig. 1.7. Two of the devices are planned for deployment at Site 1. Extreme environmental conditions are listed for each site.

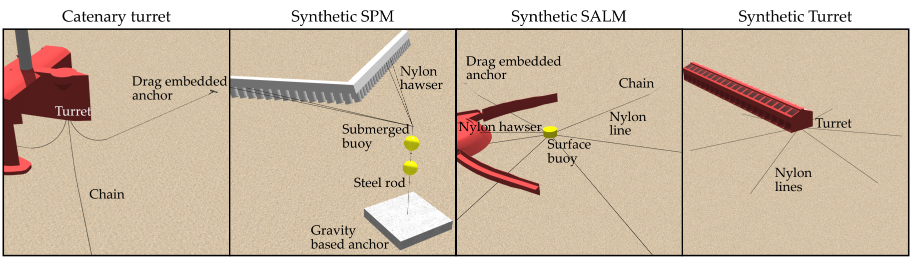


Fig. 3.2: Initial mooring solutions designed for the WECs in Fig. 1.7

	Restoring Force	Inertia	Seabed footprint	Snap Loads
Catenary turret	Geometric changes Weight of chains	Heavy lines	Large	Slacking
Synthetic SPM	Elasticity of lines Buoyancy of buoy	Light lines	Less	None
Synthetic SALM	Elasticity of lines Buoyancy of buoy	Light lines	Limited	None
Synthetic turret	Elasticity of lines	Light lines	Less	None

Table 3.1: Relevant effects in the four different mooring solutions.

Naturally, the layouts and materials highly influence the effects from the mooring system and provide different characteristics of them. Some relevant parameters are listed in Table 3.1 (Thomsen et al., 2015; Davidson and Ringwood, 2017) (Paper [A]).

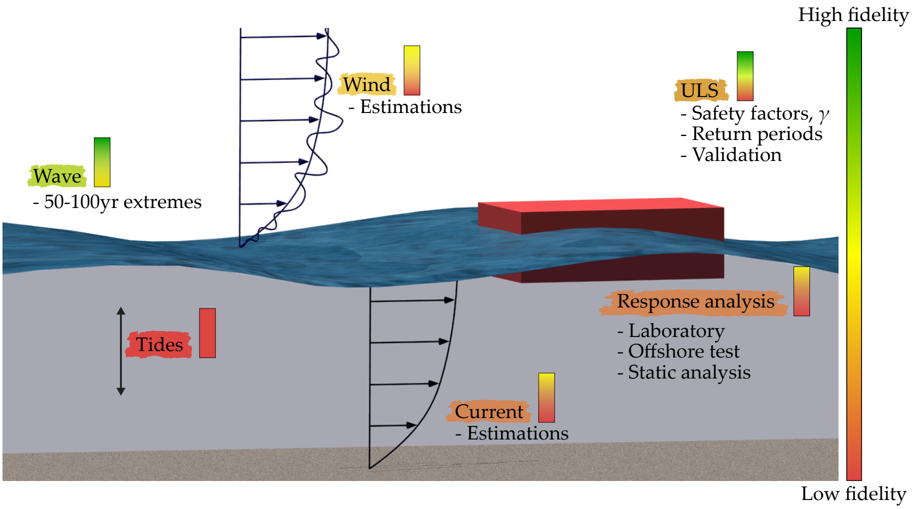


Fig. 3.3: Illustration of the parameters which should be considered in a mooring design procedure. The colors indicates the level of fidelity, hence to what extent each parameter has been considered in the design process. The vertical color bars at each parameter illustrates the variation in the level of fidelity between the four investigated WECs, while the color at the text resembles the average. The text at each parameter explains how it has been included.

3.1.1 Initially Applied Design Procedures

Previous sections described that several deployed moorings have failed due to a range of different parameters. In order to clarify these, Thomsen et al. (2015, 2017f) (Paper [A] & [F]) evaluated the mooring solutions and design procedure initially applied to the WECs in Fig. 1.7. As stated, the design procedure differs significantly for each of the systems and, thereby, identifies a clear lack of standardized approach. Fig. 3.3 summarizes the parameters which should be considered in the design according to the design standards in Table 2.1, and illustrates to what extent these conditions have been considered.

It is paramount to notice from Fig. 3.3 that none of the parameters have been fully described. In general, the wave conditions are the most well-described of the environmental conditions, while even for this there are some differences between the WECs, particularly when observing the applied design return period of the extreme cases. All the investigated structures have used extreme data in the design, but the return period of these cases varies from 50-100 years.

There is a tendency of neglecting both wind and current loads at this stage of the design, while it is merely based on estimations in the cases where they are considered. Only one of the cases has considered currents, while three have established estimations on the wind loads. These load estimations are in all cases treated as steady loads and, hence, not the time-varying loads that are defined in the standards mentioned in Chapter 2. None of the investigated cases have considered the tides and variation of Still Water Level (SWL), which is of paramount importance in shallow waters.

A crucial parameter in the design process is the assessment of the response of the WEC when exposed to environmental conditions. As it will be presented in Chapter 4, the methodology significantly influences the accuracy of the obtained design tensions and motions. Because of the present development stage of all the devices, all have been undergoing laboratory tests, some also smaller scale offshore tests, and thereby obtaining an understanding of the wave loads on the structure. In these cases, the only considered environmental loads arise from the waves. For the offshore tests, wind and currents have been considered, but without any thorough description of them. In many cases, the geometry was also modified after these tests, with the loads from the original layout assumed applicable to the new layout with estimations of a scale factor. In almost all cases, the considered mooring solutions were validated by use of a static procedure, without inclusion of any dynamic effects.

When considering all of the above statements, it is obvious that not much focus has been put in ensuring design according to the specifications in any of the design standards. Since primarily waves have been considered and with varying return period, the specifications for analysis in ULS are not satisfied. Only a single case has applied safety factors, γ, on tensions and breaking strength. As presented in Chapter 2, floating WECs need to ensure that there is no tension in the umbilical and, hence, it is necessary to define excursion limits or include it in the mooring design. Similarly, it is necessary to account for specifications on the deployment site regarding surrounding structures, allowable footprint radius etc. None of this has been considered in the current state of the mooring design.

The differences and shortcomings in the designs process for the considered cases can be summarized into:

- Inconsistency in the considered return periods.
- Insufficient consideration of environmental loads.
- Varying methodologies applied for the response calculations.
- Lacking of validation according to any design standard.
- No design limits on motions have been defined.

It is clear that the presented mooring solutions in most cases are at the very first steps in the design phase, and not yet considered final solutions. There is no clear tendency in the applied design procedures and despite having proposed a solution, the safety and survivability of the structures cannot yet be guaranteed.

3.2 Discussion of Potential Mooring Solutions

Mooring solutions in the wave energy sector appears to vary greatly in the use of material type and layouts, and it is likely that some solutions are more applicable to the given type of device and deployment sites.

Each type of solution has its advantages and drawbacks, affecting both the cost and performance of the system (Thomsen et al., 2016c) (Paper [B]). A turret solution, which was considered in two of the investigated cases, has the advantage that it can be produced as a disconnectable system, where the mooring is installed as a separate system which can be winched up and locked into the bearing part of the turret. This highly eases the installation process and maintenance operations on the WEC but often entails a more expensive turret cost. Similar considerations can be made for the SPM and SALM systems regarding O&M and installation due to their disconnectability.

As presented in Table 3.1, another significant difference between the mooring layouts is the seabed footprint, which might be determining for the choice of system as some deployment sites will have restrictions on this matter. In the long term, it will also be necessary to limit the radius in order to install the WECs in arrays. The catenary system has the largest footprint as it is designed to avoid vertical loads on anchors and, therefore, has a large part of the chain lying on the seabed meaning that drag embedded anchors can be used. A disadvantage is that the lying part of the chain can get buried in mud over time, thereby changing the response of the system significantly. The systems with synthetic lines have a smaller footprint radius, but all require vertical strength and, therefore, need either special types of drag anchors or alternatively gravity base, pile or suction anchors.

The most dominating parameter on the performance of the structure is the line material. As presented in Table 3.1, chains are generally heavy and introduce difficulties for systems in shallow water. As presented in Pecher et al. (2014), the mooring stiffness will increase with decreasing water depth, introducing large loads on the lines. Thomsen et al. (2016c) (Paper [B]) consider three mooring solutions similar to the catenary turret, the synthetic SALM and the synthetic turret and design them for the same structure and environmental conditions. From these cases, it becomes clear that chains are difficult to apply to such structures. The high stiffness results in large line loads, leading to a need for strong and even heavier lines. In addition, very long lines are needed in order to introduce compliance into the system, which results in unrealistic footprint radius (more than one kilometre) and cost.

For the three cases, Thomsen et al. (2016c) (Paper [B]) showed that synthetic lines could be used. This type of line can consist of many different materials but have, so far, largely been based on polyester, which are also used in O&G applications (Banfield and Casey, 1998; Banfield et al., 2005; Flory et al., 2007; Banfield and Flory, 2010). Since the purpose of using these lines is to increase compliance, nylon has a significant advantage over polyester, as it is generally twice as compliant. Studies like Ridge et al. (2010) and Thomsen et al. (2016b) both stated that by doubling the compliance, the loads could be halved. However, nylon lines are more novel for mooring applications and have not yet any long-term deployments to provide experience.

Despite changing characteristics over time due to creep, synthetic ropes generally provide better tension fatigue than chains due to no corrosion and are now considered feasible for 20-30 years of applications (Ridge et al., 2010). This requires use of parallel type ropes and application of protective jackets in order to prevent soil and marine growth ingress (Banfield et al., 1999; Weller et al., 2015). Often, the vulnerability to physical damage and weak points at connections has been mentioned as disadvantages of synthetic lines.

However, polyester lines have been used in the O&G and naval sectors for years with an insignificant number of failures compared to steel lines (Weller et al., 2015).

In the study by Thomsen et al. (2016c) (Paper [B]), the use of synthetic lines instead of chains resulted in a 94% decrease of line material cost. This is an enormous reduction and is explained by the fact that the chain system, with the very long lines, simply is unrealistic to ever build in these conditions. Considering a SALM system, the line material cost was decreased with 98%, showing an even greater potential.

From this study, it is obvious that mooring solutions with chains are not feasible solutions for WECs in shallow and intermediate water depths. Synthetic nylon lines and SALM systems both appear to decrease the mooring loads significantly and thereby increase durability and potentially decrease cost. These types of systems will be the focus of the remaining of this thesis. It is important to notice that the response analysis in Thomsen et al. (2016c) (Paper [B]) is based on the same quasi-static analysis method as generally used in the wave energy sector. Because of the very dynamic behaviour of WECs, this is not necessarily the most reliable method. The following chapter treats this matter and presents a methodology to analyse moorings for WECs.

4 | Analysis of Mooring Systems

As presented in the previous chapters, several mooring systems in the wave energy sector have been designed using either laboratory tests or relatively simple quasi-static analyses. However, as stated in Chapter 2, extreme seas must be considered for the sake of survivability and under these conditions dynamic effects will be present as shown later in this chapter. It is critical to ensure that the applied models for mooring analysis provide results with a satisfying or known level of accuracy. This chapter treats some of the available methods for mooring analysis and discusses their validity, applicability, uncertainties and reliability.

Generally, two methods can be used for the assessment of mooring and WEC response: Physical laboratory tests and numerical analysis. While physical tests provide highly reliable results, they are often time-consuming, expensive and become complex when all environmental conditions are considered. In these cases, it is often more convenient and less expensive to use numerical modelling, due to beneficial automation, easy changing of parameters and full control of the tested conditions. However, in most cases one method is not used without some application of the other method as numerical modelling is more efficient, but the model needs verification and validation for which physical tests are needed. The concept of

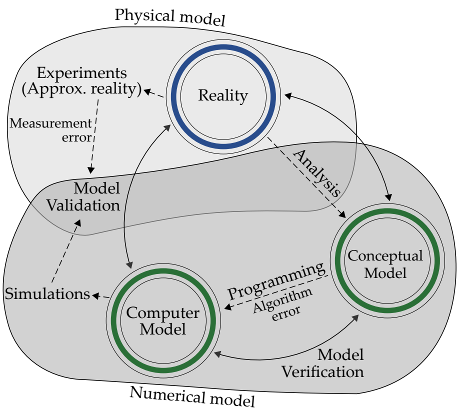


Fig. 4.1: Diagram of the concept of verification and validation. Based on Oberkampf and Trucano (2002); ASME (2009); Ferri (2014)

verification and validation is depicted in Fig. 4.1 and treated in ASME (2009); Oberkampf and Trucano (2002). Often the problem takes its basis in some reality-based problem, which can be described by different conceptual models. These models are programmed into a numerical computer model that has to undergo a verification phase, in which the conceptual models are assessed and any algorithm error characterized. In this step, the ability to simulate reality is not considered, but it is merely ensured that the algorithms function properly according to the mathematical description. The computer model is subsequently used to provide a series of simulation results. Similar, physical tests can be used to provide a database of experimental results (resembling an approximated reality), which can be compared to the simulation data. This step forms the validation phase, in which the ability of the computer models to model reality is assessed. Naturally, it is important to evaluate the measurement errors as these play a significant part in the validation. In most applications, it is not realistic to expect a complete agreement between the model and reality, since it is desirable to make simplifications to reduce simulation time etc. Therefore, the validation phase is not aimed at answering whether the model forms an exact representation of reality, but more an estimation of the range of errors and an evaluation on whether the model can be used for the specified application, being either the final or initial design.

In many early stage development of WECs, there is no experimental data available, and the devices tend to undergo a large amount of geometrical and PTO changes. Still, there is a need to understand the mooring behaviour in these initial phases and, therefore, a need for a model, which can provide reliable results in the early stages where no tests are available. This situation forms the focus in the current thesis, meaning that the numerical models need validation according to the initial design situation.

The following sections treat each of the steps in Fig. 4.1 more detailed, as they are crucial in the mooring design, starting with physical testing and followed by numerical modelling.

4.1 Physical Testing

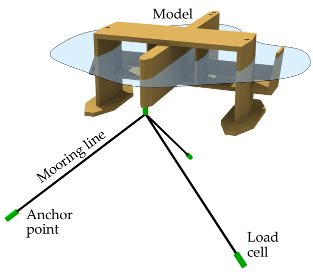
Physical testing plays an important role in any design of a mooring system. Even in cases where numerical modelling is the main tool in the design, it is necessary to perform some tests for the validation, especially for the novel structure layouts of WECs compared to O&G structures. This is further observed since it is stated in Thomsen et al. (2017c) (Paper [D]) that a final design needs validation against experimental data before the mooring can be certified by one of the certification companies.

An apparent advantage of physical tests is the high reliability of the results and a realistic realisation of the response, of course with a risk of introducing errors, cf. Fig. 4.1. In early stages of WEC development, the tests will be performed in small scale where scaling errors are unavoidable. Measurement errors can occur independently of the scale, but the inaccuracy will be more influential in smaller scale. Here, it is necessary to assess and discuss the influence from measurement and scaling errors (Thomsen et al., 2017b) (Paper [E]). In laboratories, it is difficult to include realisations of all environmental conditions like wind and current, for which reason tests are often used just for validation and not as a design tool.

In many applications, small-scale laboratory tests are used to obtain an understanding of the response of the WEC and mooring system in the relevant sea states. As mentioned in Chapter 1, large floating WECs are not required to be in resonance with the operational wave frequencies for energy production, and it is undesirable to have resonance during extreme events. Physical tests can be used to test the relevant operational and extreme conditions in order to evaluate the response and apply frequency domain analyses for determination of resonance frequencies, f_n. This is important as the mooring systems provide stiffness in the body DoFs and hence, influence the natural frequencies.

Thomsen et al. (2016b) (Paper [C]) tested a large floating WEC, illustrated in Fig. 4.2, in operational and extreme conditions, in order to evaluate motion and tension response. The objective of the tests was to evaluate the influence from a synthetic mooring system with non-linear axial line stiffness, k_{moor}, and ensure that it did not affect the response in a negative way. It was further tested how mass moment of inertia (MoI), I_{ii}, affected the response and mooring line tensions.

Decay tests were used to estimate natural frequencies of the structure for the three different configurations with an example of a decay time series plotted in Fig. 4.3 and the frequencies presented in Table 4.1. All results in



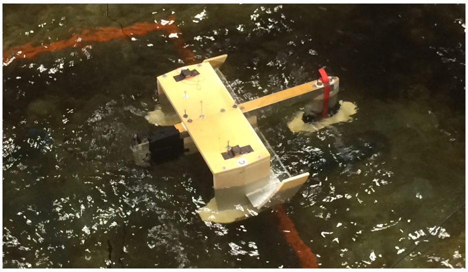


Fig. 4.2: Left: Illustration of the experimental set-up. Right: Picture from the test.

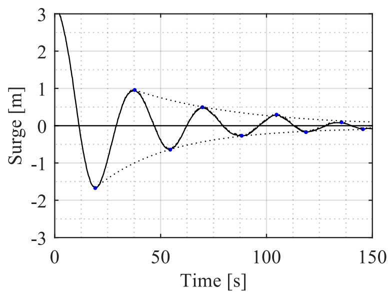


Fig. 4.3: Example of a surge decay tests in full-scale values.

Model conf.	f_n (Hz) Surge	Heave	Pitch
Basis case	0.029	0.048	0.047
$I_{yy} = +20\%$	0.029	0.048	0.049
$k_{moor} = -60\%$	0.022	0.048	0.046

Table 4.1: Measured natural frequencies, f_n for surge, heave and pitch.

this chapter are presented in full-scale values. When considering also Fig. 4.4 where the frequency band of the relevant sea states is defined together with the measured Response Amplitude Operators (RAOs), it can be seen that the dominating surge, heave and pitch natural frequencies from Table 4.1 are all outside of the wave band. When considering Fig. 4.4, a peak can be observed at 0.078 Hz in all DoF because of a coupled frequency due to the complex structure. This value is inside the wave energy frequency band and highly influences the response. A desire to decrease the frequencies as much as possible is present, and the benefit of compliant mooring lines is observable.

The mooring tension is measured during each of the investigated sea states. These can be used directly in the design of the moorings, but it requires that the full-scale lines have similar properties. Also, no current and wind are included, putting a limitation to the use of the measured tensions as design values.

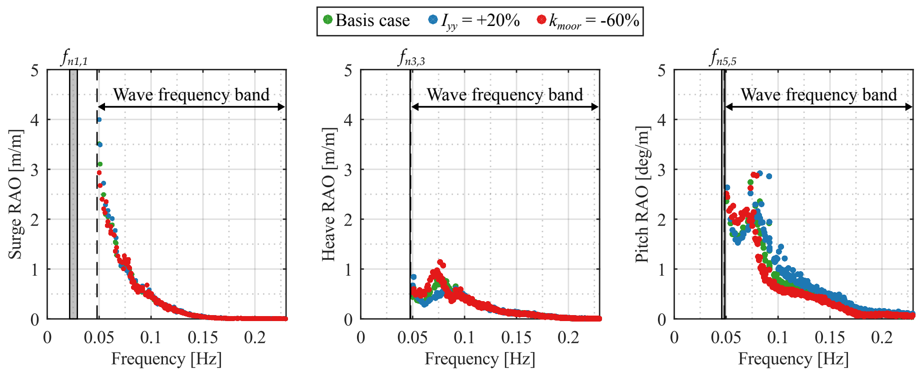


Fig. 4.4: Experimentally determined RAOs and the investigated wave frequency band. The bands of natural frequencies from the three configurations (cf. Tab. 4.1) are also presented.

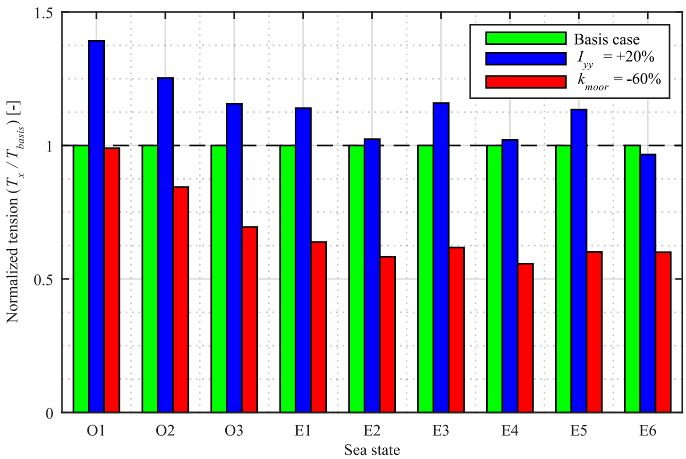


Fig. 4.5: Comparison of experimentally measured tensions from three different system configurations. 'O' indicates operational conditions, while 'E' indicates extreme.

Fig. 4.5 illustrates the normalized tensions obtained when increasing the MoI and decreasing the mooring line stiffness. There is an obvious influence on the mooring line tension from the MoI, but a dominant effect is observed from the stiffness of the mooring system. In the most extreme sea states, it is shown how the mooring loads are decreased with up to 40%, when decreasing the stiffness by 60%. This provides a strong argument for using synthetic lines, particularly the very compliant nylon.

The testing in Thomsen et al. (2016b) (Paper [C]) provides a general understanding of the mooring systems and its effect on motions of the WEC and tensions in the mooring lines. Clearly, such an analysis is insufficient for a final design, since current and wind loads were not considered. In order to use the physical tests for final design, a very accurate set-up is needed which both ensures a sophisticated and accurate realisation of the full-scale lines and includes all environmental loads.

4.2 Numerical Analysis

As indicated in the previous section, physical tests provide a reliable estimation of the mooring response, but becomes complex when all environmental conditions are included. Mooring design has an iterative nature, where many different configurations, line types and sizes must be tested before a final and valid solution is found. On this matter, numerical models are very efficient.

When modelling a moored WEC as illustrated in Fig. 4.6, the model can be divided into two overall components: a module that treats the wave-

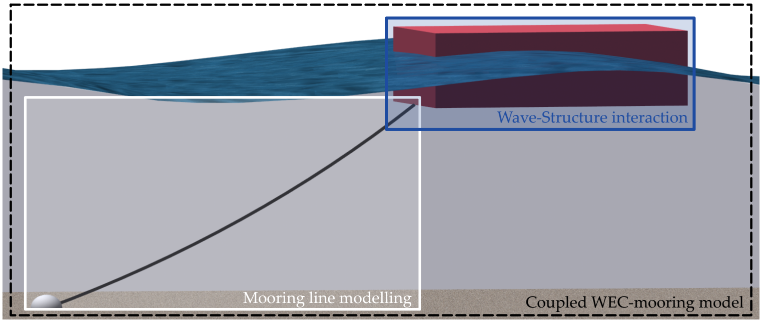


Fig. 4.6: Illustration of numerical modelling of a moored WEC.

structure interaction and a module that treats the behaviour of the mooring line including motions and tensions. The full system can be treated as either one of the following:

- A de-coupled system where the wave-structure interaction and mooring modelling are treated separately. Motions of the WEC are solved with an applied mooring stiffness, the fairlead positions are imported afterwards in the mooring solver and tensions are calculated.
- A coupled system where a global analysis is performed and all interactions between mooring system and WEC are solved directly and simultaneously.

The de-coupled approach has been considered in many early studies due to its simplicity, but was shown in Ormberg and Larsen (1998) to provide highly inaccurate motion and tension results. Consequently, most design standards require today that a coupled analysis is applied. As presented later in this chapter, the quasi-static approach follows a de-coupled approach, while most available dynamic solvers consider the fully coupled system.

In order to assess the response of the mooring, it is necessary to evaluate first the wave-structure interaction. This topic is treated in the following section.

4.2.1 Wave-Structure Interaction

Many different types of models can be used to estimate the wave-structure interaction, where some of the most well-known models are listed below:

- Morison's Equation
- Boundary Element Method (BEM)

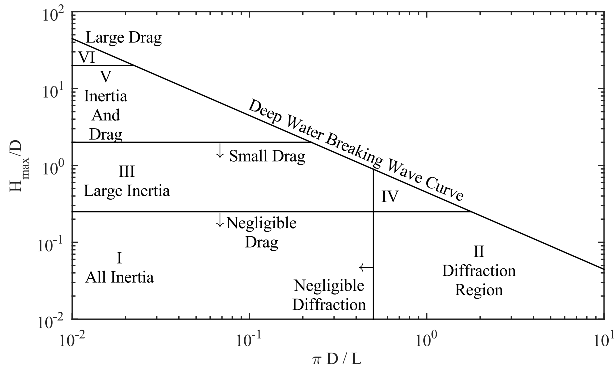


Fig. 4.7: Diagram of load regime. Based on Chakrabarti (2005) and Faltinsen (1993).

- Smoothed Particle Hydrodynamics (SPH)
- Computational Fluid Dynamics (CFD)
  - Reynold-averaged Navier-Stokes Equation (RANSE)
  - Large Eddy Simulation (LES)
  - Direct Numerical Simulation (DNS)

The level of sophistication and thereby computational time varies significantly between the models. Despite being relatively computational demanding, CFD models are becoming more and more used in analysis of WECs and other floating structures (Palm, 2017). However, for mooring design, where the design standards require long simulations (3 hours), the computational effort simply becomes too demanding. As a result, almost all practical design of mooring is based on Morison's Equation and BEM.

The two methods differ in the calculated wave effects and thereby their application areas. Chakrabarti (2005) and Faltinsen (1993) presented the diagram in Fig. 4.7, which indicated the dominating load regimes on offshore structures. As seen, when the characteristic dimension, D, of the body is large compared to the incoming wavelength, diffraction plays the most important parameter. For smaller bodies, the drag is more dominating. The methods used to calculate these loads will be presented in the following.

4.2.1.1 Morison's Equation

Morison's Equation is used for slender bodies ($D << L$), where the characteristic dimension is small and the passing wave is unaffected by the presence of the body. In such a case, primarily drag and inertia forces are imposed on the body. In Eq. (4.1), the drag force f_D and the inertia force f_I is defined, where the latter is a sum of a Froude-Krylov, f_{FK}, and hydrodynamic mass, f_M, force (Morison et al., 1950).

$$f(t) = \underbrace{\overbrace{\rho V_b \ddot{v}(t)}^{f_{FK}} + \overbrace{\rho C_a V_b \left(\ddot{v}(t) - \ddot{u}(t)\right)}^{f_M}}_{f_I} + \underbrace{\tfrac{1}{2}\rho C_D A_b \left(\dot{v}(t) - \dot{u}(t)\right) \left|\dot{v}(t) - \dot{u}(t)\right|}_{f_D} \tag{4.1}$$

Here, ρ is the fluid density, V_b is the body volume, A_b is the body area and v and u are respectively the body and fluid displacement. C_a and C_D are respectively the added mass and drag coefficients.

Even though large floating WECs are considered in this thesis and Morison's Equation is only valid for slender bodies, there is a need for inclusion of the inertia and drag forces on large bodies in extreme conditions. Here, the wavelength is considerable and, cf. Fig. 4.7, the loads on the structure will be affected by these two contributions. As the structure is relatively large, diffraction loads must still be considered and, consequently, a hybrid model should be applied, which combines the Morison drag force contribution with the radiation-diffraction loads calculated in the BEM.

4.2.1.2 Boundary Element Method

For bodies with larger characteristic dimensions, the passing wave field is disturbed by the presence of the body from both scattering and radiation from the body motions. In the BEM, the load on the structure from this wave field is solved using linear potential theory as described in Newman (1977); Faltinsen (1993); Falnes (2002); Lee and Newman (2005).

The model is invoking an assumption of an ideal fluid, meaning an inviscid, incompressible and irrotational fluid. This indicates that the velocity can be represented by a velocity potential, Φ, which satisfies the Laplace equation, Eq. (4.2), in the entire fluid domain. Due to the assumption of inviscid fluid, the drag forces are not calculated in this formulation, but can be included in the mentioned hybrid model.

$$\nabla^2 \Phi = 0 \tag{4.2}$$

Here $\nabla^2 = \left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}\right)$.

In order to solve the equation, a number of boundary conditions are defined, including kinematic conditions at the seabed and water surface. These dictate that there are no vertical flow through the seabed and the water particles remain at the water surface. At the surface, an additional dynamic boundary condition can be established ensuring that the water pressure is similar to the atmospheric pressure. Both boundaries at the water surface are non-linear and, therefore, the BEM assumes linear, deep-water waves with small steepness ($H/h << 1$). Additionally, the body motions are assumed to be of a small amplitude, and because of these simplifications, the boundary conditions can be linearized, thereby neglecting any 2$^{\text{nd}}$ and higher order terms, and a solution found.

Since the domain and solutions are considered to be linear, the total potential of the moving WEC can be solved as the sum of the velocity potential from the diffracted (Φ_d) and radiated (Φ_r) potentials.

$$\Phi = \Phi_d + \Phi_r = \Phi_i + \Phi_s + \Phi_r \tag{4.3}$$

Here, the diffracted potential, Φ_d is the sum of the potential, Φ_i, which describes the incident, undisturbed wave field, and Φ_s, which resembles the scattered wave field due to the presence of the fixed body. Finally, the radiation potential, Φ_r, describes the disturbance in the fluid domain due to the motions of the body.

When solving the forces on the structure, the 3D problem is mapped into a 2D problem by approximating the body geometry by panel surfaces (Lee and Newman, 2005). Naturally, the accuracy of the obtained results becomes dependent on the discretisation of the surface. In order to quantify and limit this error, it is important to perform studies that test the sensitivity of the results on changes in discretisation. As the computational requirements are determined by the number of panels, it becomes beneficial to find a balance between satisfying accuracy and computational time.

As indicated in Eq. (4.3), the radiation and diffraction components can be solved separately and in the frequency domain due to the linearization of the problem. The diffraction component (or excitation component, as is often denoted) is the sum of the forces exerted by the passing waves on the fixed body. As shown in Eq. (4.3) this component is found by considering the potential from the undisturbed and diffracted wave field separately. The force contribution from the undisturbed waves is denoted the Froude-Krylov force, $\overline{F}_{FK}$, and is only valid for small-body problems where the wave field is unaffected, while the diffraction force, $\overline{F}_d$, corrects the Froude-Krylov force and accounts for the presence of a larger body. By use of Bernoulli's equation, the water pressure from each contribution can be found and integrated over the wetted surface to find the resulting forces. The frequency dependent wave excitation force, $\overline{F}_{exc}$, becomes:

$$\overline{F}_{exc}(\omega) = \overline{F}_{FK}(\omega) + \overline{F}_{d}(\omega) \tag{4.4}$$

The radiation force, $\overline{F}_r$, arises from the motions of the body in still water, hence $\overline{F}_r$ represents loads exerted on the body from the motion of the body itself. The frequency dependent force is calculated from Eq. (4.5), and is separated into two terms: one proportional to the body acceleration (added mass) and one to the velocity (radiation damping) (Lee and Newman, 2005).

$$\overline{F}_r(\omega) = \boldsymbol{C_A}(\omega)\ddot{\overline{v}}(\omega) + \boldsymbol{C_B}(\omega)\dot{\overline{v}}(\omega) \tag{4.5}$$

Where $\boldsymbol{C_A}$ and $\boldsymbol{C_B}$ are respectively the added mass and radiation damping coefficient matrices, while v is the body displacement and ω is the angular frequency.

There are different numerical codes for solving the above hydrodynamic coefficients, e.g. the commercial software package WAMIT (Lee and Newman, 2006) and the open-source code NEMOH (Babarit and Delhommeau, 2015). The latter has been used for all work presented in this thesis.

The loads presented so far are all first order loads, but as mentioned, the second order drift effects can become paramount particularly for compliant mooring in shallower water depths. Software packages as WAMIT have the potential to calculate the second order Quadratic Transfer Functions (QTFs), while NEMOH provides the Kochin functions, $H_k(\omega)$, that can be used to calculate the drift coefficients, $C_{drift}(\omega)$, using the far-field formulation (Newman, 1967; Lee and Newman, 2005). These coefficients can be used to estimate the second order wave loads, Q_d, by use of the Newman Approximation (Newman, 1974), which allows for calculation of second order effects using only first order quantities.

When considering WECs, the linear assumption will be valid in most operational conditions, even though it can be stressed when modelling the PTO since it is designed for resonance with the waves. For mooring design, where particularly extreme conditions are considered, the assumptions of linear waves tend to become invalid, particularly because of the deployment in shallow to intermediate water depths. It is also crucial to notice that the linearization of the SWL causes inaccuracy for particularly overtopping devices, e.g. the Wave Dragon depicted in Fig. 1.7, as the transmission of energy over the device is not accounted for in the estimation of force coefficients. Furthermore, the assumption of small body motion amplitude is stressed in the extreme waves, particularly for compliant mooring systems, which are considered in this study. Finally, the Newman Approximation, used to determine the second order load from first order quantities, is determined for deep-water conditions. The accuracy of the method decreases for shallower water depths and when the natural frequency of the system is within the wave frequency band. The former might cause a challenge in the given application, while the latter should be without influence, as the system always should be designed with the natural frequency outside the wave spectrum of at least extreme waves.

Together, this forms a critical need for validation of the numerical tool and evaluation of the magnitude of errors when using the BEM for mooring design. In the remaining part of this chapter, the NEMOH code is coupled with a quasi-static and full dynamic approach by means of a hybrid model with the drag contribution from the Morison's equation. The objective is both to validate the use of linear potential theory for estimation of hydrodynamic coefficients and to evaluate the range of errors in the two types of response analyses.

4.2.2 Quasi-Static Analysis

The quasi-static approach is commonly used in the O&G sector, due to the large masses and slow responses. For a large floating WEC, it could be expected that the same methodology can be applied to some extent.

The method often implies analysis in the frequency domain and consideration of only the surge DoF, hence no inclusion of vertical displacement. In order to evaluate the influence from the mooring system, a static stiffness curve is calculated by statically offsetting the fairlead position of each line, calculating the induced horizontal force and projecting the curve into the axis of the surge motion, cf. Fig. 4.8. The stiffness in the mean position, K_m is then applied in a single DoF system illustrated in Fig. 4.8 together with the hydrodynamic coefficients found from the BEM, thereby indicating a de-coupled

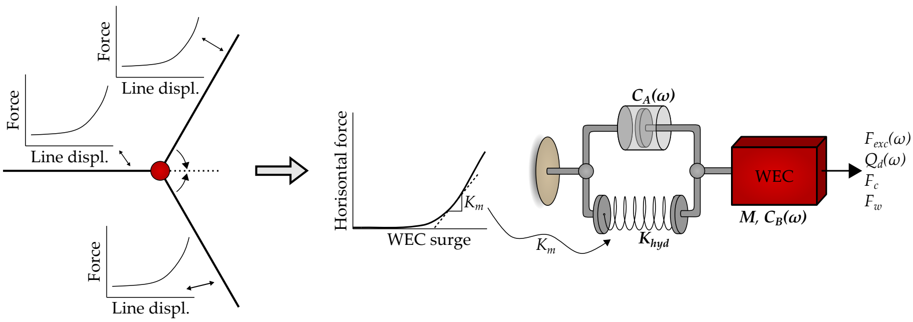


Fig. 4.8: Conceptual drawing of the system analysed in the quasi-static analysis.

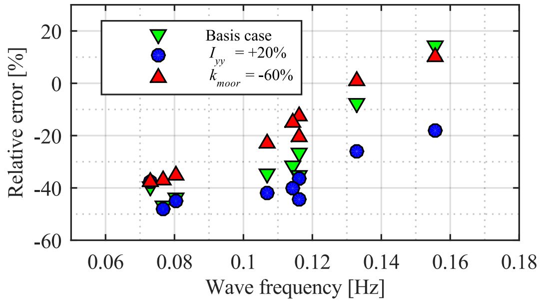


Fig. 4.9: Error between numerical quasi-static simulations and experimental data.

approach. Furthermore, the hydrostatic stiffness matrix, $\boldsymbol{K_{hyd}}$, the body mass matrix, $\boldsymbol{M}$ and the current, F_c, and wind, F_w, forces are applied. This method is e.g. described in API (2005); Bergdahl (2017).

Because of the use of frequency domain analysis, all non-linearities are linearized, including the mooring stiffness (cf. Fig. 4.8), drag loads, geometrical changes etc. and introduce critical uncertainties. Furthermore, all dynamic effects from masses and the fluid are neglected.

Thomsen et al. (2016b) (Paper [C]) used such a methodology for validation of the quasi-station approach on the application of initial mooring design. As presented in Fig. 4.9, a clear weakness of the method was found. In the operational conditions, the differences between the models are within a reasonable range, but for the extreme cases, underestimations of the numerical tensions were observed of up to 50%. This clearly indicates an insufficiency in using a quasi-static approach in design of moorings, even in the initial design, particularly because the error is an underestimation of the loads. Considering the many failures in Chapter 1 and the fact that it was shown in Chapter 3 that the present method has been applied in several design cases, the quasi-static approach forms a paramount problem. The error is clearly caused by the underestimation of motions, e.g. seen by considering the RAOs in Fig. 4.4. The configuration with the decreased MoI provides larger motions, particularly in pitch, and the quasi-static approach neglects these motions and is unaffected by the MoI, for which reason the error is seen in Fig. 4.9 to be larger for this configuration. Consequently, it is necessary to consider using a dynamic approach where all these contributions are better described.

4.2.3 Dynamic Analysis

The dynamic analysis is a more sophisticated model than the quasi-static with more included effects. Naturally, this means more computational time and effort, but it will likely provide better results. There are many different software packages, which provide a number of capabilities and applications. Many of the packages have originally been developed for the O&G or naval sectors, and with specifications that suit the analysis of the type of structures found in these sectors. Thomsen et al. (2017c) (Paper [D]) defined a list of requirements for a relevant software package. The list was produced in order to ensure that a WEC mooring system can be certified by a certification company by properly modelling all the requirements in the design standards. In general, the following parameters were found most important:

- The ability to perform a coupled analysis.
- Possibility to perform time domain analysis.
- Capability to model non-linear mooring line stiffness to account for synthetic ropes.
- Complete description of wind and current according to design standards, hence time-varying loads and vertically varying profiles.
- Inclusion of 1^{st} and 2^{nd} order wave effects.

Seven commercial software packages were assessed in Thomsen et al. (2017c) (Paper [D]), forming a simple verification that the implemented functionalities fulfilled the requirements in standards. The verification was used to short-list the software packages into two potential candidates: DeepC (DNV, 2010) and OrcaFlex (Orcina Ltd., 2015). Based on a direct comparison, it was found that there were no significant differences between the results obtained from the two codes, but some disadvantages in the usability of DeepC made OrcaFlex the choice for further analysis.

A comparison between the mooring modelling capabilities of each software package was performed, showing a range of method and levels of sophistication. This, however, is not expected to play any important influence on the results for this type of analysis, and furthermore, the capability of each software package to model tensions has been verified for other applications.

The OrcaFlex packages model the mooring line using a Lumped Mass approach (Van den Boom, 1985; Hall and Goupee, 2015), meaning that the mooring lines are divided into a number of nodes connected by a spring-damper system, with the masses lumped into the nodes at which all external (F_{ext}), internal (F_{int}) and net buoyancy (F_B) loads are acting, cf. Fig. 4.10. This forms a relatively simple methodology, compared to more traditional Finite Element (FE) or Finite Difference (FD) methods (Aamo and Fossen, 2000), but has been shown in e.g. Orcina Ltd. (2016) and Simos et al. (2004) to provide valid results. In this matter, it is important to distinguish between a FE method with a lumped mass matrix and the simpler lumped mass approach

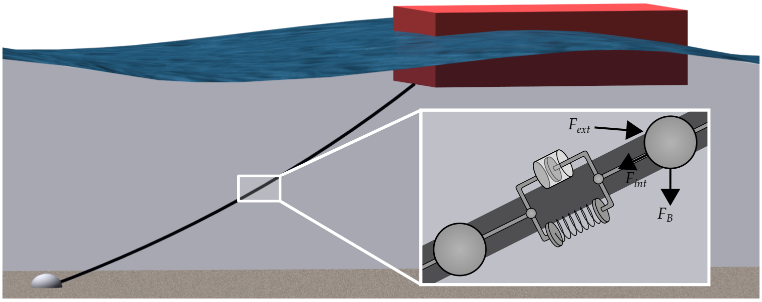


Fig. 4.10: Illustration of the lumped mass approach, where F_{ext} resembles all external environmental loads, F_{int} is the internal loads and F_B is the net buoyancy loads.

mentioned above; often the two approaches are both denoted as lumped mass approaches even though there is a significant difference between them.

The time domain model needs to be coupled with the frequency domain results from the BEM model in order to evaluate the wave load and response. In order to do so, the Cummin's Equation, (4.6), is utilized (Cummins, 1962).

$$(\boldsymbol{M} + \boldsymbol{C_{A\infty}})\,\ddot{\overline{v}} + \overbrace{\int_0^\infty IRF_r(\tau)\,\dot{\overline{v}}(t-\tau)\,d\tau}^{f_r} + f_m(t) + f_D(t) + \boldsymbol{K_{hyd}}\overline{v} = f_{exc}(t) + Q_d(t) + F_c(t) + F_w(t) \tag{4.6}$$

Where IRF_r is the impulse response function of the radiation force, $\boldsymbol{C_{A\infty}}$ is the added mass at infinity frequency, f_m is the mooring force, f_D is the viscous drag force, f_{exc} is the 1st order wave excitation force and f_r is the radiation force.

The time dependent 1st order wave excitation force, f_{exc}, can be calculated directly from the frequency response function, $\overline{F}_{exc}$, while the second order wave drift force is calculated using the Newman approximation. The mooring load is calculated in each time step, while the wind and current loads are determined in the time domain using a drag formulation. As mentioned earlier, it can increase the accuracy of the model to utilize a hybrid model, where a drag load is added in the equation of motion. In order to do so, it is necessary to determine the drag coefficients for all DoFs. For the final design, this can either be determined by use of non-linear models like CFD (Bhinder et al., 2011), SPH or by physical testing (Zurkinden et al., 2014). In the early stage and for the initial design, a simplified methodology can be applied, implying that the geometry is simplified into shapes where drag coefficients are available in literature; a method described in e.g. Wehmeyer et al. (2014). This methodology was used in Thomsen et al. (2017b) (Paper

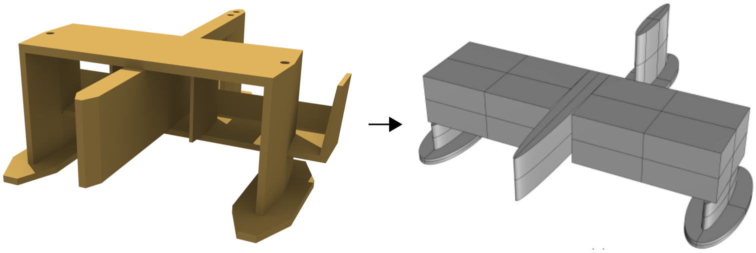


Fig. 4.11: Illustration of the simplified model used for estimation of C_D.

[E]) for a numerical model similar to the physical model in Section 4.1. The simplified model is illustrated in Fig. 4.11.

The effect of the drag contribution is clearly seen in the comparison between numerical and experimental decay tests in Fig. 4.12 (left). The additional drag provides an improvement of the quadratic damping and decreases the discrepancy between the models. The remaining error is caused by underestimation of linear damping as well. Considering Fig. 4.12 (right), by adding linear damping to the system, an almost perfect agreement can be achieved between the models. This, however, requires use of the non-linear models or experiments to determine the needed damping values.

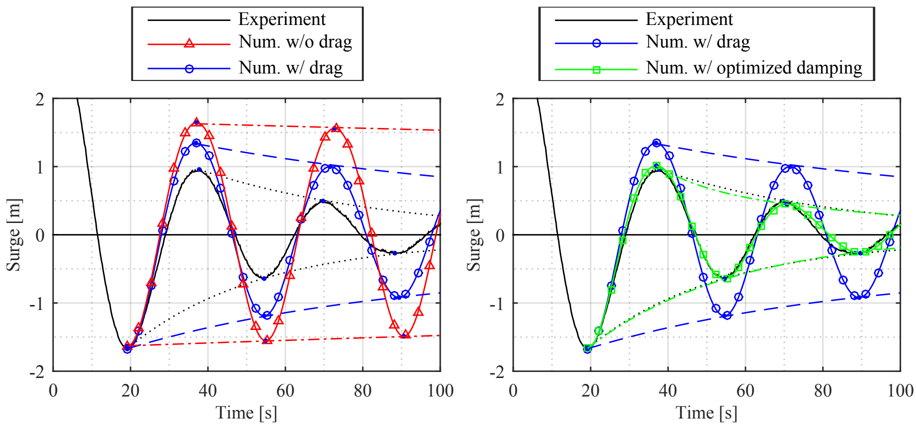


Fig. 4.12: Comparison of surge decay tests determined from experiments and numerical models. The right figure resembles an optimized and tuned model, where linear damping has been added.

Fig. 4.13 similarly illustrates how a reasonable agreement between the numerical and experimental RAOs can be achieved using the simplified model without tuning. The error between the models is most dominant at the peak frequency where the large motions put most stress on the linear theory.

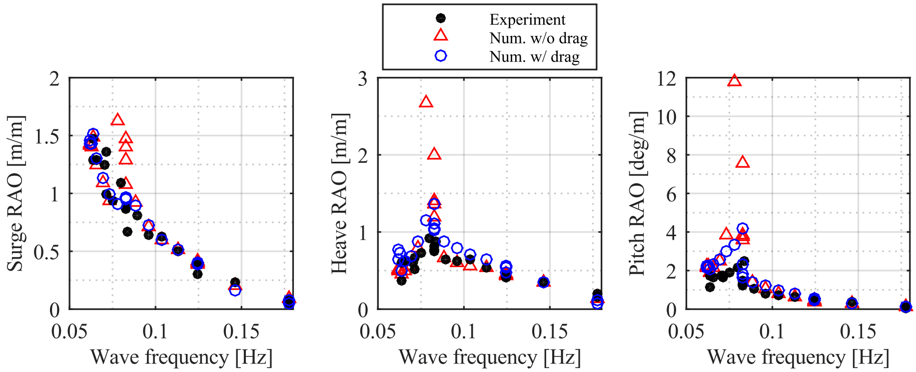


Fig. 4.13: Comparison between numerical and experimental RAOs for the models with and without drag contribution.

For mooring design and survivability in ULS, the tensions are the most important factors to model correctly. Considering the error in the numerical model, Fig. 4.14 illustrates error of maximum 12%, and more importantly it is an overestimation. The tensions are for one of the front lines, and the error in the rear line had a maximum value of 27%. The motion errors were similarly overestimated, but had a larger value than for the tension.

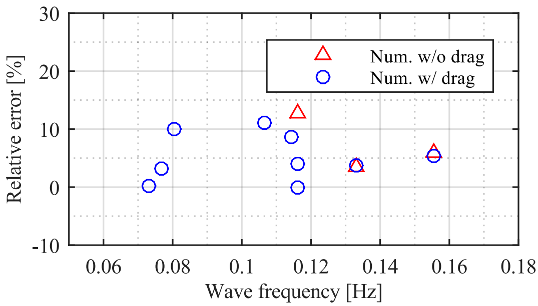


Fig. 4.14: Comparison between experimentally and numerically determined tensions in the mooring lines.

4.3 Discussion on Analysis Procedure

The previous analysis clearly indicated an insufficiency in using a quasi-static analysis, as it leads to large errors in the given applications, but more importantly underestimates the mooring loads. If considering a design scenario, this forms a critical risk of failure, even after application of safety factors. As

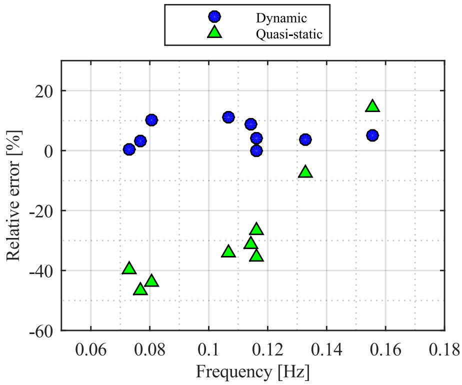


Fig. 4.15: Comparison of the error found in a quasi-static and full dynamic approach.

shown in Fig. 4.15 and in Thomsen et al. (2017f) (Paper [F]), a significant improvement is achieved by using full dynamic analysis instead, even when the model is based on the same hydrodynamic coefficient. For a WEC in extreme waves, the dynamic effects are simply too significant. Another positive effect of the use of the presented dynamic model is that the inaccuracies tend to be conservative, i.e. the loads are overestimated. Naturally, a final design will attempt to avoid this and aim at finding the best possible solution, but from a safety point-of-view, an overestimation is preferable over the underestimation in the quasi-static analysis.

As presented in Fig. 4.12, a significant improvement was achieved by adding additional damping to the model, and before a final mooring system can be certified and produced, CFD or SPH should be used to tune the model in order to make an accurate description of the damping. For an initial design, however, the error is within a reasonable range and indicates that the model can be used for a first iteration in the mooring layout and design. The use of numerical models, particularly when using BEM for the hydrodynamics and the lumped mass approach for the mooring lines, results in a very time-efficient model that today can solve simulations faster than real-time. In order to find an optimum mooring layout with a minimum amount of materials and cost, it is, however, still necessary to either compute a significant number of simulations or take advantage of a methodology that aids in the search of optimums. With the use of the numerical modelling procedure established in this chapter, the following chapter will treat this optimization.

5 Cost and Performance Optimization of Mooring Solutions

As indicated in Fig. 2.6, a mooring design process is iterative. Often, an overall system solution is chosen (cf. Fig. 2.3), and initial guesses on parameters such as number of lines, dimensions etc. have to be made and the system response verified according to the limit states. If the system is insufficient, new choices on the parameters have to be made. Every change in the layout induces important variation in the system behaviour and it is, therefore, not possible beforehand to detect which solution that is both cheap and ensures a satisfying level of survivability. One approach to find this layout is to calculate every possible combination of parameters, but this forms a significant design space with many simulations and an undesirable computational time, particularly as relatively long time series need to be simulated according to design standards. In order to find the best solution, an heuristic optimization routine can be applied and which will be described in the following sections.

5.1 Optimization of Mooring Systems

The purpose of applying an optimization routine is to find an optimized mooring layout following an efficient strategy. For a mooring solution in ULS, three overall arguments should be accounted for:

1. Survivability in ULS is ensured.
2. Design restraints on motions are followed.
3. The solution is the most cost-efficient solution.

The first two points should be ensured in all configurations in order to obtain a deployable system and even though the applied routine should be able to find these solutions, it does not form an objective for optimization, as it is rather a true/false question. Therefore, the main objective for the optimization will be to find the cheapest solution within a bound design space and such that design limits on motions and tensions are not exceeded. The optimization problem becomes:

$$\min_{x \in \mathcal{D}} f_o(\boldsymbol{x}) \tag{5.1}$$

Where f_o is the objective function which calculates the cost of a given mooring layout, $\mathcal{D}$ is the design space and $\boldsymbol{x}$ is the variable vector. $\mathcal{D}$ forms the design space and is dependent on the considered type of mooring system. Thomsen et al. (2017d) and Thomsen and Andersen (2018) (Paper [G] & [H]) considers three types of mooring systems depicted in Fig. 5.1, which resembles the type of moorings considered for the four WECs in Chapter 3. The parameters that are expected to induce variations in the response of the system and, therefore, are relevant to optimize, is listed in Table 5.1. It can be observed that the focus is put primarily on line lengths and diameters, number of lines and buoy sizes. Naturally, the optimization can be extended to also include e.g. anchors, but as mentioned in a previous chapter, this is dependent on the seabed conditions. For these studies, the type of anchor was chosen prior to the optimization and the sizes were merely found after each simulation by securing enough strength to withstand the induced loads. For a mooring optimization problem, the design space needs to be bound by outer limits of the optimization parameters. Considering e.g. the number of lines, the minimum value can be chosen following the procedure described in Thomsen et al. (2017d) (Paper [G]), where the use of less than four lines is

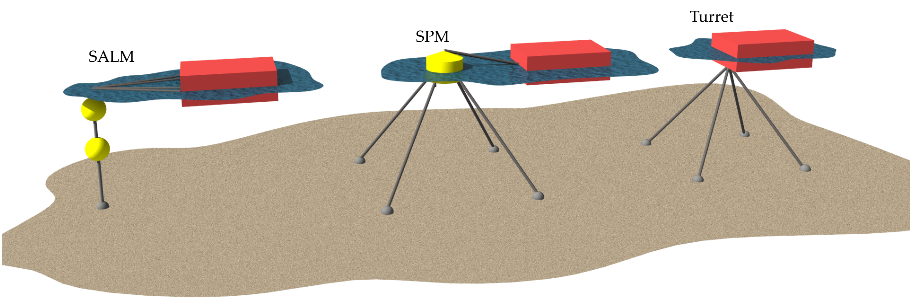


Fig. 5.1: Illustration of the three types of mooring systems considered in the four cases in Chapter 3 and in Thomsen et al. (2017d) and Thomsen and Andersen (2018) (Paper [G] & [H]).

Mooring System	Optimization Parameters
SALM	Hawser line diameter Hawser line length Upper buoy dimensions Lower buoy dimensions
SPM	Mooring line diameter No. of mooring lines Footprint radius Buoy dimensions
Turret	Mooring line diameter No. of mooring lines Footprint radius

Table 5.1: Definition of optimization parameters for the three types of mooring systems considered in Thomsen et al. (2017d) and Thomsen and Andersen (2018) (Paper [G] & [H]).

found as non-viable solutions due to ALS. Maximum limits on line number, footprint radius etc. should be defined in order to avoid unrealistic solutions and will to some extent be based on case-to-case considerations and experience. Other parameters such as line diameter and buoy dimensions, will be based on available solutions from manufacturers.

The objective function evaluates the total cost for the given system, hence is a model, which calculates cost and is capable of proving that the system can survive. This means that the validated numerical model from the previous chapter can be applied, simulated for three hours according to e.g. DNV-OS-E301 (DNV-GL, 2015d), tensions and motions extracted and verified, anchors designed to these loads and the parameters fed into a cost database, which calculates the lifetime cost of the system, cf. Thomsen et al. (2017d,e) (Paper [G]). Despite being a relatively time-efficient model, simulating the entire design space to find the optimum solution will eventually become unrealistic.

Different methodologies have been developed for optimization problems like this, where no derivative information of the response surface is available, but these often requires a significant number of function evaluations. Thomsen et al. (2017d) (Paper [G]) adapted a surrogate-based optimization routine, which according to e.g. Müller and Shoemaker (2014); Ortiz et al. (2015); Ferri (2017) is advantageous to use for computationally demanding derivative-free problems, with studies like Rios and Sahinidis (2013) and Ruiz et al. (2017)

comparing and discussing the applicability of different procedures and models. A clear benefit of the surrogate-based approach is the requirement of only a limited number of function evaluations. The surrogate model, $s(\boldsymbol{x})$, can generally be described by Eq. (5.2), illustrating how $s(\boldsymbol{x})$ is an approximation of the response surface (Müller and Shoemaker, 2014). During the optimization routine, a selected number of objective function evaluations will be executed and the surrogate model improved, cf. Fig. 5.2.

$$f_o(\boldsymbol{x}) = s(\boldsymbol{x}) + \epsilon(\boldsymbol{x}) \tag{5.2}$$

Here, ϵ is the error between the surrogate model, s, and the objective function, f_o. Since the model is evaluated throughout the entire design space and the model is fitted to this data, a description of the entire response surface is obtained, contrary to many other algorithms whose only output is the optimum value. This forms a considerable benefit of surrogate models, as it allows for manual inspection of the design space and allows the designer to search for better and more attractive solutions.

The optimization routine follows a procedure as listed below (Müller, 2014).

1. Initial Design of Experiment (DoE).
2. Evaluation of objective function in points of DoE.
3. Construction or update of surrogate model.
4. Selection of new points for evaluation.
5. Evaluation of objective function in selected points.

In the DoE, a number of layouts are selected within $\mathcal{D}$ in order to give an initial description of the response surface. The objective function is evaluated in these points, and a surrogate model is fitted to the results (cf. Fig. 5.2). By following a global search strategy, a number of new points are selected for evaluation. Here, the selected points are focused around the expected minimum in the surrogate model, while a number of randomly spread points

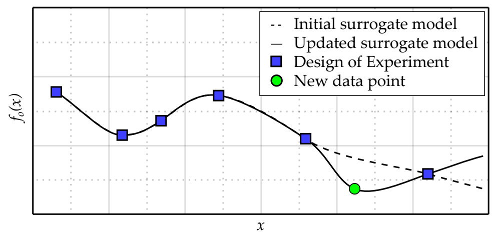


Fig. 5.2: Illustration of the concept of a surrogate model.

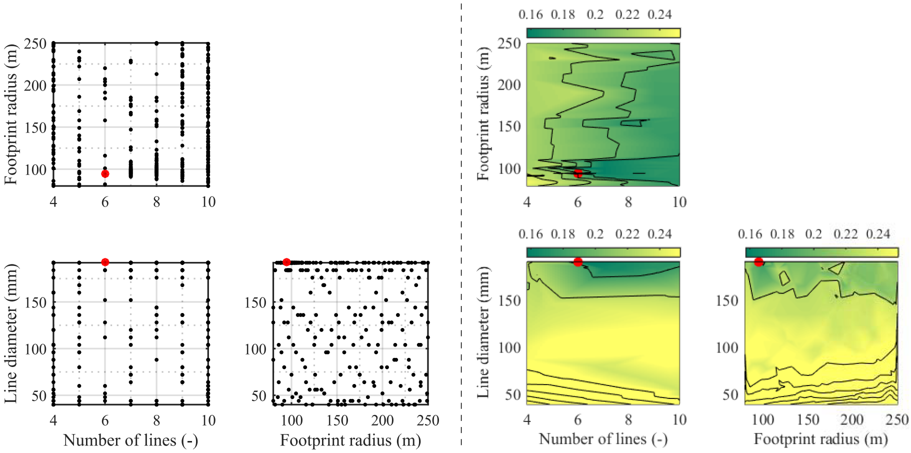


Fig. 5.3: Example of evaluated point in the design space during an optimization routine (left) and the response surface (right). In the latter, one parameter is kept constant in each graph (as the optimum value). The surface resembles the total mooring cost normalized according to the total WEC cost.

are also evaluated in order to search other areas. The process (Steps 3-5) is continued until a termination criterion is met (often a chosen number of evaluations), at which the solution should have converged and the optimized solution found.

As described in Chapter 2, WECs do not have strict requirements to motions, but will have some limit specified due to e.g. the umbilical. It is undesirable for the optimization routine to search for solution in the design space where the limits, both for excursion and line tension, are exceeded. Therefore, a cost penalty is applied which increases with expanding exceedance of the limits. Since the routine is searching for the minimum points, the surface will obtain a shape that makes it unattractive for the solver to search these areas.

Fig. 5.3 presents an example of the output from the optimization of one case in Thomsen et al. (2017d) (Paper [G]). The mooring cost has been normalized according to the overall lifetime cost of the full structure and since all evaluations are plotted, the areas with the highest cost correspond to solutions exceeding the limits and have a cost penalty applied. The left part of the figure presents the evaluated points in the design space, while the right illustrates the response surface. From the selected points, it is observed how the full domain has indeed been investigated, but most focus has been centred on the optimum (red marker).

This figure can be used for cost optimization of the mooring and, furthermore, be used for subsequent consideration on safety and reliability. The optimization routine always searches for the global minimum, but this might be located in an area of the response surface where the gradient is relatively large. This means that even minor changes or uncertainties on input and response can influence the cost significantly. Therefore, the complete description of the response surface becomes a clear advantage, as it is possible to make a manual evaluation to identify if solutions that are more reliable are available. If other areas with smaller gradients are present and where the cost differences are small, it can be preferable to consider such solution instead. Similar, it is possible to detect the influence on cost of e.g. using a higher number of lines. From a safety point-of-view, it will be preferable to use a high number, and if the cost difference is small, in most cases it would be the choice of solution. Clearly, the optimization routine is both advantageous in finding the optimum solution, but it is also allowing for easy evaluation of the results.

5.2 Cost of Different Mooring Systems

Thomsen and Andersen (2018)(Paper [H]) adapted this optimization routine and found the optimal mooring layout for three different mooring systems (cf. Fig. 5.1):

- A SALM system with two submerged buoys, a tether of steel rods and a hawser composed of four nylon lines.
- A SPM system with nylon lines, a surface buoy and a rigid hawser.
- A turret system with nylon lines.

These systems correspond to the type of systems presented in Chapter 3 and are considered by the developers of the four WECs in Fig. 1.7. In order to allow for investigation and comparison of cost, the systems were all designed for a generic barge structure and exposed to the same environmental conditions. Fig. 5.4 presents a normalized cost breakdown of the mooring systems for the found optimum layouts.

It is clear that the cheapest of the three systems for this design case is the SPM system. This is primarily a result of the significantly lower cost of the component in this system compared to the others. The component cost is further broken down in Fig. 5.4. Considering the figure, it is clear that the actual mooring line cost is with a limited influence on the overall mooring cost. In Chapter 3 and Thomsen et al. (2016c) (Paper [B]), the SALM system was highlighted as a cheap system due to its low mooring line cost, and the same conclusion can still be drawn, but it is clear that the overall component cost of the system is highly affected by the anchor cost. This type of mooring system

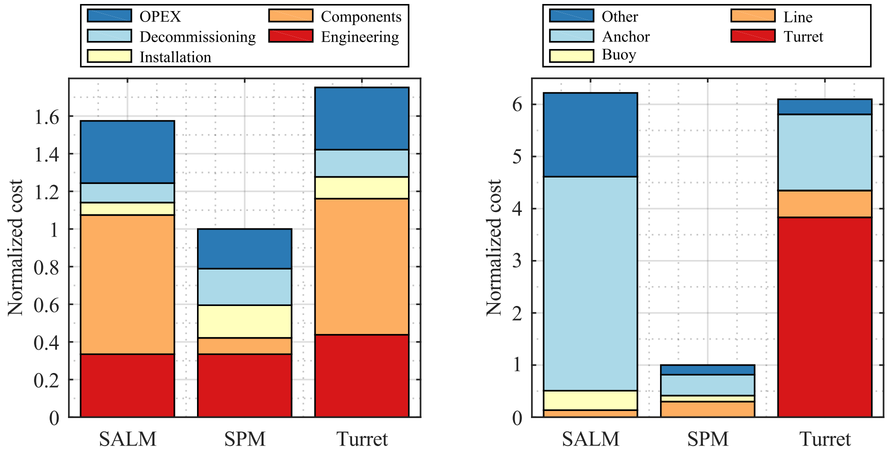


Fig. 5.4: Normalized optimized cost for the three mooring systems in Fig. 5.1 when applied to a generic barge structure and similar environmental conditions. The cost of all systems are normalized according to the cheapest solution. The left plot illustrates the total cost, while the right illustrates component cost.

requires a significant vertical strength, which here is provided primarily from gravitational forces. Consequently, large anchor material is needed, resulting in a significant anchor cost compared to the other two systems, which can use drag-embedded anchors. It is critical to consider other types of anchors with vertical strength such as piles or suction bucket for the SALM system in order to find more realistic solutions. The turret system tends to become stiffer than the other two systems and, hence, experiences larger tensions with requirement of larger anchors as well, making them more expensive for this system than for the SPM. The most dominating cost for turret system, however, is the actual turret bearing, which tends to become complex and expensive. The part labelled as *Other* covers connections, steel rods etc. This is most profound in the SALM system due to the steel tether.

The engineering cost becomes comparable for all systems, with a slightly more expensive cost for the turret system due to the complexity of designing the bearing. The installation cost is highly dependent on the number of lines and type of system, and since the SALM is limited to installation of only one SALM-leg, it becomes cheapest to both install and decommission. The OPEX cost is directly related to the CAPEX cost and will, therefore, be lowest for the SPM.

Even though each system might have some advantages over others, it is clear that from a cost point-of-view, the SPM system provides a high benefit because of its simplicity without expensive turrets and its applicability of common anchor types. Even though the cost database (Thomsen et al., 2017e) uses current and detailed data, the comparison of the three systems is still affected by some uncertainty. Several parameters, such as handling of umbilical etc. in each solution has not yet been implemented and this might cause significant influence on the cost of each system.

Thomsen et al. (2017d) (Paper [G]) further applied the optimization routine on the four large WECs illustrated in Fig. 1.7 and designed them for the site conditions shown in Fig. 3.1. The developers of these devices have all considered mooring systems like those depicted in Fig. 5.1. In addition to the optimized mooring layouts, the response surface also provided valuable information on the influence from each of the optimization parameters on the overall cost.

An example of the response surface from one case is illustrated in Fig. 5.3. Despite the previous observation on the mooring line taking up a minor part of the overall cost, it is clear to see from the figure that the mooring line diameter is the most determining for the overall cost. This is because the diameter determines the line strength and axial stiffness and, as such, the total stiffness of the system; therefore, it is necessary to have large lines for all these large devices in order to ensure sufficient strength. In order to find the optimum number of lines, it is necessary to find a balance between having few anchors of a larger size, and having a larger number of smaller sizes. When reducing the number of lines, more load is put on the anchors (and lines) and, thereby, larger types are required. Naturally, this increases the cost. Smaller lines and anchors can be allowed when using more lines, since each line only needs to accommodate for a minor part of the load. This might not necessarily decrease the cost because the installation becomes more expensive. For such problems, the optimization and surface plot in Fig. 5.3 becomes useful.

The footprint radius primarily affect the cost through changes in the system stiffness, but the influence is not as outspoken as for the remaining parameters. For the SALM system, the mooring stiffness is highly dependent on the buoy sizes, which, therefore, also affects the cost. When using buoys with minor effect on the stiffness, the restoring force primarily arises from the elasticity of lines, putting more stress on them and increasing the risk of full stretch of the system with higher loads. The optimization routine showed the influence of the top and the bottom buoy (cf. Fig. 5.1) and indicated that the bottom was without any important influence. The best solution is to consider a larger buoy close to the SWL.

Some of the same consideration can be made for the buoy in the SPM system. It was shown in Thomsen et al. (2017d) (Paper [G]) that keeping the buoy relatively small helps improve the system response and thereby the cost. When using a large buoy, there is larger pretension in the lines and insignificant vertical motions of the buoy. This means, like for the SALM system, that the restoring force primarily arises from the stiffness of the lines. Having a smaller buoy introduces vertical motion of the buoy itself and, thereby, more compliance and less loads in the system.

All the optimization is based on the procedure defined in the previous chapter, which is aimed at initial design. For the OT device, there are significant inaccuracies in the model, since it cannot account for the overtopping discharge. This means that the loads are overestimated and most likely also the cost. Still, when comparing the cost of the initial proposed solution in Thomsen et al. (2015) (Paper [A]), it was found that the optimized mooring solutions found in Thomsen et al. (2017d) (Paper [G]) could lower the CAPEX cost by averagely 16%. For the SALM system, an increase of the cost was observed. Considering the evaluation in Chapter 3, it is somewhat difficult to compare the costs, since the initial layout could not be guaranteed to survive extreme conditions due to insufficient designs. Obtaining a cost increase is, therefore, still a positive outcome of the study, since durability of the systems are improved and a standardized design procedure applied. A tentative comparison between coarsely estimated LCOE for the initially proposed moorings and those found in the optimization, indicated a reduction of 5-10%. However, this result is strongly biased by a significant uncertainty on the OPEX cost in the initial estimates. During the project period, much new experience has been gained on cost estimations, and this has highly influenced this comparison. Future studies should address this topic more detailed.

5.3 Sensitivity Discussion of Mooring Solutions

When utilizing the optimization routine, the solver attempts to find the cheapest solution, which fulfils the design criteria according to the chosen design standard. This often implies that the strength of the lines are fully utilized, which puts a significant importance on defining the environmental loads with high accuracy. Sensitivity to insecurities and uncertainties on the environmental conditions are not taken into account in the optimization routine. Thomsen and Andersen (2018) (Paper [H]) briefly addressed this problem by varying the environmental conditions with $\pm$ 15% in order to investigate the sensitivity of design tensions in the three optimized mooring systems. Fig. 5.5 illustrates the outcome of this analysis, indicating how an increase in wave height of 15% can result in an increase in the tension of up to 86%. The SALM system tends to be most vulnerable on this matter, explained by the fact that the system becomes fully stretched and results in a steep stiffness

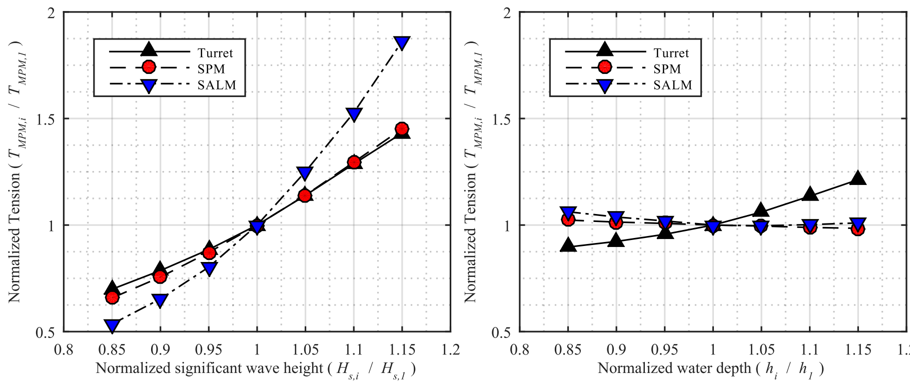


Fig. 5.5: Influence on mooring line tensions in the three mooring systems from variations in the significant wave height (left) and water depth (right).

curve, which, consequently, leads to a large relative load increment. Similar analyses were performed for the wind and current, giving maximum increase in the line tension of 10%. Increasing the wave period led to increases of approximately 20%. The three systems showed an important difference when varying the water depth. The tension in the turret system increases with 20% when increasing the water depth while the SALM and SPM were almost unaffected. Considering earlier statements on the desire to deploy near-shore and the fact that tides have been neglected in all initial design, this stands as a strong result. For a turret system, changes in the water depth are critical to include in the design.

The analysis indicated that caution should be put in assessing and defining the environmental conditions, as significant increases in tensions can occur. This further highlights the problems in Chapter 3 with incomplete description of the conditions for the initial mooring systems. Following a standardized procedure as presented in this thesis aids in improving the mooring design, and by using the optimization routine, a cost-efficient solution can be found. In order to get a better description of risk and reliability, even more effort should be put in work like the sensitivity analysis.

6 | Conclusion

In the search for renewable and sustainable energy, the focus is shifting from the traditional fossil sector and onto sectors that are more immature. The wind and solar energy sectors are already well-established, but in order to achieve a completely carbon-free energy production, a mix of multiple renewable energy sources is needed. Wave energy provides a strong theoretical potential, but none of the developed devices and concepts are yet at a stage of development where they can be considered commercial. One topic which provides an aid in the effort of making floating WECs more mature is the improvement of mooring design. Several failures have been experienced by now and it has been observed that the mooring cost is undesirably high. Despite the strong desire to lower the cost and make the LCOE more attractive, the main concern at present is to prevent any more failures of moorings and its negative effect on the public's perception of wave energy.

This thesis treated the topic of mooring solutions for large wave energy converters and addressed four objectives: Assessment of current state of mooring design, identification of potential solutions, investigation of design procedures and optimization of the considered systems. The work took its basis in four Danish devices and demonstrated the initial state of their basis mooring design. It was apparent that even though mooring solutions have been proposed, they are indeed initial guesses and not all elements in the design process has been duly included such that survivability is secured. This is both a result of the fact that the implementation of design standards often is insufficient and more importantly, because quasi-static design approaches have been used. By validation against experimental data, it was shown how this method generally underestimates the mooring line loads with a maximum error of up to 50%. This forms a critical problem in mooring design, and instead it was presented how a full dynamic numerical approach provides results that are more reliable. Even though errors in line tensions of around 10% still were present, the error was an overestimation and hence, it provides conservative estimations used in the design. Considering the many failures and the relatively limited experience in offshore deployment in this immature sector, an overestimation is acceptable by now. Consequently, the presented modelling procedure can be used in initial design, but for final designs, the model must be improved with use of more sophisticated theories or experimental data.

When searching for cost-optimized solutions, it was illustrated how a surrogate-based routine can be used. The method utilized the numerical model and a constructed cost database, and searched for optimized solutions within a defined design space. This secured that both survivability in ULS and cost-efficiency was obtained. The approach was applied for the four Danish WECs and it presented the capability to find optimized moorings. The study illustrated a reduction of CAPEX and LCOE by comparing the obtained cost with estimations for the initial solutions. However, the comparison is influenced by a significant uncertainty due to the state of the initial moorings and because more experience on estimating cost has been gained since the initial evaluations. Nevertheless, the results indicate that cost reduction can be found.

The study provided initial verified designs according to a chosen design standard but naturally, many more considerations will be needed before a final solution is achieved e.g. more limit states like FLS, but also topics such a mooring loads in misaligned environmental impact as this thesis only treated aligned loads.

Throughout the thesis, different types of mooring systems have been considered, taking basis in the initial basis systems proposed by the WEC developers. This included a SPM, SALM and turret systems with chain. From a simple comparison, it is evident that systems with chains are unfavourable for this type of structures and their deployment sites. Synthetic lines, particularly nylon lines, were found to be a very promising solution, while also SALM type systems were highlighted. The latter were later found to cause some problems because of the need for heavy gravity based anchors. In future work, it will be advantageous to put focus in other type of anchors and investigate how this can affect the cost. A promising type of mooring system was found to be the SPM system, as complex turrets are avoided and well-known and available anchor types can be applied. Further, this system was seen to be less sensitive to environmental conditions such as waves and water level variations.

Future Perspectives

The present work has all been based on linear wave theory, which causes important inaccuracies when designing compliant moorings in extreme waves.

It is critical that final designs and future work are centred on higher-order theories to improve the presented models. CFD or SPH can be beneficially used to tune the BEM model and provide a much more accurate description of the response. This will further aid in improving particularly the description of overtopping devices. CFD and SPH might not be applicable to full design due to the long simulations, but will indeed be valuable in the production of more reliable time-efficient models.

Mooring design is based significantly on known experience, but this thesis has put most focus on nylon, which at present is still relatively new and untested. This material is currently being investigated and tested comprehensively and will be deployable in the near future, providing increased experience and reliability. There is clearly great potential for this type of solution to be used when full-scale WECs will be deployed. It is, furthermore, important to notice that in addition to the types of lines treated in this thesis, many novel types of lines are being tested and will, potentially, be the centre of focus in later research.

The work has aided in gaining more experience on mooring design and methodologies in analysis of the systems. Several considerations on types of mooring and materials have been tested and the advantages and drawbacks highlighted. Even though the work is focused on wave energy converters, the results will be applicable on other types of devices where similar responses are expected, e.g. FOWT or similar. Hopefully, the conclusion can be applied in the aid of bringing wave energy closer to commercialization.

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Paper Collection

Paper A

Assessment of Current State of Mooring Design in the Danish Wave Energy Sector

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The paper has been published in
EWTEC, ISSN: 2309-1983 (08B5-5), 2015.

11th ewtec 2015 Proceedings of the 11th European Wave and Tidal Energy Conference 6-11th Sept 2015, Nantes, France

Assessment of Current State of Mooring Design in the Danish Wave Energy Sector

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***Abstract*—The mooring system is a vital part of any floating wave energy converter, both in terms of ensuring station keeping but also as it constitutes a significant share of the total cost. Motivated by the considerable amount of failures due to insufficient mooring and the cost of mooring today, the present study outlines the design procedure recommended by design standards and provides considerations on choice of tools for analysis. This is compared to the procedure used by four wave energy converter developers, to illustrate the state of their current mooring design. The study shows a clear tendency amongst the developers of using existing offshore tradition and some inconsistency in chosen design procedures and parameters, by means of varying analysis types, environmental conditions and level of details. The lack of details in the design means that for all the investigated developers, despite having an overall solution, no final design has been achieved yet and further investigation is needed.**

***Keywords*—Wave Energy Converter, Mooring system, Quasi-static analysis, Dynamic analysis, Station keeping, Floating Wave Energy Converter, Large Wave Energy Converter, WEC**

I. Introduction

Many concepts for wave energy conversion exist today, implemented in a large variety of different devices, many of which are floating structures. Offshore floating structures are exposed to a range of environmental loads, resulting in motions of the device, which need to be restrained in order to ensure station keeping and stability. Usually, a mooring system is applied, which by definition is a system of lines connecting the floating device to the seabed.

Mooring systems have been widely used by the Oil & Gas and naval sector, giving much experience and knowledge, which have led to a variety of design standards and guidelines. To a large extent, the wave energy sector has adapted this experience, but still a large number of Wave Energy Converters (WECs) have failed due to insufficient mooring [1]. Furthermore, the WEC sector needs to minimize the overall cost of mooring to make more economical mooring solutions. This need is not as pronounced in the Oil & Gas sector as mooring only covers a small part of the overall structure cost. This is illustrated by the fact that mooring only takes up 2% of the cost of a floating Oil & Gas structure, stated by [2] and illustrated in Fig. 2. In the same article, the cost of a mooring for a WEC takes up 18% of the total cost and by some authors, such as [1], is estimated to take up 30%. In addition, the finances are significantly larger in the Oil & Gas sector than in the wave energy sector. Optimizing mooring design will reduce the Cost of Energy (CoE) and bring wave energy closer to commercialization.

The purpose of the present study is to clarify the state of current mooring designs in the wave energy sector. The study is based on case studies of four WECs, characterised as being large floating structures. The paper is structured with an introducing section, characterizing the present study and followed by a description of standard design procedures. By outlining the case studies, it is clarified to what extent the standards have been used, and this is intended to illustrate the current state of mooring design in the wave energy sector and in which areas further investigation and work is needed.

II. Characterization of Present Study

A number of the existing WECs are designed to be in resonance with the possible sea states at the deployment sites to maximize energy absorption. Others are of a much larger mass, giving a different response, which results in low responsiveness in most sea conditions. These relatively large structures are the focus of this study. Examples of this kind of WEC include the Floating Power Plant [3], LEANCON Wave Energy [4], Wave Dragon [5] and KNSwing, all illustrated in Fig. 1.

Depending on the type of WEC, the mooring is either influencing the energy absorption or not. In the wave energy

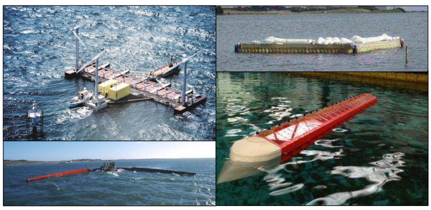

Fig. 1. Examples of WECs considered as large floating devices. Top left is the Floating Power Plant [3], Top right is LEANCON Wave Energy [4], bottom left is the Wave Dragon [5] and bottom right is the KNSwing.

sector, as e.g. defined by [1] and [6], mooring systems can be classified as:

Passive mooring: when the only purpose of the mooring is station keeping and it does not influence Power Take-Off (PTO).

Active mooring: when the mooring influences the dynamic response of the system and the energy production.

Reactive mooring: when the mooring provides reactive forces, influencing the PTO.

The WECs investigated in this study can be classified as self-referencing bodies with passive mooring systems. Further, they are expected to be deployed in shallow water conditions, which induce challenges in terms of the wave kinematics in shallow water. The steeper waves result in a larger net transport and excursion of the water particles, as exemplified in [2] and [7]. This makes the wave interaction more important than it would have been in deep water conditions. Since mooring stiffness was found to increase in shallow water depths, as investigated by [8], much larger mooring loads can be expected in these conditions. In addition, a load contribution on the anchor can be expected due to more shear stresses at the seabed. Deployment in shallow water therefore puts more demands in the mooring system, and yet provides less vertical span to accommodate them.

As stated, mooring systems are widely used in other offshore sectors, and many different principles and solutions exist [9], [10], all of which can be considered as passive mooring systems. A number of examples are illustrated in Fig. 3, and listed below:

Spread mooring system: consists of a number of mooring lines attached to the perimeter of the floating device. The lines restrain horizontal excursion and are preventing rotation of the structure.

Catenary Anchor Leg Mooring (CALM) system: a system of mooring lines connected to a buoy, to which the floating structure is connected. In this system, the structure is able to rotate around the buoy according to the environmental load direction.

Single Anchor Leg Mooring (SALM) system: a single mooring line is anchoring a buoy to the seabed. The floating structure is connected to the buoy allowing 360° rotation according to load direction.

Turret mooring system: a catenary mooring system connected to the floating structure and allowing 360° rotation according to the environmental load directions.

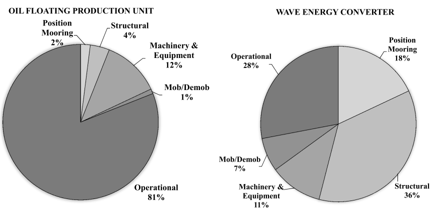

Fig. 2. Cost breakdown of oil floating production unit and wave energy converter. Adapted from [2].

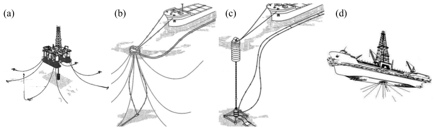

Fig. 3. Spread mooring system (a), CALM system (b), SALM system (c) and turret mooring system (d). Adapted from [10].

These kinds of systems often appears in floating Oil & Gas and naval structures, and are traditionally build up by heavy chains and steel wire ropes [9].

III. Design Procedure

Appropriate design procedures for mooring systems in the traditional offshore sectors have been described in several design standards. Mooring design and procedures for WECs have by now also been treated in several papers like [1], [11]–[14], etc. To some extent, the experience from the other offshore sectors can be applied to the wave energy sector, but some differences are present and need to be taken into account. The global procedure for designing mooring systems is illustrated in Fig. 4, and would be appropriate for the wave energy, Oil & Gas and naval sector.

The overall purpose of mooring design is to obtain a system that can keep the WEC on station in all conditions relevant for the considered site. The initial task is therefore to consider and describe the relevant site. Environmental conditions related to wave, wind and current are especially vital. Considerations on power cables, array installation, surrounding structures and mooring systems, etc. should be made in order to define restrictions on allowable excursion and footprint of the WEC.

Based on the outcome of the site considerations, a mooring layout can be defined by means of lines, anchors, buoys etc. and described by materials and dimensions.

A well-designed mooring system needs to be able to react with environmental forces without a large offset of the floating device, but at the same time comply with the offset without inducing large loads. The system should therefore be validated according to design standards through a performance analysis, providing line loads and excursions. Different limit states must be considered, covering both fatigue (FLS), accidental (ALS) and ultimate limit state (ULS). The latter one is often the one with greatest influence on the overall cost [15]. Methods that can be used in the performance analysis are outlined in the following, together with a description of guidelines from design standards.

A. Design Standards

A large variety of standards exists today primarily aimed at the offshore Oil & Gas or naval industry, with a few producers of guidelines listed below.

DNV: Det Norske Veritas
API: American Petroleum Institute
ISO: International Organization for Standardization
ABS: American Bureau of Shipping
IEC: International Electrotechnical Commission

The most relevant from these, when considering mooring design, are the DNV-OS-E301 [16], API-RP-2SK [10] and ISO 19901-7:2005 [17]. For a full design, rules and guidelines on all components must be considered including i.a. chains (e.g. DNV-OS-E302 [18]), fibre ropes (e.g. DNV-OS-E303 [19]) and steel wire ropes (e.g. DNV-OS-E304 [20]). These types of standards, together with guidelines as the EMEC [21] and Carbon Trust [22] guidelines, can be considered for mooring design for WECs.

During several occasions, e.g. in [14], a relaxation of the standards when designing for WECs has been proposed, based on less severe consequences in case of failure. IEC is at present producing a set of standards for Marine Energy Converters (MEC), with the IEC-62600-10 [23] directly aimed at mooring design. The standard is largely based on API-RP-2SK and ISO 19901-7:2005 with modifications to better accommodate the needs in the wave energy sector. At present, the standards are still in progress and not yet valid, but the Danish Standard (DS) has suggested to directly adapt these.

As IEC and DS are not yet valid, the content of this paper primarily deals with API and DNV standards.

Using the DNV-OS-E301 standard provides the possibility to consider a consequence class for the mooring system, divided in two classes: Class 1 in which failure is unlikely to lead to unacceptable loss of life, collision with other structures, uncontrolled outflow of oil, capsizing or sinking, and Class 2 where failure may lead to unacceptable consequences of these.

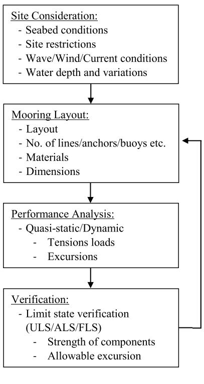

Fig. 4. Example of global design procedure for a mooring system as defined in e.g. DNV-OS-E301 [16] and [1].

TABLE I
PARTIAL SAFETY FACTORS FOR ULS/ALS DEFINED BY DNV-OS-E301 [16]

Consequence Class	**Type of Analysis**	**Partial safety Factor for Mean Tension**	**Partial Safety Factor for Dynamic Tension**
1	Dynamic	1.10 / 1.00	1.50 / 1.10
2	Dynamic	1.40 / 1.00	2.10 / 1.25
1	Quasi-static	1.70 / 1.10	
2	Quasi-static	2.50 / 1.35	

TABLE II
PARTIAL SAFETY FACTORS FOR ULS AND ALS DEFINED BY API-RP-2SK [10]

Type of Analysis	**Partial Safety Factor for ULS**	**Partial Safety Factor for ALS**
Dynamic	1.67	1.25
Quasi-Static	2.0	1.43

The API-RP-2SK standard does not consider consequence classes like these, hence consider failure of all structures to be of equal consequence. The IEC and DS standards use the API standard as baseline for safety factors etc., and assume these to be for a consequence class 3, and defines the classes as: Class 1 where failure leads to minimal human injury, minimal environmental impact, minimal financial and property damage, and class 2 with serious impact of the mentioned parameters and class 3 where failure may include loss of life, significant damage to marine environment and substantial financial and property damage.

In [13] the safety factors of DNV and API (cf. Table I and II) are compared, and it is concluded that use of API gives an increased safety margin. Use of DS and IEC, with the defined consequence classes, to some extent provides the relaxation proposed in [14].

The standards require design in several limit states, covering the ULS, ALS and FLS, but in this study most focus will be on the ULS, as this in most cases is the state determining for the cost of the system according to [15], ensuring the system's ability to withstand even the most severe storm conditions. The ALS ensures the possibility of the system to withstand an extreme situation where one line is broken. Satisfying ALS therefore also ensures the possibility of being able to remove one line for maintenance, without causing failure of the mooring system.

To ensure a satisfying annual probability of failure, DNV-OS-E301 and API-RP-2SK require use of the safety factors defined in Table I and II.

As seen from Table I, DNV-OS-E301 defines separate partial safety factors for the mean tension and for the dynamic tension. The safety factors should be applied according to Equation 1.

$$S_c - T_{c-mean}\ \gamma_{mean} - T_{c-dyn}\ \gamma_{dyn} \geq 0 \tag{1}$$

With S_c being the characteristic capacity of the material, T_{c-mean} the mean characteristic tension, γ_{mean} the partial safety factor for the mean tension, T_{c-dyn} the characteristic dynamic tension and γ_{dyn} the partial safety factor for the characteristic dynamic tension. The characteristic capacity of the material is defined by $S_c = 0.95\ S_{mbs}$, where S_{mbs} is the minimum breaking strength.

The API-RP-2SK specifies a global safety factor, which is required to be applied, according Equation 2. For the IEC and DS, a design factor dependent on the consequence class is multiplied with the safety factor, with Class 1 = 1.00, Class 2 = 1.3 and Class 3 = 1.5 (Safety factors in Table II correspond to consequence class 3).

$$\frac{S_{mbs}}{T} \geq \gamma \tag{2}$$

In ULS, the analysed environmental conditions must correspond to extreme events with return periods, slightly varying for the different standards. DNV-OS-E301 defines a combined extreme event, with a 100-year return period for wave and wind loads, and 10-year return period for current loads. API-RP-2SK and several of the other standards define a minimum of 100-year return periods for all environmental loads. The most severe load directions probable of occurring, should be considered.

In the performance analysis, it needs to be considered if a quasi-static analysis or dynamic analysis should be performed. DNV-OS-E301 states that a quasi-static approach in most cases is appropriate, while API-RP-2SK (and IEC and DS) states that it can be used as a preliminary analysis, but that a final system must be analysed in a dynamic analysis. The two types of analysis will be outlined later in this paper.

B. Environmental Loads

Floating structures are exposed to a range of different environmental loads, mainly dominated by wave, wind and current. As stated, especially loads from extreme events (ULS) are of paramount importance and shall be considered. The three types of environmental loads are described in the following.

1) Wave Loads: Interaction between the waves and the WEC results in forces on the structure. In mooring design, these forces are divided into wave frequency (WF) forces, mean wave drift forces and low frequency (LF) forces (cf. Fig. 5), all of which needs to be considered [24]. The WF forces are first order forces that oscillate at the frequency of the incoming waves and of all the wave force components, the WF forces in general have the largest amplitudes. For mooring design, this force component puts requirements to the compliance, as the mooring system needs to be compliant enough to prevent large loads in the lines.

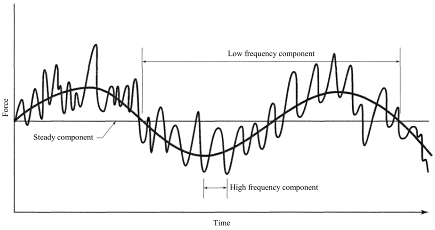

Fig. 5. Example and definition of the wave force components. Adapted from [10].

For small structures, with a diameter less than a fifth of the incoming wave length, a Morison Equation approach can be used in the estimation of the loads. For the purpose of this paper, with relatively large structures in focus, diffraction needs to be included, and a calculation procedure based on linear potential theory, e.g. through a boundary element method (BEM), needs to be adapted. Both a frequency and time domain analysis can be considered.

In addition to the first order wave force, the waves induce second order forces, combined of a steady component and a slowly varying (LF) component in irregular waves. This second order wave effect is significantly affecting the surge, sway and yaw motions, inducing a need for a stiff system that can prevent large offset of the WEC.

2) Current Loads: In general, current is treated as a steady force, and defined in API-RP-2SK [10] and DNV-OS-E301 [16] as a combination of tidal, circulation and wind-generated currents. The mean current can be determined using a drag force calculation as defined in Equation 3, and stated in several design standards.

$$F = \frac{1}{2}\ \rho\ C\ A\ V^2 \tag{3}$$

Where F is the mean current force, ρ is the fluid density, C is a drag coefficient dependent on the shape of the structure, A is the cross-sectional area and V is the current velocity. In calculation of current loads, the vertical variation of speed with water depth must be included as either a standard or a site-specific profile.

3) Wind Loads: The wind load shall be treated as a steady mean force described by a vertical, standard wind profile, combined with effects of gusts, in terms of a time varying wind force component, which results in slowly varying motion. The gust effect should be described by an appropriate wind spectrum. The mean wind force can be calculated by a drag force formulation similar to Equation 3.

C. Quasi-Static Analysis

In offshore Oil & Gas and naval sectors, mooring systems are often modelled by a quasi-static approach, justified by the very large structures and corresponding masses. The responses of the structures will be very low in most conditions and the induced velocities will be of an insignificant magnitude. For most WECs, this is not a valid argument, and much larger motion amplitudes are accepted compared to the Oil & Gas sector, giving lower forces on the WECs. For the large structures considered in this study, the assumptions of low responses might be valid in operational conditions, but for the extreme events, a more advanced dynamic analysis is needed.

As the mooring design procedure is an iterative process, cf. Fig. 4, and can include many simulations, an initial quasi-static analysis can be a useful tool in the design process. With some uncertainty, the analysis will provide an estimate of the expected excursion, tension in the lines and an estimate on the system's footprint on the seabed as result of the chosen system layout. These results are helpful in deciding relevant layout and dimensions for a full dynamic analysis.

In standards like API-RP-2SK and IEC-62600-10, a dynamic analysis is required for the final mooring system, but both state that quasi-static analysis can be considered for preliminary analysis. DNV-OS-E301 states that quasi-static analysis can be used in most cases, except for deep water conditions.

In the quasi-static procedure, described by e.g. [16] and [14], the environmental loads are accounted for by statically offsetting the floating WEC. The analysis takes into account the displacement of the connection point between the WEC and mooring lines, the weight and buoyancy of mooring system components and the elasticity of the lines. In the calculation, first the mean loads from wave, wind and current are applied to identify the mean horizontal offset, finding a mean position. To account for WF and LF motions, the equation of motion is solved, and the mooring stiffness in the mean position is mostly induced in the equations, either non-linear if using a time domain formulation or with linearised stiffness if using a frequency domain analysis.

In the quasi-static analysis, only one degree of freedom (DOF) is considered by means of horizontal surge motion, hence all vertical motion must be neglected. Furthermore, no dynamic effects associated with e.g. mass, damping and fluid motion are included in the analysis.

The outcome of the analysis is the tension in a point in the mooring line as a function of the relative horizontal excursion. Normally, the largest of the excursions calculated in Equation 4-5 is used for design and calculation of the characteristic line tension, [16].

$$X_{C1} = X_{mean} + X_{LF-max} + X_{WF-sig} \tag{4}$$

$$X_{C2} = X_{mean} + X_{LF-sig} + X_{WF-max} \tag{5}$$

Where X_{C1} and X_{C2} are the characteristic excursions, X_{mean} is the excursion caused by the static environmental loads, X_{LF-max} and X_{LF-sig} are respectively the maximum and significant excursion caused by the LF loads and X_{WF-max} and X_{WF-sig} are respectively the maximum and significant excursion caused by the WF loads.

Because of the definition of quasi-static analysis, this method is particularly appropriate for LF and mean drift motion.

D. Dynamic Analysis

In the final design, a dynamic analysis of the mooring for WEC is necessary, which includes the effects of mass, damping and fluid acceleration, and accounts for the drag and inertia forces on the mooring line components. Furthermore, the dynamic analysis includes motion in all six DOFs.

In the dynamic analysis, the motions of the connection point are imported into a dynamic cable solver, which calculates the dynamic forces in the cables. The dynamic part will be of paramount importance for the assessment of the maximum loads [13]. Examples of software packages, which can be used in dynamic mooring analysis, are:

ANSYS Aqwa [25]
FlexCom [26]
GMOOR32 [27]
MooDy [28]
MOSES [29]
Orcaflex [30]
ProteusDS [31]
SeaFEM [32]
Sesam DeepC [33]
WHOI Cable [34]

Naturally, each software packages provides a variety of different options and specifications.

A vital parameter for the software performance is the method used to model the mooring cables. Many different numerical models have been used, covering e.g. the finite difference method [35], used in WHOI Cable. More recent methods cover the Finite Element (FE) method [36], used by the remaining software in the list. A large amount of the software has implemented a Lumped Mass method, as treated in [37]. Many of the software are originally developed for flexible steel structures and the implementation of tension force is based on a small strain approximation. This means, that simulation of stiff materials like e.g. chains is very efficient, but for elastic materials, the simulation will become less accurate. Among the listed software, MooDy, MOSES and WHOI Cable have not implemented the small strain approximation and are therefore more aimed at analysis of mooring systems with synthetic materials, than the other software.

Most of the software provides the possibility to do analysis in both time and frequency domain, while a few only analyse in one domain. Considerations should be made when choosing analysis domain, since analysing in the frequency domain is faster than in the time domain, but requires linearisation of all non-linearities. In a mooring system, significant non-linearities are present, e.g. the stretching of the lines. Most often chains and steel wire ropes can be assumed linear, but for e.g. synthetic ropes, the material is highly non-linear. Another effect to consider is geometrical non-linearities, which occur in case of changes in the shape of the layout of the mooring lines. As the drag force on the lines is proportional to the square of the relative velocity (between line and fluid), this force will also be of a non-linear nature. Finally, the contact between line and seabed is a frictional process and therefore non-linear.

In a frequency domain analysis, all of these non-linearities must be linearised, to some extent affecting the results. A time domain analysis is therefore considered more accurate.

Dynamic analysis can be made as either coupled or de-coupled. In the de-coupled analysis, the motion of the floating WEC is calculated in an external software, with the mooring simulated as an applied stiffness, and the motion of the connection point is then imported into the cable solver.

For more sophisticated analysis, a software package capable of fully coupled analysis is used (e.g. Orcaflex, DeepC, etc.). Here, a global analysis is performed, calculating all interactions between the cables and WEC directly and simultaneously. Calculation of the floater motion is often based on Response Amplitude Operators (RAO) and Quadratic Transfer Functions (QTF), which for most software packages as e.g. Orcaflex and ProteusDS are input from external diffraction software such as WAMIT [38] or NEMOH [39]. WAMIT is capable of determining both first and second order loads, hence both RAOs and QTFs, while NEMOH at present only determines first order loads (RAOs). The diffraction analysis can also be performed in the software itself, e.g. a possibility in ANSYS Aqwa [25], MOSES [29] and Sesam DeepC [33].

When applying loads on the structure during simulation, it is relevant to look into how the software models the environmental loads in terms of profiles and spectra of wind, wave and current. All of the listed software allows vertical profiles and standard spectra, together with implementation of actual time series of wind, wave and current. In modelling of the waves, different wave theories are provided and for the extreme events in shallow water, which is especially vital for ULS and ALS, the waves become non-linear and need a wave theory of high order, e.g. the Stream Function or a higher order Stokes theory. Some software provides the possibility of Stream Function and fifth order Stokes theory, and some, e.g. ANSYS Aqwa, only Stokes theory up to second order. The relevant wave theory needs to be chosen dependent on the specific site and its conditions, using e.g. the ranges defined by Chakrabarti [40], which is also stated in DNV-RP-C205 [41]. In case of simulation of irregular waves, most of the software allows different standard spectra, which is modelled by superposition of regular, linear waves.

IV. Current Mooring Design in WEC Sector

With the large amount of mooring failures in the WEC sector, a need to optimize both the design process and layout seems present. A clarification of the current state in the wave energy sector is therefore necessary. For the present study, the mooring designs for four Danish WECs have been investigated, and will be outlined in the following.

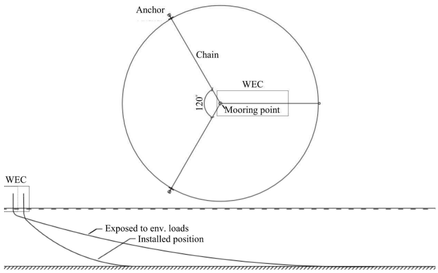

Fig. 6. Illustration of turret mooring system used by the WEC developers treated in Case 1. Top: Top view of layout. Bottom: Side view of a single line in the system, both in installed position and when exposed to environmental loads.

A. Case 1

The WEC considered in Case 1 is a devices taking advantage of the pitching body principle, and has in the mooring design, in great extent relied on existing offshore experience and practice. At present, the developer has constructed and carried out offshore tests with a demonstration plant for which a mooring solution was designed. A commercial scale size plant is undergoing development and a mooring system has been suggested.

1) Site Considerations: The design procedure was initiated with site considerations for the demonstration plant. Extreme events with a 100-year return period were described, and formed the basis for the design of the mooring. For the current mooring system for the commercial scale device, actual site considerations have not been done, and all environmental loads are based on upscaling and assumptions based on the loads determined from the smaller scale device.

2) Mooring Layout: In choosing the mooring layout for the device, offshore traditions have been adapted and a three-legged slack moored catenary turret system, which is anchored to the seabed with drag-embedded anchors, has been designed. The restoring force in this system is a result of the weight of the suspended length of the lines, and heavy chains have been chosen. The system is designed to provide the possibility of rotating 360° in order to face the incoming waves. As described in Section II, this type of mooring system is widely used in other offshore sectors.

3) Performance Analysis: The mooring system performance was analysed by a marine engineer, which in great extent used existing knowledge from the offshore sector and based the analysis on a static analysis of the system. The mooring for the demonstration plant was verified through a series of both offshore and scaled laboratory tests. The results were used as the basis for the commercial scale mooring design, as measured loads were used in dimensioning of chains by upscaling through an equipment number, EN, described in [42], which is a number used for vessels to determine dimensions of anchors and chains. The number is calculated from the dimensions of the structure and is used in table look-up to determine chain and anchor sizes. Layout of the mooring system by means of length of lines, etc. was based on the static analysis.

4) Verification: It appears that no design standards have been used for verification of the mooring design. Since 100-year extreme events were used, it is indicated that the system was designed in the ULS but not with implementation of any safety factors. For the commercial scale device, no verification of line strength was stated and the system was only designed by means of ensuring satisfactory weight of lines to ensure station keeping. Other limit states as ALS and FLS were not considered, and detailed analysis of all components in the system such as connections etc. have not yet been considered.

B. Case 2

The WEC studied in Case 2, harvests wave energy through the oscillating water column (OWC) principle. At present, the device faces small-scale offshore tests, for which a full mooring system has been developed. Considerations on a commercial scale device with mooring system have been made at present.

1) Site Considerations: Different locations have been considered for this device, dependent on size and status in the overall progress towards a fully commercial WEC. In general, an expectation towards wave conditions in commercial scale has been made and locations corresponding to smaller scale conditions have been chosen. When choosing design conditions the main focus has been put in extreme wave conditions with a return period of 50 year. Assumptions on wind conditions have been made, but currents have not been considered.

2) Mooring Layout: The mooring layout was designed in order to give a desired stiffness similar to a stiffness previously used in small-scale laboratory tests. In order to prevent heavy vertical loads on the WEC and to reduce the overall footprint by avoiding mooring lines on the seabed, a SALM system was chosen. The mooring system consists of a gravity anchor and a single mooring leg build up by steel rods connected by multiple universal joints, providing the possibility of the leg to deform. Two submerged buoys are attached to the leg, giving buoyancy to the system. The WEC is connected to the mooring point through four synthetic, nylon lines, connected to the WEC at four points to reduce loads on the structure. The restoring force in the system is a combination of the elasticity of the lines and buoyancy of the buoys. The connection at the mooring point allows free 360° rotation in order to face the incoming waves.

3) Performance Analysis: The performance of the mooring system has only been analysed through laboratory tests. Initial tests were performed with different mooring stiffness, in order to evaluate loads and motions when exposed to wave loads. The chosen stiffness afterwards formed the basis of the mooring layout, as different layout and dimensions were statically tested in dry conditions, in order to estimate

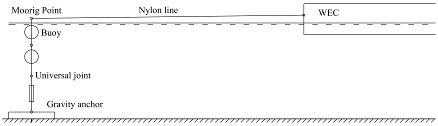

Fig. 7. Side view of SALM mooring system adapted by the WEC developer described in Case 2.

expected behaviour of the mooring system. At present, the device faces offshore, small-scale tests with the constructed mooring system. The test results will expectedly be used in the evaluation of the mooring performance and used in design of a larger scale device. For now, the commercial scale mooring design is a direct upscaled layout of the designed small-scale system.

4) Verification: The mooring system for this device has not been validated according to any design standards or guidelines. Despite that extreme events have been considered (50-year extremes and mainly for waves), actual considerations on limit states and corresponding safety factors have not been included in the design procedure yet. In the design, detailed analysis of all components, as e.g. connections, etc. have not been performed. Similarly, no restrains on allowable excursion have been stated.

C. Case 3

The WEC described in Case 3, takes advantage of overtopping in the wave energy absorption. The device has been undergoing a long series of tests, both offshore and in laboratory, and by now faces deployment in sea in a scale expected to be commercially satisfactory.

1) Site Considerations: For the design of the current mooring system, a specific site has been considered and a large amount of data on environmental conditions has been collected, mainly covering measurements of wave conditions. The wave data has been used in a statistical analysis to determine extreme events with a 100-year return period. Current and wind loads have been estimated based on experience and existing work.

2) Mooring Layout: The initial layout of the mooring system was intended to be a CALM system with mooring chains. The layout was however changed to consist of synthetic nylon lines. This was based on the fact that mooring with heavy chains did not seem economical in shallow water depth. The overall layout of the system consists of six lines equally spread 60° apart. As seen in Fig. 8 the mooring line comprises a suspended nylon line and a chain located at the seabed to avoid wearing of the nylon. All lines are connected to a surface piercing main buoy, to which the WEC is connected through a rigid element. The connection is constructed to allow 360° rotation, but installation of an aft anchor is considered, restraining the rotation to $\pm$ 60°, which keeps the WEC in place and avoids drifting over the mooring point.

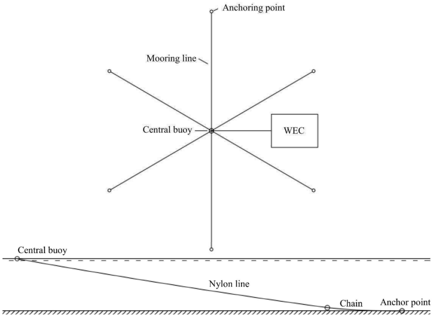

Fig. 8. Illustration of CALM system used in the mooring system for the device treated in Case 3. Top: Top view of system. Bottom: Side view of a single line.

3) Performance Analysis: The performance of the mooring system was evaluated through a full dynamic analysis using the software Mimosa [43], which is a frequency-domain analysis tool directly aimed at mooring analysis. Results from previous laboratory tests were used in validation and calibration of the model, which was set up to include the environmental loads, strengths and elasticity of lines, all linearised as Mimosa is a frequency domain tool. Statistical analysis was used to evaluate maximum and significant loads in mooring lines together with excursion of the WEC.

4) Verification: The mooring system for the present WEC was verified according to the design standard DNV-OS-E301. Both ULS and ALS were considered with corresponding safety factors (cf. Table I) for consequence class 1. All lines were verified according to the strength and resulted in determination of dimensions for included components. The analysis however only covered the lines and no considerations regarding buoy, anchors, connections etc. were included.

The excursion of the WEC was evaluated, but with no further concerns.

D. Case 4

The final case considered in this study, absorbs the wave energy by the OWC principle, and is by now in an early stage of the mooring design.

1) Site Considerations: By now, the developer of the device has considered a deployment site for a full-scale commercial device, and described the extreme events with a 100-year return period. The main concern for now has been wave conditions, while current and wind have not been thoroughly evaluated.

2) Mooring Layout: The considerations regarding the mooring layout are highly based on the naval sector. Existing mooring layouts for vessels have been investigated together with the mooring for other WECs. A slack moored system has been considered, but a turret mooring system was chosen. Instead of mooring chains, it is expected that the lines will be of synthetic material to reduce loads. At present, the overall layout has been adapted from a different WEC and has a similar layout as illustrated in Fig. 6, but using synthetic ropes instead of chains.

3) Performance Analysis: For now, a series of small-scale laboratory tests have been used as validation of the mooring system. The tests evaluated the mooring loads and excursions, and provided a necessary stiffness of the mooring system.

4) Verification: Verification according to design standards has not yet been performed, and only wave loads have been included in tests.

V. Conclusion

An investigation of the current state in the design of mooring solutions for large floating WECs in Denmark has been performed and outlined in the present study.

The investigated devices intend to be deployed in shallow water depths, inducing challenges for the mooring systems. The purpose of the mooring system is to provide enough stiffness to avoid large excursion of the device, while at the same time giving enough compliance to limit the mooring loads.

Design standards and design procedures stated by several authors were briefly outlined, focussing on the existing standards, which are mainly aimed at the Oil & Gas and naval sector. Standards for the wave energy sector are by now undergoing work. The importance of site consideration and its restrictions was stated. It was mentioned that quasi-static analysis seems appropriate and useful for preliminary analysis, but a full dynamic analysis including all dynamic effects and ideally all non-linearities, needs to be used in final design. Tools, and considerations to do when choosing one, were described, illustrating influence of a range of parameters. Choice of analysis domain highly influences computational time, but also inclusion of non-linearities and thereby accuracy of the model. The choice of wave load theory should be based on the given site and conditions and the material in the mooring system can be determining for the choice of the dynamic cable solver.

The current state and design procedure of four existing WECs, classified as being large floating structures with passive mooring, were investigated and compared to the described standards and design guidelines. A range of different challenges in the sector seems present.

General for all devices is the use of experimental tests to form the basis of the design. However, the investigated developers have not used any standard design procedure. In most cases, offshore experience has been used in choice of layout and in some extent in the dimensioning of components, by means of a quasi-static analysis. This is done despite the fact that dynamic analysis seems necessary.

A difference in use of environmental condition is present in most cases. Return periods for extreme events are varying, which must be expected to influence the mooring design. Similarly, it seems clear that most attention, and in some cases all attention, is aimed at wave loads, and influence of wind and current effects are neglected or given low priority. This is illustrated in the level of details in considerations on the different deployment sites, where also restrictions regarding the site, covering e.g. allowable excursion, is not accounted for.

Another fact is that only the main component in the mooring system is analysed and designed. A fully designed system has therefore not been achieved in any of the cases.

Stated in some design standards and previously by other authors, WEC mooring systems need to be analysed in a fully dynamic analysis including safety factors according to the limit states. Except for one case, this has not been done, and must be considered a vital task in the progress towards achieving a sufficient mooring system.

The four investigated cases are assumed to illustrate the current state in mooring design for large floating WECs in the Danish wave energy sector, showing a clear need for further investigation. All developers have ideas on layout, materials, etc. but lack detailed description and dimensioning of the system, before it can be considered a complete design. As a result of the choice of different materials and layout it seems reasonable to investigate further what mooring design should be applied to these kind of WECs and if the same solution can be used for all. A need of thinking the entire process into the procedure, including buildability and large-scale production, could be highly beneficial for the mooring and probably reduce the overall cost of the device. This is considered a vital task in the WEC sector, where mooring for now takes up a large amount of the total cost.

ACKNOWLEDGMENT

The work on the present study was funded by the Energy Technology Development and Demonstration Program (EUDP) through the project "Mooring Solution for Large Wave Energy Converters". The authors further wish to acknowledge the WEC developers Floating Power Plant, LEANCON Wave Energy, Wave Dragon and KNSwing for provision of information needed in the study.

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Paper B

Initial Assessment of Mooring Solutions for Floating Wave Energy Converters

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Stephen Banfield

The paper has been published in
ISOPE, ISBN 978-1-880653-88-3, 2016.

Proceedings of the Twenty-sixth (2016) International Ocean and Polar Engineering Conference **www.isope.org**
Rhodes, Greece, June 26-July 1, 2016

ISBN 978-1-880653-88-3; ISSN 1098-6189

Initial Assessment of Mooring Solutions for Floating Wave Energy Converters

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ABSTRACT

The present study investigates three different types of mooring systems in order to establish potential cost reductions and applicability to wave energy converters (WECs). Proposed mooring systems for three existing WECs create the basis for this study, and the study highlights areas of interest using a preliminary cost estimation and discussion of buildability issues. Using synthetic rope and variations in the mooring configuration has the potential of influencing the cost significantly. In order to quantify this potential, a simple quasi-static analysis is performed, which shows that a SALM type system can provide a paramount cost reduction compared to a traditional CALM type system with chain lines. Similarly, use of nylon ropes similarly appears to provide low cost.

KEY WORDS: Mooring; Station-keeping; Wave Energy; Synthetic; Chain; Quasi-static; Buildability

INTRODUCTION

Wave energy converters (WECs) have been the focus of much research and investigation throughout the last decades leading to a number of different concepts and devices. Some of these devices are floating structures with a need for systems that keep the structures on station and secure a minimal effect of environmental loads. Often a mooring system is applied which, by definition, is a system of lines connecting the floating structure to the seabed.

The layout of the mooring system and the applied components can vary significantly, give much different characteristics to the system and result in important differences in the cost. In moorings for WECs, the influence on energy absorption is also considered, and the systems are therefore characterized as either passive, active or reactive. The latter defines a system where the mooring provides reactive forces in the WEC and thereby influences the power take-off (PTO) while the passive and active systems respectively define a system with no influence on the energy absorption and a system where the mooring influences the dynamic response and energy absorption (Martinelli et al., 2012).

Mooring is a vital part of all floating structures and is a well-known concept in the oil, gas and naval sectors with a variety of different design standards, as e.g. DNV (2010), API (2005) and ISO (2013). The wave energy sector has adapted the experience from these sectors to a large extent, but still a large number of failures have been observed due to insufficient moorings as stated by Martinelli et al. (2012). In addition, the cost of moorings represents a large part of the total structure cost, estimated to be in the range of 10-30% (Carbon Thrust, 2011; Fitzgerald, 2009; Martinelli et al., 2012). In comparison, the same articles estimate the cost of station keeping systems in the oil and gas sector to be approximately 2% of the total structure cost, which is also more easily covered by the available finances, and consequently the encouragement towards further optimization is not as distinct. Additionally, Carbon Thrust (2011) states that there is only little potential of cost reduction in the existing types of mooring system and highlights innovation and use of alternative materials as potential approaches towards cost reduction.

This study investigates the potential of a number of mooring solution candidates to be applied on an existing WEC. This device is characterized as being a large floating structure with a passive mooring system and represents devices such as Floating Power Plant, LEANCON Wave Energy, KNSwing and Wave Dragon, cf. Fig. 1.

Figure 1: Large floating WECs with passive mooring systems. Top left is *Floating Power Plant* (2015), top right is *LEANCON Wave Energy* (2015), bottom left is *Wave Dragon* (2015) and bottom right is KNSwing.

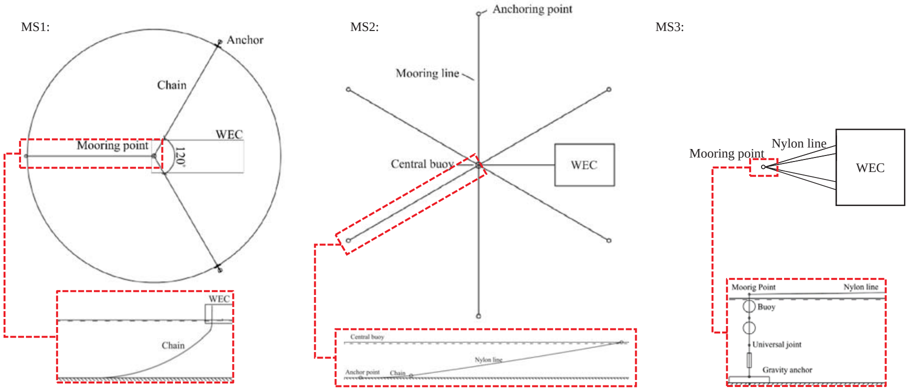

Figure 2: Conceptual drawings of the current preliminary mooring systems investigated in the present study. Adapted from Thomsen et al. (2015)

The study takes its basis in the existing mooring solutions for three devices described in Thomsen et al. (2015) and investigates the cost and buildability of each in order to highlight potential areas for cost and usability optimization.

The investigation will be used to establish three mooring solution candidates for the present device, which will be coarsely designed using a simple quasi-static method. The quantity of mooring lines in each solution, and hence also the cost, will be used to discuss which mooring solution that should be used in future detailed investigation and design.

CURRENT MOORING DESIGN

Thomsen et al. (2015) investigated mooring systems applied to different WECs from the Danish wave energy sector. The current state of these systems was found to be insufficient and in need of further investigation and design. As a starting point, these systems are used to describe potential mooring candidates and the requirements from different developers. A short introduction to each system, denoted MS1-MS2, is found in the following and is illustrated in Fig. 2.

Mooring System MS1

Mooring system MS1 (cf. Fig. 2) is planned for deployment at a location near the Belgian Coast and is quite similar to what is seen in the naval and oil and gas sector. The system is required to allow 360° weathervane in order to face incoming waves and is, therefore, based on a turret mooring concept with three slack chain catenary legs. The system is moored using drag embedded anchors.

In order to ease installation and maintenance, the system is desired to be disconnectable.

Mooring System MS2

System MS2 is a single point mooring (SPM) system, consisting of a surface piercing buoy and six lines, composed of partly chain and nylon rope, which are anchored to the seabed by drag embedded anchors. A nylon rope connects the device to the buoy. The system allows for weathervaning, but an aft anchor restrains the rotation to ± 60° according to the installed position. This aft anchor is not illustrated in Fig. 2.

Mooring System MS3

The final mooring solution is of the single anchor leg mooring (SALM) type and consists of a single mooring leg, two submerged buoys and four nylon ropes. The system is allowed to rotate freely according to the incoming waves and a restrain need to be applied in order to ensure that the device does not drift over the mooring point in calm weather. The system is anchored using a gravity anchor, here in form of a concrete block.

BUILDABILITY CONSIDERATIONS FOR MOORING SOLUTIONS

Each of the listed mooring solutions takes its basis in existing knowledge and experience with mooring of other offshore structures, particularly the oil and gas sector. It is clear that the requirements to displacements in these sectors are highly restrained while for WECs much larger motions can be tolerated, which secure smaller loads on the structure. This puts high demands on the compliance of the mooring system, which arises from either geometrical changes in the system and the weight/buoyancy of the components, or from the elasticity of the lines.

For MS1, the compliance of this system arises primarily from the weight of the lines together with the geometrical configuration and its changes when exposed to environmental loads. Often a much heavier system is achieved which puts more demands on the installation equipment and leads to an increase in the cost of the system. MS1 is intended to be a disconnectable turret system with 360° weathervaning, which highly increases the cost since sophisticated engineering and manufacturing are required. It should be noted that in order to easily disconnect the mooring, the turret should be located at the tip of the device, which highly influences the response of the WEC in e.g. the pitch degree of freedom. In order to ensure disconnectablity, the mooring lines will be attached to a separate buoy which can be winched up, mated and locked into the bearing part of the turret built into the WEC hull. This clearly simplifies the hook-up installation and allows for periodical disconnection for inshore maintenance or repair. However, the installation of the mooring on site still needs to be considered, and the size and weight/buoyancy of the disconnectable part must be designed in a way that makes disconnection possible, secures station-keeping and survival in extreme conditions and also ensures that the WEC can be reconnected afterwards.

For a catenary system like MS1, many different types of anchors can be used e.g. hammer piles, suction piles, plate anchors or gravity anchors, each with advantages and drawbacks, but with the lowest cost option being drag embedded anchors. This type of anchor primarily provides a horizontal strength. Many requirements to the installation of anchors are present, and the choice of type is highly dependent on the site and its seabed condition. At some sites it might not be possible to install e.g. drag embedded anchors.

MS2 is classified as a single point mooring (SPM) consisting of a number of synthetic lines connected to a single buoy with a rotating turntable that enables 360° weathervaning. Similar to the turret, this increases the difficulty of the design and manufacturing process. If an aft anchor is applied or the load direction is restricted to one direction with small spreading, the rotation can be accommodated by the lines and the turntable can be avoided. In MS1, the lines are synthetic and taut; hence, the compliance of the system is mainly resulting from the elasticity of the lines. Using synthetic line instead of chain gives some advantages and drawbacks which will be described in the succeeding text. The type of anchors can be chosen similar to the MS1 system, but it must be ensured that some vertical strength is available also.

MS3 is of the SALM type system and can be considered to be readily disconnectable, which allows for easy installation hook-up and disconnection for inshore maintenance or repair. If the system is equipped with a turntable, it will have the possibility to weathervane, which results in the same issues as described for the two other systems. An issue for a SALM system is the considerable vertical loads on the anchor in addition to horizontal loads. Selecting, designing and installing an anchor that can handle this loading in an effective way can be difficult and is highly dependent on the seabed condition at the given site. In order to have e.g. a gravity type anchor with sufficient mass, often a volume that is unrealistic to manufacture, transport and install is required. In addition, it might decrease the available water depth and possible length of lines in an insufficient way.

Use of synthetic rope instead of traditional chain induces a much different system. A wide variety of synthetic fibre can be used e.g. polyester, nylon or high modulus polyethylene (HMPE) (Weller et al., 2015). The most apparent difference between a line of steel and synthetic fibre is the weight and stress-strain behaviour. Clearly, using synthetic rope results in a mooring system with a much lower total weight than for a steel system, which therefore eases the transportation and installation significantly and minimizes the effect from the line weight on the WEC.

The stress-strain relationship of chains and synthetic ropes differs significantly and has different requirements throughout the system's life time. In order to avoid snap loads and high peak loads, a more compliant system is beneficial. Pecher et al. (2014) investigated a CALM type system and found that the stiffness of the system increased with decreasing water depth, illustrating the disadvantage of using chains in shallow water depths. Many WECs are planned for deployment close to shore in order to ease maintenance and inspection; hence, it can be expected that installation in shallow water depths will be desired. For a CALM system, the compliance is resulting from the weight and geometry of the lines while the compliance of a taut system primarily is a combined effect of the length of lines, L, their cross-sectional area, A and the elasticity of the lines E. As an approximation, Ridge et al. (2010) states that by doubling the compliance of a system, the cyclic loading and peaks in both lines and anchors will be reduced by half which therefore gives more incentive to use compliant synthetic ropes. This is partly caused by the synthetic lines' ability to absorb and dissipate energy, cf. Weller et al. (2015), which is a clear advantage when dealing with snap loads. As a result, the stress-strain relationship for synthetic ropes is much more complex than the relationship for chains that can be approximated by a linear trend. In the design and installation process, it needs to be taken into account that synthetic lines creep under a constant load as illustrated in Fig. 3. When the load is reduced, the line will recover but with some permanent extension (cf. Fig. 3). This is especially seen in newly manufactures ropes due to the response of the material and rearrangement of the fibres in the rope. It is vital to include this behaviour in the design in order ensure that e.g. the pretension applied at installation results in sufficient values over time.

When choosing different types of rope, much different characteristics of the line can be obtained. At present, a considerable amount of research and experience on polyester is available (Davies et al., 2014; Flory et al., 2015; Banfield and Flory, 2010; Flory et al., 2007; Banfield et al., 2005; Leech et al., 2003; Banfield and Casey, 1998), but in shallow water depths even polyester might be too stiff, and as a more compliant alternative, nylon can be used. It should be noted that the structure of a rope impacts its compliance and fatigue performance. Much research has been done on braided lines, but Weller et al. (2015) and Ridge et al. (2010) both highlight the parallel type ropes as the most durable type and most appropriate for long-term moorings.

The higher compliance of nylo, causes lower peak loads as previously described, and the fatigue performance is very close to polyester due to improvements in rope and design and marine grade lubricants. Therefore it also appears that synthetic ropes are sufficient for 20-30 year application, which in most cases corresponds to the lifetime of a WEC. In addition, it can be noted that synthetic ropes generally provide a better tension fatigue performance than e.g. chains and steel wire ropes in water; a potential result of no corrosion problems. Tests with polyester moorings after 12 years deployment showed 96% remaining strength (Ridge et al., 2010).

It is necessary to consider that some environmental aspects affect the strength, endurance and performance of synthetic lines. While polyester e.g. does not absorb water, nylon is more vulnerable and loses approximately 10% of the strength when wetted. The long-term effect of these aspects is vital to consider and include in the design process.

Using synthetic ropes causes buildability problems with handling and protecting the ropes throughout transportation, installation and under operation. Naturally, both nylon and polyester are more vulnerable to physical damage than e.g. steel chain and need to be transported and installed with caution. Similarly, during operation it might be necessary with protective measures as e.g. applying jacketing. To keep particles out of

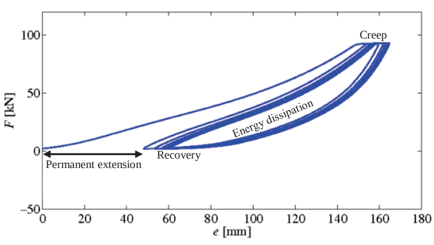

Figure 3: Stress-strain relationship for a synthetic rope under cyclic loading The figure is adapted from Weller et al. (2015).

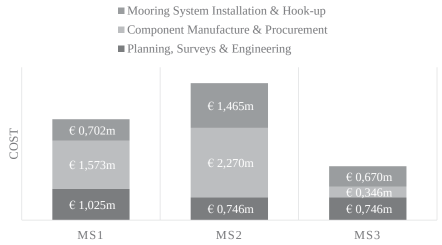

Figure 4: Indications of CAPEX for the three systems with cost basis in 2014.

the load bearing core a filter membrane is fitted. These were researched and designed by Banfield et al. (1999) for the oil and gas industry. The vulnerability also puts high demands to the connections in the mooring system by means of their materials and configuration.

In order to ensure the performance of lines, it is necessary to perform periodic control of lines, independent on whether it is steel or synthetic materials, but much dependent on the site and environmental conditions. This also allows for inspection of marine growth.

COST ESTIMATION OF CURRENT MOORING SOLUTIONS

Thomsen et al. (2015) investigated the state of the mooring systems MS1-MS3, and found that the designs are not yet final and more parameters need to be included. This means that the systems at this stage are not designed according to any standards, not all environmental loadings are included, and it is therefore not certain that the mooring systems are capable of surviving in the sea. However, preliminary cost estimation was performed, giving indications on the differences in the cost of each mooring system. It should be noted that the systems were designed for different WECs and load conditions, which influence the price. Fig. 4 illustrates the CAPEX of each system. The cost estimation was estimated based on experience from other projects, and indicative cost of components with basis in 2014 prices (cf. Table 1).

With a total cost of €3,300,000 for MS1, €4,481,000 for MS2 and €1,762,000 for MS3 the difference in cost is significant. As seen in Fig. 4, the main reason for the low price of MS3 is the manufacturing and procurement of components. By consisting of only four nylon ropes, the price of lines becomes low compared to the other two systems. Even though the cost of synthetic materials per tonnes is much higher than e.g. chains, much less synthetic material is in general used compared to chain material. MS1 consists of mooring chains combined with heave anchors, which together gives a significant contribution to the cost. The high price of MS2 mainly results from the chain part of the six lines together with the very long aft chain. The seven anchors similarly increase the cost significantly. For MS1, it is also notable that the system needs a turret that allows 360° rotation, increasing not only the price of the components, but also the design and engineering process. When observing the cost of installation, it is seen that installing seven anchors in MS2 results in the highest cost, which is due to the installation of seven anchors.

From Fig. 4 it appears that most cost reduction can be obtained by applying a SALM type system (MS2) and potentially also by using synthetic materials. Since the systems have been designed for different locations and using much different approaches, the following sections define simplified mooring systems based on MS1-MS3, which will be designed for a single WEC. By using the same procedure for all systems, it should be more reasonable to compare the quantities of materials.

MOORING SOLUTION CANDIDATES

Considering the buildability and cost of the investigated systems, different parameters appear to be relevant to consider for optimizing the cost of moorings. This does not only concern the components and manufacturing, but also installation, maintenance etc. The following two parameters appear to have positive effect on the cost of the mooring system and will be investigated further.

- Applying a SALM system.
- Using synthetic ropes instead of chains.

In order to compare the influence of these parameters, three mooring candidates are designed for the investigated WEC, which is a large floating structure that absorbs wave energy through the principle of pitching bodies. The performance of the device is sensitive to the wave direction, and the developer therefore has a desire towards a disconnectable turret system that allows 360° rotation. The device is planned to be deployed at a location with extreme characteristics as defined in Table 2. The characteristics are based on requirements in DNV (2010), hence 100-year wind and wave conditions together with 10-year current velocity are used. In this study, it is assumed that wind, current and waves are acting in the same direction. However, in a detailed investigation it is necessary to consider a situation where the wind, wave and current are not aligned. The three candidates are noted C1-C3 and cover a traditional slack chain mooring system (C1), a taut SPM system with nylon ropes (C2) and a SALM system (C3). Despite the desire for a turret system, C3 is composed with a hawser of nylon and a tether of chain, and is therefore not considered as a turret. The systems are illustrated in Fig. 5.

In the present study, anchors, connections etc. are neglected and only the quantity of mooring line material is considered.

QUASI-STATIC DESIGN OF CANDIDATES

A simple quasi-static method is used for designing mooring lines and buoys.

The quasi-static method has been described by several authors, such as Pecher et al. (2014), Johanning et al. (2005) and Bergdahl and Kofoed (2015), and has been used widely in other offshore sectors such as the oil and gas sector, and mentioned in several design standards like e.g. (DNV, 2010). As previously described, the requirements to displacements in these sectors are stricter, and in the wave energy sector a higher response can be tolerated. In addition, e.g. an oil platform has a much higher mass than a WEC and therefore also induces a much lower response and velocity, justifying the use of quasi-static calculations.

In the quasi-static procedure, initially the displacement induced by the mean wave, wind and current load is calculated and the linearized stiffness of the mooring system in that position is determined. Using e.g. a boundary element method (BEM) code such as Nemoh (LHEEA, 2014), the hydrodynamic properties of the WEC are calculated and the response

Table 1: Indicative cost data for mooring components based on 2014 prices.

Material/Component	€ / tonne
Forged Steel Mooring Chain	4,140
Wire Rope - Spiral Strain	8,280
Synthetic Rope (Polyester/Nylon)	16,100
Drag Embedment Anchor	7,038
Concrete Gravity Anchor	230

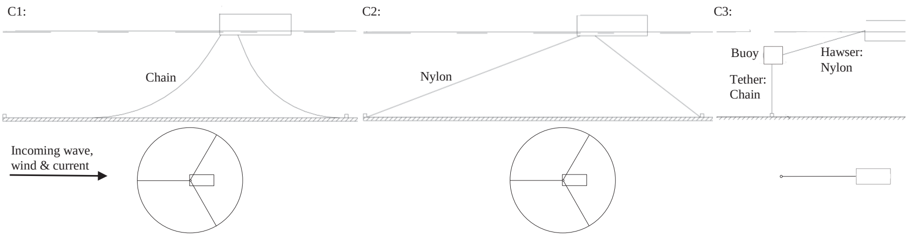

Figure 5: Conceptual drawings of mooring candidates C1-C3, used in the quasi-static analysis.

around the mean position is stated, thereby taking into account some of the dynamic behaviour and the linear effect of the mooring. In the calculation procedure, the WEC is given a static offset and the corresponding tensions in the mooring lines are calculated. As a result, all dynamic affects associated with mass and fluid motions are neglected and often only one DOF (surge) is considered. In the present study, a frequency domain solution has been used to determine the hydrodynamic response, taking into account current and wind together with first and second order wave load effects. In general, significant non-linearities are present in mooring systems, and especially synthetic ropes have a non-linear behaviour. In order to use a frequency domain solution, all of these non-linearities are linearized. This procedure follows the method described in Bergdahl and Kofoed (2015) and satisfies the requirements to the ultimate limit state (ULS) and the corresponding safety factors as defined in DNV (2010). Despite that the safety in this standard has been suggested to be relaxed in several papers as Paredes et al. (2013) and Johanning et al. (2005), this standard is still widely used and therefore also considered in this paper, thereby making the results more conservative.

Clearly, the use of a quasi-static approach is not recommended for the dynamic behaviour of WEC and moorings, and the systems stated in this paper need a detailed dynamic analysis in order to achieve a full description of the mooring system characteristics. For this preliminary analysis, performed in order to understand the applicability of different layout and material, a quasi-static approach is considered to be sufficient. The following section describes the results from the analysis and can be seen in Table 3.

RESULTS AND DISCUSSION

Based on the quasi-static analysis, mooring line dimensions and quantities were determined for the mooring candidates C1-C3. The possibility of optimization by changing e.g. spreading of line was not investigated. The results from the analysis are presented in Table 3 and illustrate a significant difference in the required materials for each candidates.

Table 2: Site characteristics used in the quasi-static analysis. The values are based on wind and wave events with 100-year return period and current with a 10-year return period.

Parameter	Site 1
Significant wave height, H_s	6.55 m
Peak wave period, T_p	10.25 s
Water depth, h	30.00 m
Current velocity, v_c	1.20 m/s
Wind velovity, v_w	33.00 m/s
Wave Spectrum	JONSWAP (γ = 3.3)

As a result of the relatively shallow water depth, designing a catenary system causes severe problems. As stated by Pecher et al. (2014) the horizontal mooring stiffness increases with decreasing water depth, resulting in significant loads and tensions and therefore also a need for chains with a high strength. In order to achieve this, chains with larger dimensions must be applied, leading to larger weight and also stiffness. Decreasing the chain diameter decreases the stiffness but results in a lack of strength. For the given case it was not possible to achieve a sufficient system by only adjusting the chain diameter. In order to achieve a reasonable stiffness and a chain that can withstand the tension, a very long line is required. For the given case a need of 1160 m chain in each leg is required, which might be considered unrealistically long in 30 m water depth, and especially if the device is desired to be installed in an array. Also, since the mooring lines are very long and the excursion is small in comparison, only a very little part of the chain is suspended and most lines are always located at the seabed. Problems occurring with the lines being damaged and buried in the seabed must be considered and will result in more maintenance. A solution to bringing more compliance into the system could be to disregard the turret and apply a buoy and synthetic hawser. This depends on the developer's requirements, but might optimize the system.

C2 benefits from achieving much compliance from the elastic nylon lines clearly illustrated by the difference in horizontal stiffness. The low stiffness results in a low natural frequency of the system, much lower than the storm frequency, inducing a more desirable response of the WEC. If the higher displacement can be accepted, it is much easier to achieve a system that fulfils the requirements to strength, and significantly less materials than in C1 is needed. This also leads to a more reasonable seabed footprint, and a paramount difference in cost of the lines. For both C1 and C2, the disconnectablity has not been considered. If this should be achieved, a buoy should be applied to the system, providing enough buoyancy to lift the lines and allow connection. In the quasi-static analysis, it is assumed that the connection points between the lines and buoy do not experience any vertical motions; hence, the buoy does not affect the response. The buoy is therefore not considered an active part of the system for now and is not designed.

C3 is the mooring candidates which require the least mooring line material. By applying this system, high compliance is easily achieved and is a result of not only the elasticity of the lines but also the buoyancy of the submerged buoy. The overall geometry of lines and buoy highly affects the stiffness and response of the system, and much potential for optimization is present. For this preliminary study, standard guidelines have been followed. C3 was chosen to consist of a chain tether in order to secure high strength which allowed for the use of a very buoyant buoy while the hawser consists of elastic nylon. Applying nylon material in

Table 3: Results from the quasi-static analysis of C1-C3 (cf. Fig. 5)

	C1	C2	C3
Max displacement	17.7 m	33.5 m	17.8 m
Max tension	7261 kN	3065 kN	7505 kN
Linear horizontal stiffness	348 kN/m	51 kN/m	163.7 kN/m
Cable length	Ø80 R5 studless chain: 3 × 1160 m	Ø120 Nylon: 3 × 267 m	Ø92 R4 studless chain: 22.2 m Ø184 Nylon: 30 m
Seabed footprint radius	1149.6 m	268.3 m	-
Cable Mass	Chain (128 kg/m): 445.44 t	Nylon (8.17 kg/m): 6.544 t	Chain (169.3 kg/m): 3.758 t Nylon (16.2 kg/m): 0.486 t
Cable cost	Chain: € 1,844,122 Total: € 1,844,122	Nylon: € 105,358 Total: € 105,358	Chain:€ 15,558 Nylon: € 7,824 Total: € 23,382
Buoy mass	-	500 t -	41 t (340 t buoyancy)
Total mass	445.44 t	6.544 t t	45.244 t

both tether and hawser might benefit the system and provide additional compliance.

When considering only the mooring line material in Table 3, C3 appears to be the lightest and, therefore, also the system with the lowest cost. It should be noted that C3 requires a large buoy, which brings the total mass up to 45 t, which is higher than C2 but still much smaller than C1. Additionally, it is paramount to consider that no anchors have yet been designed. As stated, drag embedded anchor is often the solution with lowest cost and would preferably be applied to C1 if allowed by the seabed conditions. C2 and C3 need anchors that provide both horizontal and vertical strength and might be more expensive, than the anchors for C1.

CONCLUSION

Different types of mooring systems have been considered for use in WECs, based on traditions in other offshore sectors, but with increasing focus on new types and materials. Three systems suggested for use on large floating WEC were investigated and used as basis for investigation of potential cost reduction and buildabilty issues. Based on a preliminary and coarse cost estimation, it was found that use of a SALM system and synthetic line materials could optimize cost of the moorings.

Since large displacements often can be accepted for WECs, high compliance can be considered which easy arises from use of synthetic materials and a SALM system. Different buildability considerations were addressed, and parameters to consider when designing the different systems were highlighted.

By use of a quasi-static approach, mooring lines were designed for three systems, illustrating the difficulty of applying chains in moorings of WEC that are often deployed in shallow water depths. Paramount cost reduction was found in the SALM system and the SPM system with nylon lines. The calculations were, however, simplified, and many parameters still need to be considered in order to fully understand the potential. However, the results illustrated the potential of synthetic materials and use of SALM system, and will form the basis for future investigation and more sophisticated dynamic analysis.

ACKNOWLEDGEMENT

The work on the present study was funded by the Energy Technology Development and Demonstration Program (EUDP) through the project Mooring Solution for Large Wave Energy Converters.

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Paper C

Experimental Testing of Moorings for Large Floating Wave Energy Converters

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The paper has been published in
RENEW, ISBN 978-1-138-62627-0, 2016.

Progress in Renewable Energies Offshore – Guedes Soares (Ed.)
© 2016 Taylor & Francis Group, London, ISBN 978-1-138-62627-0

Experimental testing of moorings for large floating wave energy converters

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ABSTRACT: This paper presents the outcome of a test campaign, which investigates the behaviour of a synthetic mooring system applied to the Floating Power Plant wave energy converter. The study investigates the motion and tension response under operational and extreme sea states expected at the deployment site and studies the influence of variations of mass moment of inertia and mooring line stiffness. A simple quasi-static analysis based on existing guidelines was performed in order to compare results and validate the method. In most cases, the calculations proved to underestimate the tensions if compared to the measured tensions.

1 INTRODUCTION

During the last decades, the demand for renewable energy has increased significantly and resulted in a focus on wave energy. Today, a considerable amount of different concepts and devices exist (cf. e.g. Harris et al., 2004, Martinelli et al., 2012 and Gao and Moan, 2009) but despite comprehensive research, the wave energy sector is still only at a pre-commercial stage, with significant areas that require further development. One of these areas is the mooring system, since by now it has led to failure of several Wave Energy Converters (WECs) and, furthermore, it is so costly that it is estimated to take up to 30% of the total CAPEX of a WEC (Martinelli et al., 2012, Fitzgerald, 2009). In Carbon Thrust (2011) optimization of mooring layout is mentioned as a potential to cost reduction.

When considering WECs, moorings are often categorized into three groups depending on their influence on the energy absorption (Martinelli et al., 2012). In this study, only a passive system is considered; hence, a mooring system that does not affect the energy absorption but only ensures station keeping and minimal loads on the structure.

In order to understand the behaviour of moorings and to improve the design procedure, there is a need to optimize the tools and mooring system layouts. Recent studies by e.g. Ridge et al. (2010), Weller et al. (2015) and Thomsen et al. (2016) concluded that use of synthetic mooring lines has a potential to significantly reduce cost of moorings, and the concept is well proven and investigated in studies as e.g. Banfield and Flory (2010), Banfield et al. (2005) and Casey (1998).

As an initial step towards a better understanding of mooring behaviour and its applicability to a given WEC, experimental work can be carried out, providing data on the response and illustrating in which direction further development should head. The experimental data is additionally crucial in validating the tools established to design the moorings.

This study investigates a structure similar to the Floating Power Plant (*Floating Power Plant*, 2015), which combines both wave and wind energy absorption into one device. The structure is combined of a floating platform which carries a wind turbine, and four floaters installed for harvesting of wave energy through the principle of pitching bodies. The Floating Power Plant WEC is undergoing comprehensive development and research and it has previously been deployed as a 37 m wide model (named P37) for real sea testing in the Danish sea as illustrated to the left in Fig. 1. As a result of the test campaign, the structure was undergoing changes (cf. Fig. 1 to the right), and the next step towards the full-scale, commercial 80 m wide P80 model, is the 60 m wide P60 model, which forms the basis of this study.

Figure 1. Left: The P37 prototype during sea testing in the Danish sea. Right: Conceptual drawing of the P60/P80 device considered in the present study (*Floating Power Plant*, 2015).

A two-month test campaign was carried out to obtain initial estimations of expected response in relevant sea conditions and to obtain data for subsequent validation of numerical tools for dynamic analysis. This paper provides a description of these tests and the initial analysis of results, and it is structured with a general description of the test set-up and programme, followed by a presentation of the obtained results. The research is concluded by comparing the results with data obtained from a simple quasi-static analysis, following the procedure defined in several design standards such as API (2005), DNV (2010) and ISO (2013).

2 OBJECTIVES OF EXPERIMENTAL TESTS

The present test campaign does not aim to find a final mooring solution for the investigated WEC and the mooring is, therefore, not optimized prior to the tests. The main purpose of this testing is to investigate which parameters affect the motion and load response of the structure, in order to provide a basis for future investigation. The study explores the effect from the given mooring systems on the natural frequencies and mooring line tensions during operational and extreme conditions. This will provide a guidance for future research areas on which more investigation should be focused. Additionally, the tests provide the possibility to validate numerical simulations and design procedures. In the present study, this is limited to a quasi-static approach, but will be extended to full dynamic simulations in future research.

3 EXPERIMENTAL SET-UP

The experiments for the present study were carried out in the wave basin at The Hydraulic and Coastal Engineering Laboratory, Aalborg University, Denmark over a period of two months from December 2015 to January 2016. The following subsections provide a general description of the test set-up, investigated models, mooring lines and conditions.

3.1 *Test programme and procedure*

The test campaign was conducted in the wave basin at Aalborg University, with a test set-up as illustrated in Fig. 2. The basin was equipped with a snake-type wave maker in one end, controlled by the software package AwaSys 7 (2016), and a passive absorber (gravel beach) in the other. The origin of the basin coordinate system was located in the waterline at the bow of the device as illustrated in Figs. 2 and 4.

During the tests, the generated surface elevation was measured using a total number of five resistant type wave gauges located in a 3D array approximately 1.5 m in front of the model. Prior to the tests, all sea states were generated without the model in place to measure the wave reflection in the basin. The measurements showed insignificant reflection in the y-axis. For the longest waves, the maximum reflection coefficient in the x-axis was found to be approximately 0.2 and, therefore, considered acceptable for these tests. Acquisition of surface elevation and mooring loads, together with reflection analysis of surface elevation was performed with the software package WaveLab 3 (2016).

Mooring loads were measured in each test using three *FUTEK LSB210 50/100 lb* load cells located at the anchors of the mooring lines and one *VETEK 30 kg IP68* at the connection point between lines and device. Motions were measured using the motion tracking system OptiTrack (2016), with four OptiTrack Flex 13 cameras and five reflective markers. The data was acquired using the Motive 1.9.0 software.

The test programme was structured to allow for measurements of mooring loads and motion response in operational and extreme wave conditions at the P60 deployment site. A total number of nine irregular sea states were simulated and 23 regular sea states for validation of the measured response. The irregular sea states were simulated with a JONSWAP spectrum with a peak enhancement factor $\gamma = 3.3$ in a water depth $h_d = 0.7$ m (45.2 m in full-scale). Fig. 3 shows the target sea states determined from available scatter diagrams and the measured sea states from the experiment, both presented in full-scale. In general, there is insignificant difference between the target and measured wave peak periods, and a maximum difference of 9% between the target and measured sig-

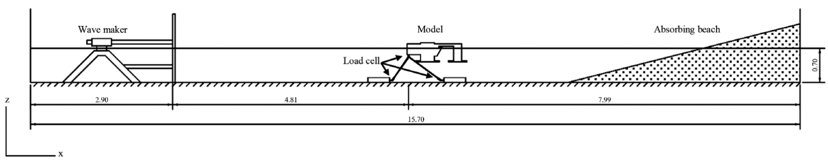

Figure 2. Illustration of used test set-up. All measures are given in [m].

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nificant wave heights. Since the results later in this paper is presented in frequency domain and the desired frequency range is still obtained, the difference in wave height is considered to be acceptable.

During the campaign, different mooring and model configurations were tested. In order to allow direct comparison of motion and load time series, the generated surface time series were stored and replayed for each configuration; hence, an exact copy of each time series could be generated for each configuration.

To determine natural frequencies of the system, allowing for comparison with the sea state frequencies, each model test set-up was initiated by a decay test. Similar, the stiffness of the mooring system was determined from a quasi-static test, where the model was displaced at low velocity in the surge DOF while the horizontal applied force was measured. During this movement, the only restoring force in the system was resulting from the mooring lines.

3.2 *Model*

The tested model represents a simplified geometry of the Floating Power Plant P60 WEC in scale 1:64.5. For these preliminary tests, only the floating platform was tested; hence, the floaters and wind turbine were discarded.

The model was constructed with a base structure, composed of high density foam (cf. Fig. 4) and additional mass was applied, which allowed for variation of the mass distribution over the structure. This was done in order to investigate the influence of the mass Moment of Inertia (MoI) on the mooring load and motion response, and for future investigation of the capability of a numerical code to simulate these variations.

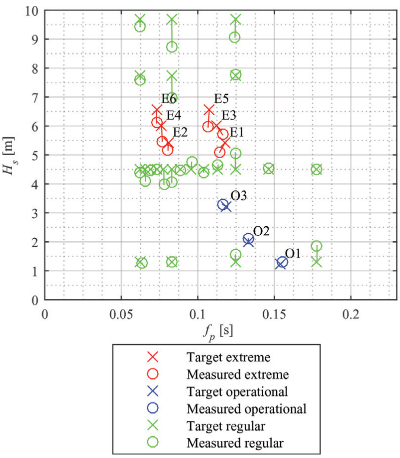

Figure 3. Comparison of target and measured wave conditions. Data is in full-scale (1:64.5).

In addition, two different types of mooring lines were tested. Considering Table 1, the configurations were classified as follows:

- C1 is considered the basis configuration.
- C2 investigates the influence of a lower MoI around the y-axis.
- C3 is used to investigate influence of a more compliant mooring line and the WEC characteristics are therefore similar to C1.

The centre of gravity and moment of inertia were measured in the tests and estimated from geometry and weight by use of a CAD software. As previously mentioned, the mooring layout and lines do not represent a final and optimal solution. Based on the existing mooring system applied to P60, the system was chosen as a turret system with three lines, spread equally apart (120°) as illustrated in Fig. 5.

Table 1. Definition of properties for the three model configurations used in the test campaign.

	C1	C2	C3
Centre of gravity (x,y,z) [m]	(32.9, 0.0, –4.8)	(35.5, 0.0, –4.5)	(32.9, 0.0, –4.8)
Moment of inertia I_{xx} [m² kg]	1.645×10^9	1.843×10^9	1.645×10^9
Moment of inertia I_{yy} [m² kg]	2.819×10^9	2.333×10^9	2.819×10^9
Moment of inertia I_{zz} [m² kg]	3.712×10^9	3.377×10^9	3.712×10^9

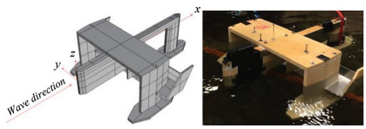

Figure 4. Left is notation of coordinate system and right is the constructed lab model.

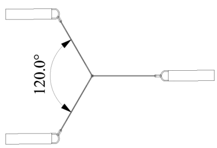

Figure 5. Illustration of mooring layout.

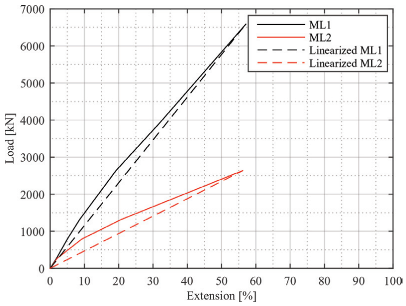

Figure 6. Measured force-extension curve for the mooring lines applied to experimental set-up. The dashed lines indicate the linearized stiffness.

Table 2. Definition of mooring line properties in full-scale values.

	Line 1	Line 2
Configuration	C1–C2	C3
Unstretched length [m]	46.12	45.47
Linearized stiffness [MN]	11.54	4.69

3.3 *Mooring lines*

The previous mooring system suggested for the Floating Power Plant, consisted of mooring chains, but as recent research has presented the benefits of using synthetic lines, the mooring lines in the experiments were constructed of elastic, non-linear material with a measured stiffness as shown in Fig. 6. Additional mooring line characteristics can be found in Table 2.

Despite the fact that the mooring lines do not resemble any actual mooring lines available, the linearized characteristics can resemble nylon and synthetic lines to some extent. The shape of the force-extension curve differs but using e.g. Ø80 mm and Ø120 mm *Bridon Superline Nylon* or Ø48 mm and Ø72 mm *Bridon Superline Polyester*, similar linear stiffness is obtained (Bridon, 2016).

4 TEST RESULTS

The following section presents the initial analysis of test results. The focus is on natural frequencies, response amplitude operators and extreme line tension. All results are presented in full scale (scale 1:64.5) by use of Froude scaling law.

4.1 *Static stiffness*

From the quasi-static tests, Fig. 7 presents the force-displacement curves for the two types of mooring lines.

During the model and line installation, it was attempted to acquire similar mooring line pretension in the two configurations. This was found to be approximately 0.45 MN. The horizontal linearized stiffness for C1 and C2 is approximately two times the stiffness of the system used in C3. The influence from this is seen in the natural frequencies found in the decay tests.

4.2 *Decay tests & natural frequencies*

The decay tests were performed in order to evaluate the natural frequencies in the surge, heave and pitch DOF for each of the configurations. The tests were conducted by applying a static displacement in one DOF, releasing the model and allowing the motion to decay naturally. An example of the decay response of C1 is illustrated in Fig. 8.

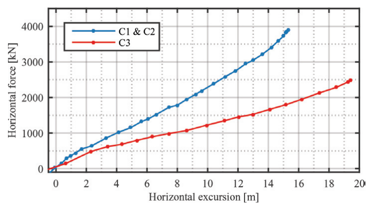

Figure 7. Measured static horisontal stiffness for the two types of mooring line.

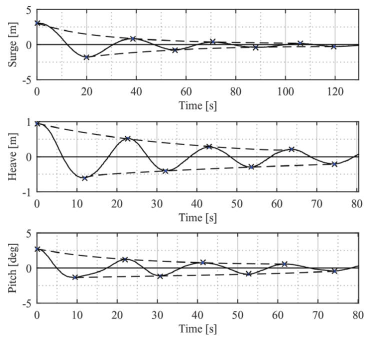

Figure 8. Example of decay reponse in surge, heave and pitch for configuration C1. Note the difference in the x-axis.

From the decay response, it is possible to determine natural frequencies and damping of the structure. The latter will not be considered in this study. Information on the natural frequencies is important in understanding the expected response of the device at a given site. By having natural frequencies close to the peak frequencies of the sea states, resonance can occur and results in an undesirable response.

Table 3 presents the results from the decay tests. Since the natural frequencies in heave and surge are defined by the stiffness and total mass, the results in these DOFs are identical for C1 and C2. As a result of the lower MoI around the y-axis in C2, the natural frequency in pitch is approximately 5% higher than for C1. In C3, the mass and MoI is identical to C1, but the different mooring stiffness, should affect all three DOF. The lower stiffness results in lower frequency in both surge and pitch, while an identical value is observed in heave. This is potentially a results of high hydrostatic stiffness compared to the mooring stiffness.

In the decay tests, oscillations with a frequency of 0.078 Hz for C1 and C2, and 0.075 Hz for C3 were additionally observed and resemble a cross-coupling between the DOFs.

4.3 *Motion response*

The motion of the device was analysed in the frequency domain and transformed into motion RAOs. The frequency domain analysis often implies use of a bandpass filter, as the wave energy above and below certain frequencies are insignificant, and, when calculating RAOs, results in much disturbance. Since the objective of the tests is to investigate the response in the relevant sea conditions, the main focus is put on the frequency band $f = 0.048$–0.23.

In Fig. 9 the calculated RAOs for C1 is plotted, and the minimum cut-off frequency is illustrated by the dashed black line. It is possible to detect a peak at 0.075–0.078 Hz, while the natural frequencies, as shown in Tab. 2, are below the cut-off frequency. The dark green dots indicate results from the regular sea states, and good agreement is found with the irregular sea states.

Table 3. Experimentally measured natural frequencies and frequencies determined from numerical results.

	Surge [Hz]		Heave [Hz]		Pitch [Hz]	
	Exp.	Num.	Exp.	Num.	Exp.	Num.
C1	0.029	0.032	0.048	0.052	0.047	0.042
C2	0.029	0.032	0.048	0.052	0.049	0.042
C3	0.022	0.022	0.048	0.056	0.046	0.052

Fig. 10 compares the RAOs for the three configurations. Observing the surge DOF indicates that a similar response can be expected for the three configurations in the investigated frequency band. In heave, the small peak observed, indicates that C1 and C2 approaches a similar response, both in terms of motion amplitude and peak frequency. C3 shows a higher amplitude at the peak, and the peak frequency is offset from 0.078 to 0.075, expectedly a results of the more compliant mooring system.

The same observation on the peak frequency can be made for the pitch DOF, but the amplitude shows larger variations. The higher MoI in C1 results in a lower motion response, while it is difficult to detect the difference in peak frequency. The C3 configuration shows an amplitude similar to C1 and C2 around its peak frequency, but is generally lower in rest of the frequencies. The stiffness applied in C1 and C2, therefore, seems to increase the pitch motion response.

4.4 *Line tension response*

Fig. 11 presents and compares the maximum measured line tension in the front starboard line for the three configurations. For C1 and C2, the tension increases from approximately 0.4–0.6 MN in the smallest sea state to 2.1 MN in the most extreme. In all sea states, C3 experiences the lowest tensions due to the low mooring stiffness. By considering sea state E3-E6 it is additionally seen that the peak wave period and, therefore, the wave steepness, has a large impact on the observed tensions. E3 and E4 have similar wave height, but different wave period; the same for E5 and E6.

Observing Fig. 12, which compares C2 and C3 with C1, it is seen that C2 provides larger tensions in the operational sea conditions and approaches the same value as C1 for the extremes. This corresponds well with the motion RAOs, since it was seen that C2 experiences a larger pitch motion amplitude in the investigated frequency band. For the extreme sea states with lower peak wave frequencies, the difference between the motion response of C1 and C2 decreases, and the measured maximum tensions approaches similar values.

The tension in C3 approaches the tension in C1 in the smallest waves, but decreases to 55% of C1 in the extremes; again, explained by the difference in motion RAOs, natural frequencies and the more compliant mooring system.

The same results are noted when observing the line tension RAOs in Fig. 13. The C1 and C2 configurations are close to identical, with C1 showing a slightly lower tension amplitude response, while C3 has much lower amplitudes. C1 and C2 have a RAO

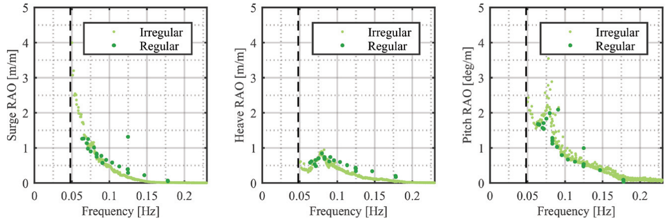

Figure 9. Motion RAOs for the surge, heave and pitch DOF for both regular and irregular sea states. The dashed line indicates 1/3 of the minimum encounter peak frequency. The results are representative for C1.

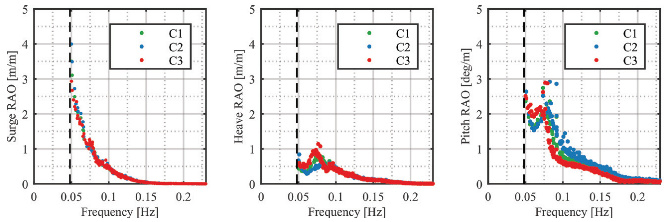

Figure 10. Comparison of RAOs for the three configurations.

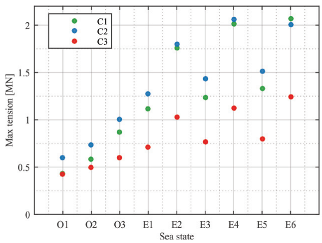

Figure 11. Measured maximum tension in each sea state. The maximum tension can be directly compared since the generated sea state for each configuration is identical.

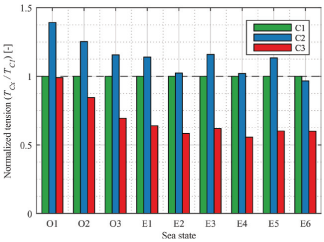

Figure 12. Comparison of tension in each configuration by means of normalized tension.

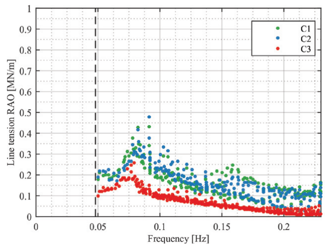

Figure 13. Comparison of line tension RAOs for the three configurations.

peak at 0.077 Hz, while C3 has a peak at 0.075 Hz. As expected, it is, therefore, seen that the tension RAO peaks at frequencies found in the decay tests.

5 QUASI-STATIC MODEL

In mooring design for large oil and gas platforms, a quasi-static approach is often used. By now, this has also been seen in the Danish wave energy sector when designing mooring system, cf. e.g. Thomsen et al. (2015). In a quasi-static approach, the device is statically offset, and the corresponding line tension is calculated. It is common to use a frequency domain approach where all non-linearities from geometrical change and the line material behaviour are linearized. In addition, all dynamic effects from fluid and masses are neglected, and only one DOF is considered (surge). This approach has been described in e.g. Bergdahl and Kofoed (2015), API (2005) and DNV (2010). Since the method has been used in design of moorings and since the objective of this study is to provide a better understanding of mooring design procedures, this method is applied to the experimental set-up, and the calculated tensions compared to the experiments, thereby illustrating whether this method provides representative results.

When estimating the design offset used to calculate the maximum line tension, DNV (2010) states that the larger of the following two excursions is to be used:

$$X_1 = X_{mean} + X_{LF-max} + X_{WF-sig} \tag{1}$$

$$X_2 = X_{mean} + X_{LF-sig} + X_{WF-max} \tag{2}$$

where X_1 and X_2 are the characteristic excursions, X_{mean} is the excursion caused by the static environmental loads, X_{LF-max} and X_{LF-sig} are respectively the maximum and significant excursion caused by the low frequency loads and X_{WF-max} and X_{WF-sig} are respectively the maximum and significant excursion caused by the wave frequency loads. When using this method, the waves are assumed to be Rayleigh distributed, and the calculated maximum peak tension, corresponds to the tension exceeded once in a thousand peaks.

The described procedure was used for comparison of numerical quasi-static and experimental results. Using the boundary element method code Nemoh (Babarit and Delhommeau, 2015), the hydrodynamic properties were calculated, the linear mooring stiffness in the mean position was introduced into the equation of motions and the significant and maximum excursions were found using statistical expressions. Based on this, line tensions were calculated and compared to the obtained experimental results, see Fig. 14.

In Table 3 the natural frequencies from the numerical analysis is compared to the measured values, indicating 8–16% difference, with the largest difference found in the heave and pitch DOF. These two DOFs are neglected in the quasi-static analysis, but the difference of 8–10% in the surge DOF indicates that the estimated mooring stiffness can result in a deviation from measured tensions. The mooring stiffness used in the equation of motion was found by projecting the stiffness from each line into the x-axis, calculate he excursion from mean loads and finally estimate the total stiffness in that position.

For the operational sea conditions, it is found that the numerical approach can overestimate

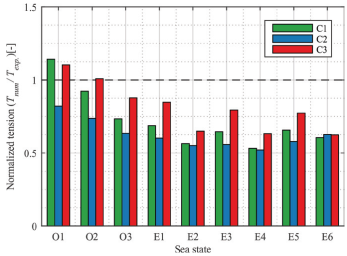

Figure 14. Comparison of experimental data and numerically determined tension from a quasi-static approach.

the tension. For larger sea states, the tensions are underestimated and correspond to 52% of the measured tension in the worst case. Since the quasi-static approach only considers surge, similar results are calculated for C1 and C2 despite the fact that the laboratory results showed variations. In all sea states, the calculated tensions in C2 are lower than the experimentally measured tensions; in the worst case it is 52% of the measured. This clearly indicates the influence of including all DOFs in the calculations. Previously, it was shown that the higher pitch natural frequency in C2 resulted in higher tensions, and by not including this in the quasi-static analysis, a significant underestimation is obtained.

Finally, it is seen that the largest difference between numerical and experimental results are obtained for the extreme sea states where a more severe response with more dynamic effects are expected. As an example, the larger displacements of the structure in extreme sea states can lead to slacking of lines followed by a considerable tension peak. By statically offsetting the structure in the quasi-static analysis, this effect is not taken into account.

6 DISCUSSION AND CONCLUSION

The present study presented and analysed data from a two-month test campaign on the mooring system of a large floating WEC, the Floating Power Plant P60. Different configurations were tested in order to investigate influence of different mass moment of inertias and mooring stiffness. By applying the chosen mooring solution, low natural frequencies were found. By comparing with operational and extreme sea states from the deployment site, it was found that these natural frequencies were outside the frequency band where wave energy is present, and the peak wave frequencies were much higher than the structure's. Due to the type of WEC, this is considered to be desirable.

The tests showed a tendency of decreasing the line tension in the case with the highest moments of inertia around the y-axis because the natural frequencies for this configuration were much lower than the peak wave frequencies of the sea states. For the extreme events, the results approached similar values. A clear benefit was found in using the compliant line; hence, much smaller line tension was detected. In extreme events the maximum tensions was reduced by 55%. Ridge et al. (2010) states that if the compliance of the mooring system is doubled, the peak tension will be halved. This corresponds well with the results obtained in this study where the horizontal stiffness in the stiff configurations (C1 and C2) are approximately double the stiffness of the compliant system (C3). In the operational conditions the maximum tensions in the compliant system were similar to the tensions in the stiff systems.

The resulting horizontal and vertical excursions were significantly larger in C3, while the pitch motion was smaller compared to the stiffer system. The choice of mooring system, therefore, also needs to be decided based on the restrictions at the site, as the allowable excursions need to be considered.

A quasi-static approach, used in several studies and described in design standards, was used and the results compared with the experimental data in order to evaluate its applicability. In the case with the largest deviation, the calculated tension was determined to be 52% of the measured experimental tension. This can be explained by the many simplifications and since the approach neglects all dynamic effects. Therefore, it is crucial to use a full dynamic analysis tool that includes all these effects.

The present test campaign has established a large database of motion and tension measurements under various sea conditions, and this data should form the basis for validation of dynamic numerical codes in future research.

ACKNOWLEDGEMENT

The work on the present study was funded by the Energy Technology Development and Demonstration Program (EUDP) through the project "Mooring Solution for Large Wave Energy Converters". The authors wish to acknowledge Floating Power Plant for providing data and input to the research.

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Paper D

Screening of Available Tools for Dynamic Mooring Analysis of Wave Energy Converters

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The paper has been published in
Energies, ISSN 1996-1073, 10(7):853, 2017.

energies
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Article

Screening of Available Tools for Dynamic Mooring Analysis of Wave Energy Converters

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Academic Editor: John Ringwood
Received: 30 March 2017; Accepted: 23 June 2017; Published: 27 June 2017

Abstract: The focus on alternative energy sources has increased significantly throughout the last few decades, leading to a considerable development in the wave energy sector. In spite of this, the sector cannot yet be considered commercialized, and many challenges still exist, in which mooring of floating wave energy converters is included. Different methods for assessment and design of mooring systems have been described by now, covering simple quasi-static analysis and more advanced and sophisticated dynamic analysis. Design standards for mooring systems already exist, and new ones are being developed specifically forwave energy converter moorings, which results in other requirements to the chosen tools, since these often have been aimed at other offshore sectors. The present analysis assesses a number of relevant commercial software packages for full dynamic mooring analysis in order to highlight the advantages and drawbacks. The focus of the assessment is to ensure that the software packages are capable of fulfilling the requirements of modeling, as defined in design standards and thereby ensuring that the analysis can be used to get a certified mooring system. Based on the initial assessment, the two software packages DeepC and OrcaFlex are found to best suit the requirements. They are therefore used in a case study in order to evaluate motion and mooring load response, and the results are compared in order to provide guidelines for which software package to choose. In the present study, the OrcaFlex code was found to satisfy all requirements.

Keywords: mooring; station keeping; dynamic analysis; wave energy; OrcaFlex; DeepC

1. Introduction

There is a variety of environmental loads on offshore structures, dominated by loads arising from current, wind and wave, which contribute to the motions of the structure. Therefore, these are vital for consideration when analyzing and designing floating structures, as it is necessary to control the motions in order to satisfy restrictions on the allowed excursion limit or to ensure the stability of the structure. A range of different solutions is available, but usually a mooring system is applied, which by definition is a system of lines connecting the floating structure to the seabed [1,2]. The offshore oil, gas and naval sectors have a long tradition of using moorings as a station-keeping system, and they have gained much experience and knowledge. Today, there is a wide range of design guidelines like DNV-OS-E301 [1], API-RP-2SK [2] or ISO 19901-7:2013 [3], and several authors like [4–8] have dealt with the topic.

The wave energy sector has many different concepts and devices for wave energy conversion; many of which are floating structures with the need for a mooring system [9]. Despite the large knowledge gathered from other offshore sectors, there is still a large amount of failures of wave energy converters (WECs) due to insufficient mooring, which causes significant damage to the devices and their development [4]. In addition, the need for optimizing WEC moorings is significantly greater compared to the mooring of other structures since some authors, as, e.g., [10], state that the cost of moorings for WECs takes up 18–30% of the total structure cost, while it is only 2% for an oil platform. This difference is partly due to the difference in size of the structures and partly the different requirements to and response of the WEC moorings. This puts new demands and focus on the choice of materials, design procedure and, in particular, analysis tools. Studies, as, e.g., [11], focus decidedly on moorings with synthetic ropes in order to optimize the cost and function of the devices, resulting in a great need for a proper tool for analysis. This can, e.g., be experimental work, but in order to include all effects, it is necessary to use sophisticated and expensive experiments. Often, numerical analysis is considered, which can cover both quasi-static and dynamic analysis. In [12], it was found that quasi-static analyses have a tendency to underestimate tensions in extreme waves, and therefore, this study considers only dynamic tools.

There is a great amount of commercial tools available, each of them with a set of specifications, capabilities and limitations, for the investigation of structure response and mooring loads. Many of the tools are mainly aimed at the traditional offshore sector and its requirements, resulting in uncertainty of the applicability to the wave energy sector. The purpose of this study is to identify the requirements of the software tools for WEC mooring analysis when the main concern is to have a tool that allows for obtaining a certified mooring system. In the final design, it will be necessary to validate the tool against experimental work or more sophisticated models following, e.g., the procedure in [13], but this is not the aim of this paper. Each design standard has requirements to the effects that must be included in the analysis, and the purpose of this paper is to identify these and evaluate whether the software packages provide the specifications to satisfy them. Additionally, the promising tools will be compared to each other in a case study in order to highlight potential differences in the capabilities.

The research only focuses on mooring of WECs that can be considered large relative to the incoming waves and only considers passive mooring systems; hence, systems where the mooring is not taking an active part in the energy absorption. The paper first introduces mooring modeling followed by an introduction to the main requirements defined in the design standards. There will be an introduction and comparison of a number of relevant commercially-available software packages, and the most promising tools are chosen for a simple WEC case study. In addition, there will be a comparison of the mooring line tensions and device motions in order to illustrate potential differences, together with advantages and drawbacks from each software package. This should allow for the selection of a dynamic tool, which can be validated in future work.

2. Dynamic Modeling of Floating WECs with Moorings

The combined current, wind and wave load exerted on the structure affect the dynamic behavior of the floating body. Using the Newton–Euler formulation, the equation of motion with reference to the center of gravity (CoG) is defined in Equations (1) and (2) [14]:

$$\overline{f} = m\ddot{\overline{x}} \tag{1}$$

$$\overline{\tau} = \mathbf{I_c}\ddot{\overline{\omega}} + \dot{\overline{\omega}} \times \mathbf{I_c}\dot{\overline{\omega}} \tag{2}$$

where $\overline{f}$ is the resultant force vector, m is the structure mass, $\ddot{\overline{x}}$ is the linear acceleration vector of the system, $\overline{\tau}$ is the resultant moment vector, $\mathbf{I_c}$ is the inertia matrix and $\ddot{\overline{\omega}}$ and $\dot{\overline{\omega}}$ are respectively the angular acceleration and velocity of the structure. The resultant force and moment vectors are the combined forces and moments acting on the structure, hence consisting of a contribution from wind and current together with the total force and moment from wave exposure. The latter arises from the combination of hydrostatic and hydrodynamic pressure fields and results in a time-varying force. The resultant vectors are therefore defined as:

$$\overline{f}_{ex} + \overline{f}_{rad} + \overline{f}_{hyd} + \overline{f}_{c,w} + \overline{f}_m = m\ddot{\overline{x}} \tag{3}$$

$$\overline{\tau}_{ex} + \overline{\tau}_{rad} + \overline{\tau}_{hyd} + \overline{\tau}_{c,w} + \overline{\tau}_m = \mathbf{I_c}\ddot{\overline{\omega}} \tag{4}$$

where $\overline{f}_{ex}$, $\overline{\tau}_{ex}$ and $\overline{f}_{rad}$, $\overline{\tau}_{rad}$ are respectively the wave excitation and wave radiation loads, which can be decomposed from the total wave load based on a linear assumption. $\overline{f}_{hyd}$ and $\overline{\tau}_{hyd}$ are the hydrostatic loads; $\overline{f}_{c,w}$ and $\overline{\tau}_{c,w}$ are the wind and current loads; and $\overline{f}_m$ and $\overline{\tau}_m$ are the mooring loads. The gyroscopic moment ($\dot{\overline{\omega}} \times \mathbf{I_c}\dot{\overline{\omega}}$) from Equation (2) can initially be discarded in Equation (4) [14]. The excitation loads are defined as the loads exerted by the waves on the static structure, while the radiation loads arise from waves that are induced by the moving structure. The hydrostatic loads are the loads exerted by static pressure on the wetted body surface.

A Morison approach or boundary element method (BEM) is commonly used to solve the wave load contribution, but there are methods available that are more sophisticated such as computational fluid dynamics (CFD) or smoothed particle hydrodynamics (SPH).

When applying a mooring system to the floating WEC, the mooring loads are introduced into the equation of motion, as defined in Equations (3) and (4). For each time step, the equations of motion for the mooring lines are solved, and the mooring load is given as input to the equation of motion for the WEC. In general terms, the governing equation for mooring dynamics can be expressed as:

$$\gamma\ddot{\overline{r}} = \overline{F}_T + \overline{F}_B + \overline{F}_\tau + \overline{F}_H + \overline{F}_C \tag{5}$$

where γ is the cable mass, $\ddot{\overline{r}}$ is the acceleration vector, F_T is the force due to axial tension, F_B is the force due to bending moment, F_τ the force due to torsional moment, F_H the force due to hydrodynamic loading and F_C represents the contact forces.

The solution to Equation (5) is commonly found using a finite element (FE) method where the mooring lines are discretized into several elements, and it is possible to advance in time and space by using a range of different methods with different orders of accuracy [15]. An approximation commonly used in mooring analysis tools is the lumped mass approach [16–20]. This can denote that the FE mass matrix is approximated with a lumped mass matrix where the properties of each element are redistributed to the elemental nodes, which results in a diagonal mass matrix and, therefore, eases the calculations and reduces the simulation time. Additionally, the lumped mass approach also implies a method where the lines are divided into a number of elements that are treated as point masses and springs.

Different software packages are able to solve the equations and simulate the mooring response, but how the mooring forces are implemented in the model and how the solution is computed vary for each package and to some extent affect the accuracy of the results.

3. Design Standards

There are several certification companies worldwide with the purpose of ensuring that the design of offshore structures is reliable and safe. By fulfilling the requirements from these companies, it is possible to certify the structure and allow it to be deployed at the desired location. These requirements are all specified in a number of design guidelines covering, e.g., analysis methodology, safety factors, material requirements, etc. Examples of design standards for floating structures are the DNV-OS-E301 [1], API-RP-2SK [2] and ISO-19901-7:2005 [3]. For WEC mooring design, the recent IEC-62600-10 [21] can be used. There is no significant difference between these design standards, and overall, they provide the same requirements. The main difference is on the safety factors, which this study will not treat any further. The following section briefly summarizes the requirements stated in the design guidelines on how to model the floating device, the induced loads on both body and mooring line and what type of analysis should be performed.

From a commercial point of view, the initial objective of the mooring design procedure is to be able to get a certified mooring system; hence, a certification company needs to certify the calculations and tools used. According to DNV-GL, a certain tool is not more likely to give a certified system than others are, and no requirements are stated on which software package should be used. However, it is expected for the software package to be verified against laboratory experiments on a case-to-case basis. For WECs, the ability to model the power take-off (PTO) is a crucial parameter.

3.1. Choice of Analysis

The oil and gas sector typically models mooring systems by a quasi-static approach and, therefore, states it as an appropriate methodology in some design standards. However, the reason for using a quasi-static approach is the large structures and corresponding low responses with insignificant velocities. For an oil and gas platform, large displacements can compromise the safety and functionality of the structure, and so, the system is designed to significantly reduce the response. In the WEC sector, the structures are typically of much smaller masses, and often, there is a tolerance of or even a desire for larger displacements. Therefore, a more distinct response with much higher velocity, but smaller loads, is often experienced, putting more demands on the type of analysis. According to both API-RP-2SK [2] and IEC-62600-10 [21], this requires a dynamic analysis, while DNV-OS-E301 [1] states that a quasi-static analysis is only appropriate in some conditions. However, for the type of mooring layout and site specification typically involving WECs, also DNV-OS-E301 [1] requires a dynamic analysis. For large WECs, a quasi-static analysis could be justified in a fatigue assessment in operational conditions, but as there must be a validation of the mooring in the ultimate limit state (ULS), the need for a dynamic analysis is still present. For a preliminary analysis, a quasi-static analysis might be more efficient due to the simplicity of the procedure, followed by dynamic analysis for the final design and validation. For both quasi-static and dynamic analysis, it is required that the weight and buoyancy of all components are included, together with the elasticity of the lines and interaction with the seabed. Often, the quasi-static approach neglects all dynamic effects from mass, damping and fluids and only considers horizontal displacements, while the dynamic analysis solves the equation of motion for all degrees of freedom and includes all dynamic effects [2].

In a mooring system, significant non-linearities are present, which the software package must model appropriately. Non-linearities cover, e.g., non-linear stretching of lines, changes in geometry, fluid and bottom effects. When solving the equations of motion, it is possible to use either a frequency or time domain approach. In the frequency domain, a set of linear equations of motion treats the motions and calculates and combines statistical peak values for different motion contributions. Since the equations of motion are linear, analyzing in the frequency domain requires linearization of all non-linearities, thereby introducing inaccuracy between nature and the model. This linearization is unnecessary in the time domain where direct numerical integration will solve the system. A time domain analysis, therefore, also provides time histories of, e.g., WEC displacements and line tensions. It should be noted that this could also be found from the frequency domain parameters as these contain information on phase and amplitude. When having the time series, it is then possible to use statistical analysis to find extreme values. Using, e.g., a finite element method to model the mooring lines in the time domain gives the opportunity to include all of the dynamic effects from mass, damping and fluid loading. The time domain analysis should be modeled long enough to ensure statistical satisfaction, and often a total number of 1000 waves or three hours of simulation is recommended [1]. When solving the equation of motions, two different methods can be used, namely the coupled or uncoupled method [1]. In the uncoupled method, the motions are initially solved either by neglecting the influence of the mooring lines or by including them as a constant stiffness. The fairlead motion of the WEC is then fed into a cable solver, and tensions in the mooring lines are solved. In the coupled method, the complete system of equations for both WECs and lines are solved simultaneously. In order to fully model the influence of mooring lines on the WEC motions and the full non-linear behavior of the mooring system, the design standards state that a coupled analysis should be used in almost all cases. Similarly, this requires the use of a time domain simulation with all non-linearities and dynamic effects.

3.2. Environmental Loads

The ability to model the environment and induced loads is a vital capability of the software. The design standards have specific requirements on how to model and include loads, listed below.

3.2.1. Current

Current can be treated as a steady force calculated by a drag force formulation as:

$$F = \frac{1}{2}\rho C_d A v^2 \tag{6}$$

where F is the current force, ρ is the fluid density, C_d is the drag force coefficient, A is the cross-sectional area and v is the fluid velocity. When calculating the loads on the WEC and the lines, the current must be modeled with a vertical variation according to profiles specified in the design standards.

3.2.2. Wind

The wind load modeling must consist of a steady mean component and a time-varying gust effect. The mean component should be described with a vertical wind profile and time-variation described by relevant wind spectra. A drag load formulation will calculate the load similar to the current load.

3.2.3. Wave

The wave load can be decomposed with good accuracy into a few components, i.e., a wave frequency (WF) load, oscillating at the frequency of the incoming waves, a second order low frequency (LF) load and a second order mean drift load. The description of the response must include both the WF, LF and mean drift. For small structures with a diameter less than a fifth of the wavelength, radiation and diffraction loading are negligible, and a Morison approach can be used, cf. [1] and Figure 1a. For larger structures, radiation and diffraction dominate the wave loads, and BEM codes, as e.g., NEMOH [22] or WAMIT [23], can be used.

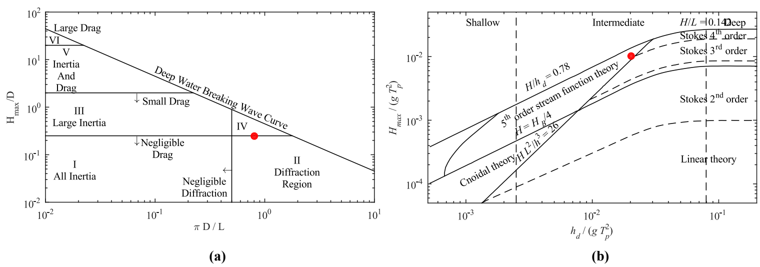

Figure 1. (**a**) Plot of the case investigated later in this paper (red marker) as a function of wave height H, characteristic diameter D and wavelength L. The defined wave force regimes are in accordance with the boundaries defined in [1,24]; (**b**) Plot of the investigated case (red marker) as a function of wave height H, wave period T and water depth h_d and with the indication of needed wave theory. The boundaries are based on the definition in [25].

4. Dynamic Mooring Analysis Tools

There is a large number of software packages for mooring analysis of floating structures, covering, e.g., quasi-static tools for design of ports and offshore oil and gas structures and likewise many tools for dynamic analysis. For the WECs, only dynamic tools are considered. A preliminary analysis provided the list of relevant software candidates defined in Table 1. The versions of the software packages correspond to the version available at the time of the screening.

Table 1. List of relevant software to consider for dynamic mooring analysis.

#	Software Package	Developer	Ref.
1	Aqwa v. 15	ANSYS Inc.	[20]
2	Flexcom v. 8.3	Wood Group Kenny	[26]
3	MOSES	Bentley System	[27]
4	OrcaFlex v. 10.0	Orcina Ltd.	[19]
5	ProteusDS 2015 v. 2.25.0	Dynamic System Analysis Ltd.	[28]
6	SeaFEM v. 13.6.5	Compass	[29]
7	DeepC v. 4.8	DNV-GL	[30]

There are many other dynamic tools, e.g., the in-house code CASH by GVA, ZenMoor by Zentech Inc., WAMSIM by DHI Group and the MooDy code by Chalmers University of Technology [31]. A widely-used software package is the Ariana-3Dynamics by Bureau Veritas, which uses a cable dynamic module, developed by Wood Group Kenny and which is based on the Flexcom software. From these additional codes, this paper will only consider MooDy further. The following sections will include a short description of each software package, followed by a full comparison of specifications.

Aqwa [20] is a software package developed by ANSYS Inc. used to investigate the influence from environmental loads on fixed and floating offshore structures. The software benefits from having a hydrodynamic BEM code implemented and is capable of solving both first and second order wave effects in both the frequency and time domain. Both wind and current are modeled according to the design standards. The dynamic mooring lines are modeled using a discrete lumped mass approach where each line is divided into a number of segments where the mass is concentrated into a corresponding node. The code uses a two-stage predictor corrector method for advancing in time.

Flexcom [26], developed by MCS Kenny, is a structural analysis tool aimed at the offshore sector. The software is capable of including all environmental loads specified by the design standards and takes the hydrodynamic coefficients from external diffraction-radiation analysis software as input, including response amplitude operators (RAOs) and quadratic transfer functions (QTFs) for the calculation of respectively first and second order wave effects. The software models the mooring lines using a finite element solution based on a hybrid finite element beam model where the axial displacement and rotation are given by a linear basis function, a cubic basis function for the transverse displacement and a constant basis function for the axial force and torque moment. Time stepping is done by a second order implicit generalized-α method with the option of variable time steps based on the current period parameter.

MOSES [27] is a software package for modeling, designing and planning for offshore floating structures, including the calculation of the hydrodynamic coefficients. The environmental loadings are well described, and mooring lines are solved for tension, bending and torsion. The beam equations are solved by linear finite elements. An implicit Newmark β-scheme, typically of second order, is used for time stepping.

OrcaFlex [19] is a marine dynamic software package developed by Orcina Ltd. allowing full analysis in the time and frequency domain. The software is capable of calculating wave loads from a Morison approach, and for radiation-diffraction loading, input is needed in terms of RAOs and QTFs. Wind and current loading are both considered according to design standards. OrcaFlex solves both tension, bending and torsion using a discrete lumped mass approach with a time-stepping scheme that can be either explicit by the first order forward Euler method or implicit by the second order generalized α-method.

ProteusDS [28], developed by Dynamic System Analysis Ltd., is a software for full dynamic analysis of offshore structures in the time domain. The software uses a Morison approach for the calculation of wave loads or additionally user-specified hydrodynamic coefficients. The present version of ProteusDS does not calculate the second order wave effects, and the wave drift and LF motion are not computed. A part of the contribution to the wave drift is by now included, and the implementation of QTFs is expected in future releases. The software does contain advanced mechanism modeling capabilities suitable for predicting WEC performance, important for characterizing the PTO of WECs. The mooring lines are solved for both tension, bending and torsion, using a cubic spline finite element method, meaning that the solution inside an element is given by a fourth order polynomial with a lumped mass matrix in order to speed up calculations. A fourth order Runge–Kutta scheme with an adaptive time step is used for advancing in time. The numerical error introduced by the time integrator is maintained below a defined level by adjusting the time step.

SeaFEM [29] is a software package developed by Compass with the aim of seakeeping and maneuvering simulation with the implementation of wave, wind and current effects on the structure. The software can perform both frequency and time domain analysis, and SeaFEM has the advantage of using a 3D FE model for solving the total fluid domain and its interaction with the floating structure in each time step. The software has not directly implemented wind load, which needs to be defined as a user-specified load. Currents can be included, but only with a constant velocity over the water depth. The dynamic mooring equations are solved by linear bar elements, and the solution is updated in time using a second order Newmark scheme.

DeepC [30] is a software package distributed by DNV-GL consisting of two pieces of software Riflex and Simo and combined with the HydroD package for hydrodynamic analysis. In combination, the DeepC software is capable of analyzing the environmental impact on floating structures and mooring lines accounting for first and second order wave effects, current and wind. The lines are solved for tension, bending and torsion and come with a linear bar element and with hybrid bar elements using a combination of linear basis functions and cubic basis functions. The mass matrix can be lumped for computations that are more efficient. The time integration uses an implicit Newmark β-scheme, typically second order, and solved with a Newton–Raphson method.

MooDy [31] differs from the other software by being an in-house code of Chalmers University and not being a complete software package. The code is merely a dynamic cable solver and needs to be combined with other codes that are capable of solving the interaction between WEC and cables. A feature of MooDy is the use of the spectral/hp discontinuous Galerkin method, i.e., an arbitrary order (set by user) finite element method. The code uses explicit time stepping, including the third order Runge–Kutta scheme and a second order leap-frog scheme. Since MooDy is not a commercial software, it is only included in the following comparison in order to illustrate the potential of cable solvers and will not be considered for the case study.

4.1. Comparison of Software Packages

Largely, the software packages provide the same specifications, and since most codes are validated, it is expected that the obtained results will be in the same range. Still, some of the software packages excel when comparing all specifications. For comparison, the requirements from the design standards are considered. According to these, moorings need to be analyzed in the time domain using a fully-coupled analysis, and in addition, the software needs to be able to model the wind and current loads as profiled in the vertical direction. Modeling of the time-varying nature of the wind is a requirement, while it is only necessary to model a steady current velocity. Table 2 compares the software according to these requirements.

Table 2. Comparison of software capabilities concerning analysis type and ability to model wind and current. ✓and ✗ denote respectively that the capability is available or unavailable.

	Analysis		**Wind**		**Current**	
	Time domain	**Coupled**	**Profiled**	**Spectrum**	**Profiled**	**Time varying**
Aqwa	✓	✓	✓	✓	✓	✗
Flexcom	✓	✓	✓	✓	✓	✗
MOSES	✓	✓	✓	✓	✓	✗
OrcaFlex	✓	✓	✓	✓	✓	✓
ProteusDS	✓	✓	✓	✓	✓	✓
SeaFEM	✓	✓	✗	✗	✗	✗
DeepC	✓	✓	✓	✓	✓	✓

As seen from Table 2, all software fulfils the requirements, though SeaFEM is only capable of modeling a uniform current and wind.

When considering shallow water depths, in which many WECs are planned for deployment, the need for high order wave theory is present. All software needs to be able to implement diffraction/radiation loads in the case of large structures either by performing the analysis itself or using input variables. Table 3 specifies the capabilities for each software package concerning the wave influence.

Table 3. Comparison of the software capabilities of modeling waves. ✓and ✗ denote respectively that the capability is available or unavailable.

	Wave Theory				**Hydrodynamic Analysis**			
	Linear	**Stokes**	**Stream**	**Irregular**	**Diff. /Rad. Loads**	**Morison Loads**	**Diff./Rad. Input**	**2nd order Wave Effects**
Aqwa	✓	2nd	✗	✓	✓	✓	✗	✓
Flexcom	✓	5th	✓	✓	✗	✗	✓	✓
MOSES	✓	5th	✓	✓	✓	✓	✗	✓
OrcaFlex	✓	5th	✓	✓	✗	✓	✓	✓
ProteusDS	✓	5th	✗	✓	✗	✓	✓	✗
SeaFEM	✓	✗	✗	✓	✓	✓	✗	✓
DeepC	✓	5th	✗	✓	✓	✓	✓	✓

Some variation of the implemented wave theories is present, but in general, high order Stokes theory is available. All software packages are capable of simulating irregular sea states, and diffraction/radiation or Morison load can either be computed by the software packages or input by the user. Most software includes second order wave effects, except from ProteusDS, which at present only includes some contribution to the wave drift. SeaFEM has implemented a second order solver, but this cannot be enabled together with current loads in the available version.

In mooring analysis, the dynamic cable solver is obviously one of the main features. Table 4 defines the specification of each software package, now also considering the MooDy code. The comparison considers the included contribution to the calculated mooring loads, together with the order of the schemes used for advancing in time and space. The use of a lumped mass matrix in the software is stated, but the table does not differ between the two meanings of the approach as described previously. Finally, the table lists the capability to model non-linear axial mooring line stiffness, which is an important feature when modeling, e.g., synthetic ropes.

Table 4. Comparison of capabilities of the implemented cable solvers in the investigated software packages. ✓and ✗ denote respectively that the capability is available or unavailable. ∗ indicates that no information could be found in the public available theory manual. LM and FE denote respectively a discrete lumped mass and finite element approach. *p*-th indicates that the order is user-specified.

Dynamic Cable Solver							
	Tension	**Bending**	**Torsion**	**Spatial Order**	**Temporal Order**	**LM/FE**	**Non-linear Stiffness**
Aqwa	✓	✓	✗	∗	∗	LM	✓
Flexcom	✓	✓	✓	2nd	2nd	FE	✓
MOSES	✓	✓	✓	2nd	2nd	FE	✗
OrcaFlex	✓	✓	✓	∗	2nd	LM	✓
ProteusDS	✓	✓	✓	4th	4th	FE	✗
SeaFEM	✓	✗	✗	2nd	2nd	FE	✗
DeepC	✓	✓	✓	2nd	2nd	FE	✓
MooDy	✓	✗	✗	*p*-th	3rd	FE	✗

The capabilities are identical in most software packages, while codes such as MooDy and ProteusDS are highlighted due to the high spatial and temporal order. However, it is still expected that similar results will be achieved in all software, despite the order of the code.

Tables 2–4 are suitable for choosing an optimal tool for dynamic mooring design and ensuring that the design standards can be satisfied. Many of the software packages have apparent similar specifications, but OrcaFlex, DeepC, ProteusDS and SeaFEM are considered strong candidates. As seen in Table 4, the dynamic cable solver in ProteusDS is most advanced, but at present, the main drawback of this software is the lack of capability to calculate the second order wave effect. Considering WECs in shallow water with compliant mooring, the drift effect is of paramount importance.

The SeaFEM software mainly advances because of its hydrodynamic solver, based on an FE formulation of the entire fluid domain. The software is, therefore, not dependent on frequency domain results (RAOs and QTFs) for calculation of motions and will provide the best description of non-linear irregular waves. Similar, the code has the potential to provide a better description of the current effect as it is calculated from the pressure on the body and not from a drag coefficient. However, for large structures with a high Reynolds number and turbulent flow, SeaFEM does not model this properly since the solver assumes steady streamlines. At present, the solver is also not capable of solving second order effects in combination with current, and computing just the second order wave effects puts such high demands on the mooring solver that the solutions often diverge. Another important factor to consider if using SeaFEM is the much higher computational time, compared to the other software packages. It should be noted that this is caused by the fact that SeaFEM solves the entire domain in each time step and, therefore, gives a better description paid by the longer simulation time.

Considering Tables 2–4, DeepC and OrcaFlex appear to have similar specifications. The DeepC solver might be more advanced than OrcaFlex as it can solve the mass continuously over the mooring line, while OrcaFlex uses the lumped mass approach and models the lines as point masses and springs. However, studies have indicated that an acceptable level of accuracy can also be achieved with this method. Both DeepC and OrcaFlex have been validated in several studies, and OrcaFlex is widely used for commercial purpose in different offshore sectors, while DeepC is developed and distributed by the certification company DNV-GL. A drawback of the DeepC package is the need for calculating static position in additional software, and for long simulations like, e.g., three hours, the software introduces a limitation of the number of calculations, which results in higher time steps and therefore possibly unstable solutions.

This paper will compare the performances of the OrcaFlex and DeepC packages, as these seem to fulfil the requirements in the design standards. No validation of the software packages will be conducted in this paper, as they are both commercial software packages, which have been validated for other applications as stated by [32,33]. The purpose of the following section is to investigate the potential difference between the results from each software package and allow for selection of the final tool. In a later publication, the selected software package will be validated for the present application.

5. Case Description

The considered case resembles a range of large floating WECs from the Danish wave energy sector. Examples of this kind of structure could be the Floating Power Plant [34], LEANCON Wave Energy [35], KNSwing and Wave Dragon [36] (cf. Figure 2) with widths in the range of 28–152 m and lengths of 60–240 m.

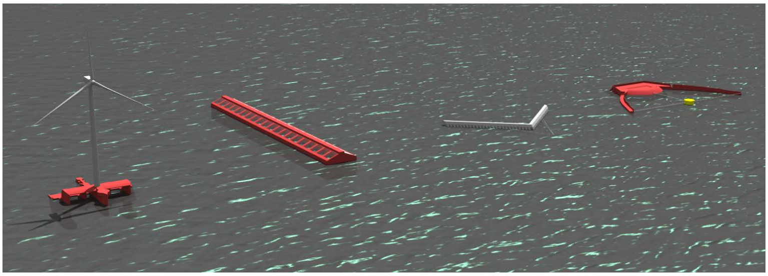

Figure 2. Examples of large floating WECs. From left to right: Floating Power Plant, KNSwing, LEANCON Wave Energy and Wave Dragon.

The case will analyze the mooring in the ULS and consider extreme events comparable to sites where these types of WECs are expected to be deployed. Because of this, it is assumed that the WEC is in storm protection, and the PTO is not included in the calculations. For simplicity, the structure is assumed to have the shape of a barge with dimensions in the range of the mentioned WECs. The geometry of the device is illustrated in Figure 3 and the dimensions specified in Table 5.

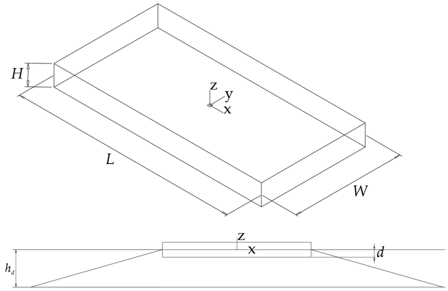

Figure 3. The investigated WEC with the illustration of the mooring system.

Table 5. Geometrical specification of the investigated WEC.

Parameter	Unit	Value
Width, W	(m)	60.0
Height, H	(m)	12.0
Length, L	(m)	120.0
Draught, d	(m)	6.0
Mass, M	(kg)	4.428×10^7
Estimated drag coefficient, C_d	(-)	1.1
Center of gravity, (x,y,z)	(m)	(0.0, 0.0, 0.0)
Fairlead 1, (x,y,z)	(m)	(60.0, −30.0, 0.0)
Fairlead 2, (x,y,z)	(m)	(60.0, 30.0, 0.0)
Fairlead 3, (x,y,z)	(m)	(−60.0, 30.0, 0.0)
Fairlead 4, (x,y,z)	(m)	(−60.0, −30.0, 0.0)

The device is spread-moored with four mooring lines consisting of synthetic rope, with specifications as listed in Table 6.

Table 6. Mooring line and anchor specifications.

Mooring Line Specifications		
Diameter, D	(m)	0.104
Unstretched length, l	(m)	150.0
Nominal mass in air, m_{air}	(kg/m)	6.67
Nominal submerged mass, m_{sub}	(kg/m)	0.64
Minimum breaking strength, MBS	(kN)	2461
Extension at max. load, $\varepsilon_{\mathrm{MBS}}$	(%)	27
Linearized stiffness, EA	(kN)	9114.8
Young's modulus, E	(MPa)	1072.9
Anchor Specifications		
Center of gravity, (x,y,z)	(m)	(0.0, 0.0, 0.0)
Anchor 1, (x,y,z)	(m)	(166.1, −136.1.0, −30.0)
Anchor 2, (x,y,z)	(m)	(166.1, 136.1.0, −30.0)
Anchor 3, (x,y,z)	(m)	(−166.1, 136.1.0, −30.0)
Anchor 4, (x,y,z)	(m)	(−166.1, −136.1.0, −30.0)

The characteristics of the barge, in terms of moment of inertia I and hydrostatic stiffness K_{hyd}, are calculated either by the software packages or manuallyand is listed in Table 7. The characteristics from each method have been compared and show identical results.

Table 7. Characteristics of the investigated WEC.

Parameter		Value
Hydrostatic stiffness,	$K_{hyd_{3,3}}$	7.24×10^7 N/m
	$K_{hyd_{4,4}}$	2.04×10^{10} N/rad
	$K_{hyd_{5,5}}$	8.55×10^{10} N/rad
Mass moment of inertia,	$I_{x,x}$	1.38×10^{10} kg m^2
	$I_{y,y}$	5.38×10^{10} kg m^2
	$I_{z,z}$	6.64×10^{10} kg m^2

A site suitable for the type of devices in Figure 2 is considered and is characterized by the parameters defined in Table 8. The sea state is described by a JONSWAP spectrum, and for simplicity, a current profile is assumed with no variation in velocity over the depth. In these simple simulations, any wind force is not included.

Table 8. Sea state characteristics investigated in the case study.

Parameter	Unit	Value
Water depth, h_d	(m)	30.0
Significant wave height, H_s	(m)	8.3
Peak wave period, T_p	(s)	12.3
Peak enhancement factor, γ	(-)	3.3
Current velocity, v_c	(m/s)	1.0
Fluid density, ρ_w	(kg/m^3)	1025

Figure 1a shows that the wave loads on the WEC are dominated by the radiation and diffraction components. Considering also Figure 1b, it is seen that the sea state can be considered to be in intermediate water depths with non-linear waves. It should be noted that the calculation of hydrodynamic coefficients is based on linear theory and that Figure 1a is also for deep water conditions.

The HydroD module of the DeepC package is capable of solving the hydrodynamic coefficients for added mass, radiation damping and the first and second order wave excitation force. OrcaFlex needs these coefficients as input from additional software. This case uses the open source BEM code NEMOH [22]. When comparing the results found from DeepC and NEMOH, there is good agreement between the results. Figure 4 presents the calculated motion RAOs from NEMOH and DeepC.

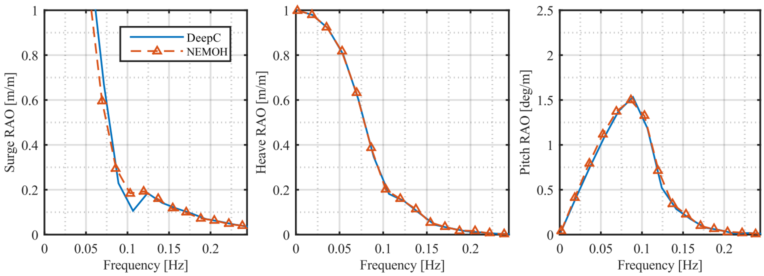

Figure 4. Motion response amplitude operators (RAOs) for the surge, heave and pitch degree of freedoms (DOFs) of the unmoored WEC calculated from the DeepC and NEMOH code.

6. Mooring Analysis

In order to illustrate the performance and usability of the software, a simulation with the described sea state and model is performed for a time duration of 15,000 s. The following sections illustrate the differences between the results obtained from each software package.

6.1. Static Configuration

The static configuration of the system is obtained from equilibrium between weight, buoyancy and pretension of WEC and mooring lines, without the presence of any contribution from waves, wind and current. Table 9 shows the comparison between static line tensions in the two codes. Almost identical results are obtained, but it needs to be noted that DeepC is limited in the calculation of the static configuration as it assumes that the buoyancy of the WEC is equal to its mass. When applying mooring lines, the additional mass and potential pretension are not accounted for, and the WEC is pulled down. Therefore, it is necessary to calculate the static configuration beforehand and to apply a compensation force. For the present case, this force was calculated in a self-made script. For a simple configuration, this is relatively easy, but for more advanced systems, it might slow down and complicate the analysis.

Table 9. Comparison of simulated static mooring line tension in OrcaFlex and DeepC. The analytical value is also listed.

Software Package	Static Tension in Mooring Lines
Analytical	183.5 kN
OrcaFlex	183.16 kN
DeepC	183.19 kN

6.2. Dynamic Response Analysis

Based on the hydrodynamic properties of the WEC, the motion response of the device and the corresponding tensions in the lines are simulated. Figure 5 presents the motion RAOs calculated from the output response from both software packages. The results are bandpass filtered with lower and upper limits of respectively $1/3\ f_p$ and $3\ f_p$ (where f_p is the peak frequency) due to insignificant wave energy below and above these values, resulting in high disturbance on the calculated RAOs. The motion amplitudes are oscillations around the mean value of respectively 10.7 m and 11.2 m for OrcaFlex and DeepC for the surge degree of freedom (DOF).

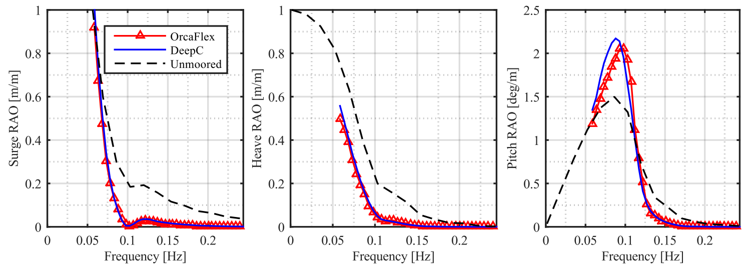

Figure 5. Motion RAOs for the surge, heave and pitch DOF of the moored WEC obtained from both the OrcaFlex and DeepC packages.

Figure 5 illustrates how the mooring system reduces the motion amplitudes in both surge and heave. Due to the band pass filtering, it is not possible to detect a resonance frequency for these two DOFs, because they are both lower than $1/3\ f_p$. The analytical natural frequency for surge, for instance, was found to be 0.008 Hz and hence, significantly below the wave frequencies. This resembles a realistic case, as floating structures ideally are designed with natural frequencies outside the wave frequencies of extreme events.

The pitch natural frequency is approximately 0.1 Hz and is well defined in Figure 5. The presence of the mooring system increases the pitch motion amplitude while the resonance frequency is slightly increased. The results obtained from OrcaFlex and DeepC show good agreement and show the highest deviation in the pitch DOF where the motions calculated by DeepC are higher than those calculated by OrcaFlex, and the resonance frequency is offset.

Figure 6 reports the tension spectral density in the line on the seaward side of the WEC. The amplitudes are around the mean tension of 629.8 kN for OrcaFlex and 614.9 kN for DeepC.

Similar to motion results, the calculated tensions in the lines are in the same range, and without any difference in the spectral shape. The peak value amplitudes have a slightly higher value in the DeepC code, but the difference approaches zero for shorter frequencies. When comparing the time series, a maximum tension of 2100.7 kN is obtained in OrcaFlex, while 1794.5 kN is obtained in DeepC. The $F_{1/250}$ load is 1616.3 kN in OrcaFlex, while 1711.2 kN in DeepC, where $F_{1/250}$ corresponds to the average of the 1/250 of the maximum peaks. It should be noted that the wave time series used in the two simulations are not identical, but only following the JONSWAP spectrum with identical frequency domain parameters. To obtain more reliable results for comparison, more time series should be computed and statistical information identified.

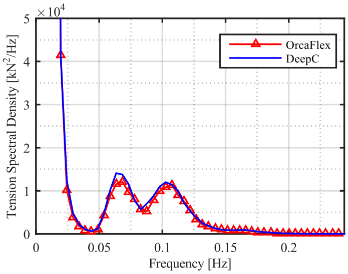

Figure 6. Tension spectral density in the mooring line on the seaward side of the WEC.

6.3. Response to Harmonic Waves

In order to provide a direct comparison of a time series computed by DeepC and OrcaFlex, a simulation of the WEC exposed to harmonic waves is used. The harmonic wave series has a wave height and period similar to the significant wave height and peak period from the previous case.

Figure 7 presents a comparison between OrcaFlex and DeepC time series for surge and line tension where no wave drift and current is present. The two software packages provide approximately similar results and seem to have identical values at the wave crests and small variations in the wave through. As previously stated, it is a vital parameter to be able to model non-linear material properties, as synthetic ropes seems to become subjected to more research. Figure 7 additionally illustrates the ability of the software packages to model the non-linear axial stiffness of a nylon line. If the stiffness curve for this material were linearized, it would result in a similar stiffness as used previously in the case. There is an agreement between the two software packages, together with a reduction in the mooring line tension when using non-linear axial stiffness.

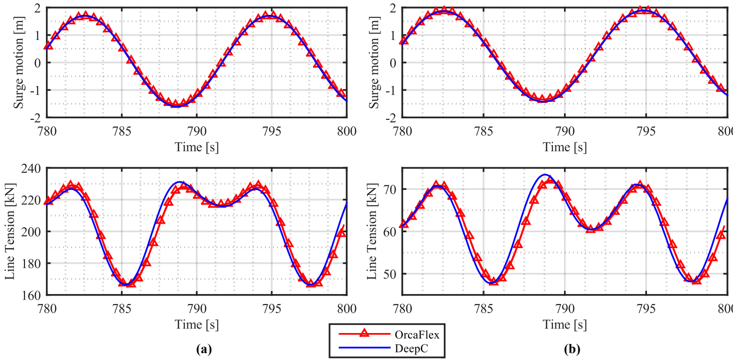

Figure 7. Comparison of surge motion and line tension time series obtained in OrcaFlex and DeepC. (**a**) is for linear axial stiffness of the lines, while (**b**) is for non-linear stiffness.

Both OrcaFlex and DeepC model the mean equilibrium position as the combination of two excursion components; an excursion arising from the presence of current and an excursion from wave drift. The load on the device from the current is in both software packages expected to be similar as they are based on a drag formulation based on input variables like the drag coefficient, current velocity and cross-sectional area. The induced excursion and corresponding line tensions arising from current only are, therefore, also found to be similar, cf. Table 10, which presents the induced horizontal surge motion and line tensions.

Table 10. Comparison of surge motion and line tension obtained by OrcaFlex and DeepC. The table compares exposure to current and mean wave drift separately and combined.

	Exposure of Current	Mean Wave Drift	Combined
OrcaFlex	1.79 m/259.1 kN	2.53 m/306.7 kN	5.06 m/414.7 kN
DeepC	1.78 m/258.4 kN	2.27 m/288.9 kN	5.38 m/428.5 kN

The mean wave drift is calculated based on the input horizontal wave drift coefficients. These are the direct out-put of the HydroD analysis and, therefore, used in the DeepC package, or they can, e.g., be calculated from the output of Nemoh. The test case for the two software packages uses similar drift coefficients. However, as seen in Table 10, the difference between the results of OrcaFlex and DeepC is more prominent than for the current only. When considering the case where both wave drift and current are present, again the difference is present. Despite the fact that OrcaFlex provides the highest surge motion and line tension when considering the separate effects of the current and wave drift, DeepC provides the largest values when combining the contributions. An explanation could be how the software packages calculate the wave-current interaction. From Table 10, the importance of being able to include second order wave drift is clear. By using a software package only capable of including the current load and no mean wave drift, only approximately 60% of the actual line tension is taken into account.

7. Discussion

The comparison of software packages for mooring analysis illustrates that most of the existing software packages have the potential of fulfilling requirements defined in the design standards and thereby allowing for a certified mooring system. Some of the software seems more sophisticated than others, and a software package such as, e.g., ProteusDS excels with the dynamic cable solver while SeaFEM excels with its hydrodynamic solver. However, not fully including second order wave effects limits both of these, which is why the two software packages OrcaFlex and DeepC were chosen for testing on a simplified WEC. These two software packages provide an overall solution for mooring analysis and have the potential of including all needed effects. By comparing the results from the two cases, there was good agreement between the software. In later work, it is paramount to validate the tools against experimental data or a CFD/SPH model before they can finally be used for a certified mooring analysis. This study has now presented that the two tools provide the necessary capabilities and produces similar results. For further comparison, however, it would be advantageous to perform more than one time series simulation, which will allow for statistical comparison between the results.

The two software packages both advance by having a simple user interface and high usability when defining the model, while OrcaFlex eases the process of defining the simulation procedure, integration parameters and provides better support in defining reasonable parameters. Computational time appears to be relatively similar for the two cases, but since no convergence analysis was performed on the discretization of the mooring lines, it is not possible to know if any of the software packages can be optimized to simulate faster than the other.

8. Conclusions

The present study presents an initial study on the applicability of commercial analysis tools for WEC mooring designs. The requirements defined in the design standards were briefly described, which the tools need to be able to model in order to ensure the certification of the final mooring design. The main requirements consider modeling of environmental loads, non-linearities and dynamic effects. The study presents a list of potential software and lists their available capabilities at the time of the study. Most software packages are under a constant development and it is, therefore, recommended to investigate them before future analysis by e.g., following the same procedure as in this paper. Most software provides almost identical capabilities, while some software packages have more sophisticated methods available. From the list, the study highlights the two packages DeepC and OrcaFlex as potential software solutions in mooring analysis and simulates a test case resembling a large floating WEC. The results showed good agreement between the two software packages when considering both motion and tension responses. A comparison of the abilities of the software packages to model current and wave drift showed little variation, but still provided results in the same range. Finally, a test of the ability to include non-linear axial mooring line stiffness again showed good agreement between the two software packages and resulted in a reduction in loads, caused by the more compliant line.

For further comparison between the two cases, it would be reasonable to include more time series and statistical analysis or additionally to simulate an identical time series. Validation against experimental results or CFD simulations would similarly provide a better indication of the capabilities of the software packages and is a necessary step before the final design of mooring systems.

Based on the results from the test cases, the conclusion is that the two software packages both have the potential for use in WEC mooring analysis. DeepC has the drawback that it requires use of additional software to calculate static position and inclusion of a compensation force. In addition, the evaluation of the software package concluded it to be less user-friendly than OrcaFlex and to have a limitation in solving the number of potential problems in each simulation. This leads to higher time steps, which potentially makes the mooring solver unstable. Since relatively long time series often are necessary for statistical purposes, this might cause problems in some cases.

It is found from the analysis that similar results are obtained from both software packages. DeepC has some drawbacks, while OrcaFlex is widely used in the commercial sector and fulfills all of requirements that have been listed. Therefore, there are no arguments for not using OrcaFlex in the analysis of mooring systems for large WECs.

Acknowledgments: This study was funded by the Energy Technology Development and Demonstration Program (EUDP) through the project "Mooring Solutions for Large Wave Energy Converters" (Grant number 64014-0139). The authors wish to acknowledge DNV-GL, Compass, Dynamic System Analysis Ltd. and Orcina Ltd., for providing information and access to their software packages together with the review of this paper. The authors further wish to acknowledge Claes Eskilsson from Chalmers University of Technology for providing input to this paper.

Author Contributions: Jonas Bjerg Thomsen defined the overall outline of the study and the relevant parameters to investigate, with input from Francesco Ferri and Jens Peter Kofoed. Jonas Bjerg Thomsen collected data and information and performed the numerical analysis, analyzed the data and drafted the paper. Jonas Bjerg Thomsen, Francesco Ferri and Jens Peter Kofoed finalized the paper.

Conflicts of Interest: The authors declare no conflict of interest.

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Paper E

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The paper has been published in
Journal of Marine Science and Engineering, ISSN: 2077-1312, 5(4):45, 2017

Journal of
Marine Science and Engineering

Article

Validation of a Tool for the Initial Dynamic Design of Mooring Systems for Large Floating Wave Energy Converters

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Received: 11 May 2017; Accepted: 13 September 2017; Published: 22 September 2017

Abstract: Mooring of floating wave energy converters is an important topic in renewable research since it highly influences the overall cost of the wave energy converter and thereby the cost of energy. In addition, several wave energy converter failures have been observed due to insufficient mooring systems. When designing these systems, it is necessary to ensure the applicability of the design tool and to establish an understanding of the error between model and prototype. The present paper presents the outcome of an experimental test campaign and construction of a numerical model using the open-source boundary element method code NEMOH and the commercial time-domain mooring analysis tool OrcaFlex. The work used the wind/wave energy converter Floating Power Plant as a case study, which is defined as a large floating structure with a passive mooring system. The investigated mooring consists of a three-legged turret system with synthetic lines, and it was tested for both operational and extreme events. In order to understand the difference between the model and experimental results, no tuning of the model was done, besides adding drag elements with values found from a simplified methodology. This resembles initial design cases where no experimental data are available. Generally good agreement was found for the tensions in the lines when the drag element was applied, with some overestimation of the motions. The main cause of difference was found to be underestimation of linear damping. A model was tested with additional linear damping, and it illustrated that a final analysis needs to use experimental data to achieve the best results. However, the analyses showed that the investigated model can be used without tuning in initial investigations of mooring systems, and it is expected that this approach can be applied to other similar systems.

Keywords: wave energy; mooring; numerical; NEMOH; OrcaFlex; validation

1. Introduction

During the last few decades, the large focus on renewable energy sources has led to the suggestion of many different types of devices for harvesting energy. A number of these are wave energy converters (WECs), which use different principles for harvesting wave energy. Despite the comprehensive research, the wave energy sector is still not in a commercial state, and above all, there is a need to reduce the levelized cost of energy (LCOE). WECs vary greatly in size and structure, with some of the devices considered as being large floating structures. These form the basis for the present study, with examples of such structures being the Floating Power Plant [1], KNSwing, LEANCON Wave Energy [2] and Wave Dragon [3]. Naturally, floating structures like these need a system to ensure station keeping, which is often solved by using mooring systems. Based on the working principle of the power take off (PTO) of the WEC, the mooring system can either be considered passive, meaning that it does not take part in the power absorption, or it can be considered active or reactive if it affects the PTO [4]. Common for the type of structures considered in this study is the use of passive mooring systems.

Moorings highly affect the survivability and cost of floating WECs, since even a partial failure of this system can result in a total loss of the device. In addition, the cost of moorings has proven to take up a large part of the total structural cost [5,6] and is, therefore, vital to investigate and improve in order to lower the LCOE. Despite a large experience in mooring design from other offshore sectors, a large number of the WECs deployed offshore have failed due to insufficient moorings [5]. This can partly be explained by a tendency to apply mooring principles from the traditional oil and gas sector with a catenary system of mooring chains. Studies like, e.g., [7–9] show not only that a chain results in a very stiff system with resulting high loads, but also that there could be a great potential for cost reduction in the application of synthetic lines. As a result of this, the developers of the mentioned WECs now consider this as a relevant mooring solution. Consequently, there is a great need for research into cost and reliability optimization of the moorings. A method must be defined that can be used by similar WECs to make an initial investigation of the mooring response.

In order to secure the survivability of the mooring and structure, the system must be designed to survive in all relevant conditions, and therefore, the design parameters like tensions and motions need to be evaluated. According to standards like, e.g., [10–12], it is particularly important to ensure survival in extreme conditions, which is why extreme conditions are the focus of this research. During these conditions, the PTO of the mentioned WECs will be in safety mode and is, therefore, not necessary to consider in the design. A method to identify the parameters must be established either by use of laboratory experiments or by numerical models. Commonly, a test campaign is initiated to provide data on the motion and tension response of the mooring, which is then used for validation of a numerical model. This allows for tuning of the model so that good agreement between experiments and the numerical model is achieved for the tested sea states, mooring layout and model geometry. For instance, [13] presents a methodology for optimization of the numerical model of a WEC, and a similar one is seen in, e.g., [14,15], which considers floating wind turbine platforms, aims to prove the validity of numerical models and tunes them to available experimental tests. For many early stage WECs, however, experimental data are not available, and there is a need for understanding the mooring system behavior prior to performing tests. It often happens that the WECs experience changes during the design phases, which gives a need for new and expensive tests. It is, therefore, essential to have a model that can be used in initial studies of the potential of different mooring layouts and materials, even without access to experimental data. Where studies like, e.g., [13–15] tune the model to fit the experimental results, the present paper tends to validate a numerical tool according to the definition of validation in [16] and identifies errors and their magnitude without tuning it to experimental work. The focus is put on validating design values such as tensions and to some extent, motions, as these determine the applicability of a certain mooring layout. This should justify the use of an identical procedure for analyzing and designing initial mooring systems for other large floating WECs where no experimental data are available. For the final design, it is still recommended to perform tests and to optimize the model to get a better agreement with experiments. In order to illustrate this, an additional optimized model was produced and compared to the experimental data.

The research uses the Floating Power Plant as a case study, cf. Figure 1. This device combines wind and wave energy absorption and has been undergoing comprehensive research in small-scale tests in laboratories followed by an offshore test campaign with a 37 m-wide model. In future research, two different models will be considered: a 60 m-wide model named P60 and a full-scale 80 m-wide model named P80. This paper will consider the P60 device, illustrated in Figure 1, and the sea states from the expected deployment site at the Belgian coast.

The WEC is constructed from a floating foundation (indicated by red color in Figure 1), a wind turbine and four wave energy absorbers (blue color), which utilizes the principle of pitching bodies.

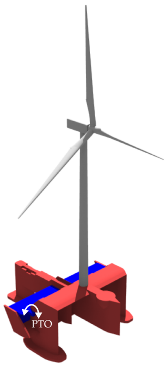


Figure 1. Illustration of the Floating Power Plant P60 device. The red color indicates the foundation, and the blue color indicates the floaters (power take off (PTO)).

The paper uses laboratory experiments performed at The Hydraulic and Coastal Engineering Laboratory at Aalborg University, Denmark, for a structure resembling a simplified model of the Floating Power Plant P60 where the wave energy PTO and wind turbine are discarded. The structure is moored with compliant synthetic lines, and the acquired results are compared to numerical results found by using the open source code NEMOH [17] and the commercial software package OrcaFlex v10.0d (Orcina Ltd., Ulverston, UK) [18].

The paper is structured with five sections including this Introduction. The following section presents the applied method in the laboratory experiments and the numerical model, and it is followed by a section presenting the results from both. Section 4 includes a discussion of the results, and Section 5 will present the conclusions and the discussion of future work.

2. Method

This section and its subsections provide an introduction of the method used for modeling of the floating structure and its mooring system. It is followed by a short description of the setup used during the laboratory experiments, and more information, description and analysis of the laboratory experiments can be found in [19].

2.1. Design and Modeling of Floating Structures with a Mooring System

There are different design standards available with requirements and recommendations for the design of mooring systems. Among the most commonly recognized are the DNV-OS-E301 [10], API-RP-2SK [11] and ISO 19901-7:2005 [12]. IEC 62600-10 [20] has been developed for mooring design for marine energy devices, while DNV-OS-J103 [21] deals with the general design of floating wind turbines. The design procedure often requires the evaluation of the extreme response and validation according to the extreme tensions in the mooring lines. A time series of an extreme sea is evaluated and the extremes identified, and the data are fitted to a statistical distribution. The requirements of the motions are not as defined as for the tensions and must be decided based on the given location, umbilical specifications, surrounding structures, etc.

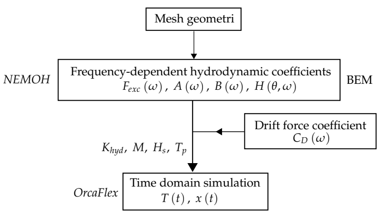
Different methods can be applied for the generation of the desired time series. One method includes the use of experimental tests, which provide highly useful results, but are time consuming and potentially expensive. A design procedure is an iterative process and often implies investigation of many different configurations before the most optimized solution is identified. A more common approach is to apply numerical models, as these are easier to automatize, more changeable and, hence, much less expensive to use. However, it is necessary to validate that the extremes found in the models resemble realistic values, for which experimental data are used. Model validation against experimental data has been done in [13] for a WEC and by several authors like, e.g., [14,15,22,23] for floating wind turbine platforms, mainly considering waves and not the coupled effect of wind and wave exposure.

Different models can be used when simulating the wave-structure and device response numerically. The most commonly-used models are the Morison approach, the boundary element method (BEM), computational fluid dynamics (CFD) or smooth particle hydrodynamics (SPH). Both CFD and SPH put high demands on the computational effort; hence, BEM or the Morison approach are more often used. Many of these models determine the wave/structure interaction, while additional code is needed to simulate the coupled response of the mooring and structure. The BEM includes the load contribution from diffraction and radiation, while the Morison approach includes drag and inertia. The choice, therefore, depends on the sea states in relation to the structure. Often, the diagram by Chakrabarti [24] is used, cf. Figure 7c.

This study uses a modeling procedure as shown in Figure 2. The open source BEM code NEMOH [17] is applied for the calculation of the wave-structure interaction in the frequency domain, which implies linear potential theory with the assumptions of an inviscid, incompressible and irrotational fluid. In addition, the theory assumes incoming waves with low steepness and also structure motions with small amplitudes [17]. Particularly, the two last assumptions are put under stress for a WEC with compliant mooring in extreme sea conditions.

The output of the BEM code is the first order wave excitation force $F_{exc}(\omega)$, the added mass $A(\omega)$, the radiation damping $B(\omega)$ and the Kochin functions $H(\theta,\omega)$. The latter is used to calculate the second order drift force coefficients using the far field formulation [25] and the code available from [26].

The results from the BEM code are coupled with the dynamic analysis tool OrcaFlex [18], which solves the coupled behavior of the mooring system and floating structure in both the time and frequency domain. The model includes the first order wave excitation force and the second order slowly-varying contribution. In this study, the Newman approximation [27] is applied for the calculation of second order motion, which implies that only the mean drift force, calculated from the first order quantities, is used to calculate the second order response. The OrcaFlex package allows for using the full quadratic transfer functions (QTFs), but this has not been considered in this study. It is stated in [18] that the accuracy of the Newman approximation decreases with shallow water depths and also if the natural frequencies are within the frequencies of the wave spectrum. The latter does not cause any concerns as it will always be attempted to design a floating system to have natural frequencies outside the wave spectrum of the extreme seas. In the present study where very large compliance is applied, this problem is even less considerable.



$F_{exc}(\omega)$	Frequency-dependent wave excitation force
$A(\omega)$	Frequencydependent added mass
$B(\omega)$	Frequency-dependent radiation damping
$H(\theta,\omega)$	Frequency- and angle-dependent Kochin function
$C_D(\omega)$	Frequency-dependent mean drift force coefficient
K_{hyd}	Structure hydrostatic stiffness matrix
M	Structure mass matrix
H_s	Significant wave height
T_p	Peak wave period
$T(t)$	Time-dependent line tension
$x(t)$	Time-dependent structure motions

Figure 2. Illustration of the numerical modeling procedure used in the present study.

OrcaFlex has the potential to calculate the response using either a radiation/diffraction approach or the Morison approach. It is expected that the use of linear potential theory to include radiation/diffraction loads will provide a certain level of accuracy, but the drag contribution will be needed, particularly at the resonance frequencies. By combining the two contributions by adding a drag element to the radiation/diffraction calculation, it is possible to obtain a better description. This requires knowledge of the drag coefficient C_d, which can either be obtained experimentally or by, e.g., CFD calculations. For cases where this is not possible, a coarse estimate must be suggested.

In order to illustrate the need and effect of this drag element, two configurations were defined for the numerical model:

- Configuration 1: Radiation/diffraction without a drag element.
- Configuration 2: Radiation/diffraction with drag elements in surge, heave and pitch.

The drag coefficients were estimated from a simplified methodology, where the model was divided into a number of simpler shapes (cubes and ellipsoids), as shown in Figure 4e, for which the drag coefficients are known and can be found in, e.g., [28]. The total drag coefficient for the model was then determined by combining the coefficients from each shape, which then provided a very coarse estimate, as it does not account for the interaction between each shape. This methodology does not require the use of experiments or advanced CFD simulations and is, therefore, applicable for initial design.

The numerical model allows for the definition of drag in all degrees of freedom (DoF) by a drag coefficient in the translational DoFs and a drag moment coefficient in the rotational DoFs. The drag moment coefficient was found with respect to the center of gravity using the simplified shapes and corresponding moment arms. The drag coefficients and areas are defined in Table 1.

Table 1. Definition of drag coefficients and areas used in the numerical model.

Parameter	Surge	Heave	Pitch
Drag coefficient, C_d	1.35	1.68	1.25
Drag area, A_d	0.545×10^3 m^2	1.92×10^3 m^2	7.63×10^6 m^5

2.2. Experimental Setup

The experiments were conducted in the deep-water basin at The Hydraulic and Coastal Engineering Laboratory at Aalborg University, Denmark, in the period of December 2015–January 2016. The wave basin was equipped with a snake-type wave maker at one end, controlled by the software package AwaSys 7 [29], and a passive absorber at the other end (gravel beach). Five resistant-type wave gauges were used for measurements of surface elevations. The array was located in front of the model, and its layout allowed for 3D reflection analysis. Acquisition, post-processing and analysis of water surface elevation and mooring loads were performed with the software package WaveLab 3 [30] and sampled at a frequency of 123 Hz. The motions of the structure were measured using the OptiTrack system [31], five reflective markers (cf. Figure 4) and four OptiTrack Flex 13 cameras. The Motive 1.9.0 software package was used for data acquisition and analysis.

The choice of scale for the model was based on the specifications of the basin. A range of full-scale sea states and a certain water depth were specified prior to the test campaign, cf. Section 2.2.2. In order to satisfy all conditions, it was found feasible to use a Froude scale of 1:64.5. This corresponds well to the recommendation in [32,33], which states that scale factors of 1:50 and up to 1:80–100 can be used when testing loadings under extreme conditions.

The basin setup is illustrated in Figure 3.

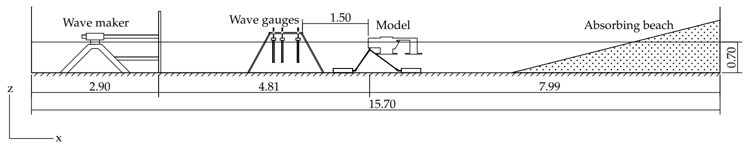


Figure 3. Illustration of the experimental setup in the wave basin. All measurements are in m.

The laboratory model resembled a simplified geometry of the Floating Power Plant P60. For simplification, it was decided to discard both the wind turbine and floaters and instead apply their mass at the foundation. In extreme events, it is expected that the loads from wind on the turbine are negligible compared to the wave loads, since the wind turbine will be shut down, and the effective area is much smaller compared to when it is in operational conditions. Despite the large wind speeds, the loads on the structure will be smaller, justifying that the turbine is neglected. Naturally, the position of the turbine mass will affect the mass moment of inertia (MoI), giving larger values. Considering the purpose of validating the numerical model, it is merely important to ensure similarity between the laboratory model and numerical model, hence discarding the wind turbine and floaters do not affect the overall purpose of the study.

Figure 4a,b illustrates the dimensions of the model in full scale, while Figure 4c shows the constructed laboratory model. In the numerical model, the laboratory layout was adapted, and a panel mesh shown in Figure 4d was constructed and analyzed. A convergence analysis was performed in order to justify the number of 1768 panels.

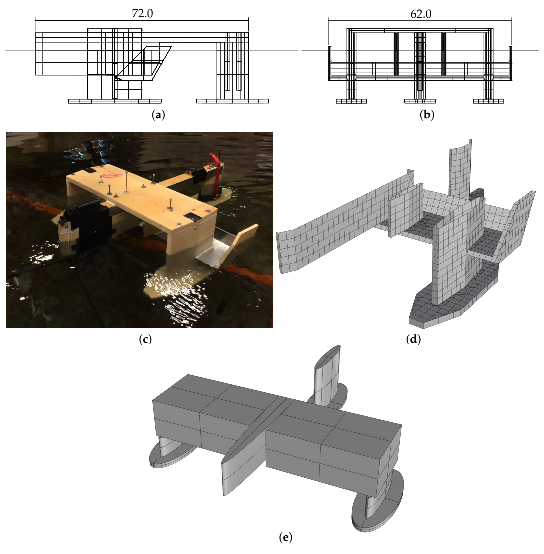


Figure 4. (**a**) Side view of the laboratory model at the prototype scale (measurement in m); (**b**) front view of the model; (**c**) picture of the laboratory model; (**d**) illustration of the mesh used in the BEM code; (**e**) simplified geometry used for the estimation of drag coefficients.

The structural parameters of the lab model were measured in the laboratory, and the values were subsequently used in the numerical model. In particular, the model mass and mass moment of inertia (MoI) are of importance in order to model the motion response correctly. Table 2 lists the measured and prototype values.

Table 2. Model specification. Note that moment of inertia (MoI) is around the center of gravity.

Parameter	Model-Scale	Full-Scale
Structure mass, m (kg)	22.4	6.0×10^6
Moment of inertia, I_{xx} (m^2 kg)	1.474	1.645×10^9
Moment of inertia, I_{yy} (m^2 kg)	2.525	2.819×10^9
Moment of inertia, I_{zz} (m^2 kg)	3.325	3.712×10^9

2.2.1. Mooring System

The mooring was applied as a standard solution with three taut synthetic lines installed as shown in Figure 5a. Despite a different shape of the stiffness curve, the linearized stiffness of the lines corresponds approximately to, e.g., Ø120 mm Bridon Superline Nylon [34]. Three load cells of the type FUTEK LSB210 50/100 lb. were installed at the connection point between the anchor and lines, and a VETEK 30 kg IP68 was positioned at the connection point between the lines and model, cf. Figure 5.

A static test determined the stiffness of the lines where each line was gradually tensioned and the elongation measured. The stiffness curve is plotted in Figure 5b, showing mild non-linearity. The mooring system in the numerical model was defined with the experimentally-measured values of line stiffness and layout. The drag and inertia coefficient of the lines were chosen based on standard values from DNV-OS-E301 [10]. Table 3 provides data on the mooring line characteristics found in the laboratory and applied in the numerical model. No structural damping in the lines was defined in the numerical model, as these values are unknown.

The mooring lines in the numerical model are divided into a number of segments and modeled as straight massless segments. All properties like mass, buoyancy, etc., are lumped to the nodes at each end, while axial and torsional properties are modeled by the segments. A higher number of segments, n, generally provides a higher level of accuracy to the model, but is paid by higher computational time. In order to find the number of segments that balances the accuracy and computational time, a convergence analysis was performed and presented in Figure 6 with the number of segments ranging from 5–30. The figure presents the error between the tensions found from each test and the test with the maximum number of segments. A relatively small error is found (less than 1%), and the results clearly converge with $n \geq 15$. The largest error is found for the tension minimums where the lines are slacked. However, the minimum tensions are not of significant importance for the design of moorings, and for $n \geq 15$, the error becomes less than 5%. Based on this, $n = 15$ was used in the numerical model, resulting in a segment length of 3 m.

Table 3. Mooring line specifications.

Parameter	Model-Scale	Full-Scale
Unstretched length (m)	0.7	46.1
Nominal diameter d (m)	0.01	0.6
Mass (kg/m)	0.04	176.0
Number of segments n (-)	15	
Segment length (m)	-	3
Drag coefficients (axial/normal) (-)	0.0/1.6	
Inertia coefficients (axial/normal) (-)	0.0/1.0	

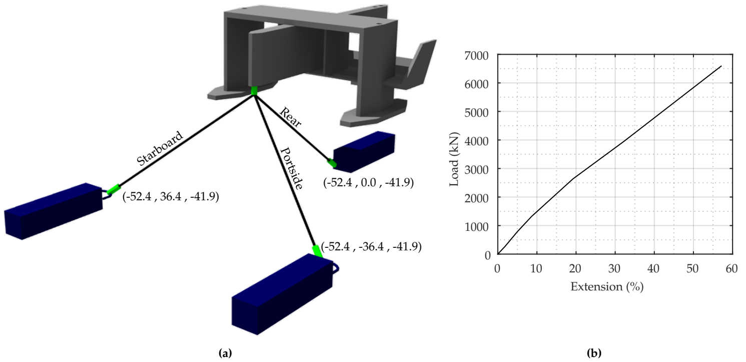


Figure 5. (**a**) Illustration of the considered mooring layout. The green shapes illustrate the load cells. (**b**) Mooring line stiffness curve determined during experiments. The values are presented at full scale.

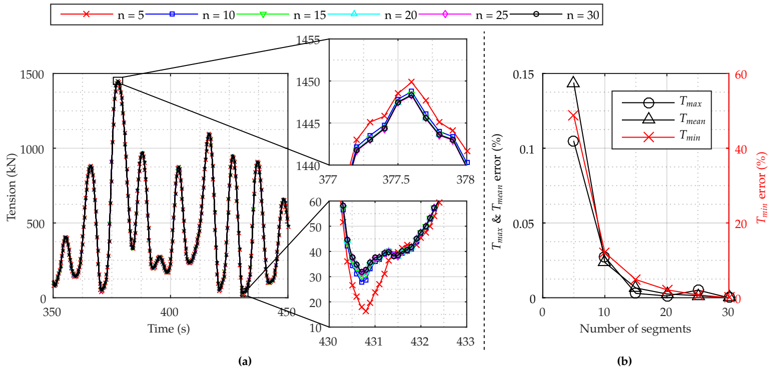


Figure 6. (**a**) Time series of segment convergence analysis with a zoom on the maximum and minimum tensions; (**b**) error of each analysis compared to the test with the maximum segment number. Note the different *y*-axes.

The mooring lines were installed with an axial pretension of approximately 315 kN, cf. Figure 8a,b. This tension results from a relatively small extension of the lines (cf. Figure 5b), which caused slacking of the lines during wave exposure.

2.2.2. Environmental Conditions

The tested sea states resemble the conditions expected at the deployment site of the P60. Three operational ($H_s = 1.3$–3.3 m and $T_p = 6.4$–8.6 s) and six extreme ($H_s = 5.1$–6.1 m and $T_p = 8.6$–13.7 s) sea states were tested in order to ensure that the expected range of wave frequencies and wave heights were covered; see Figure 7a. The sea states were simulated as long-crested waves (2D) and with a JONSWAP spectrum (peak enhancement factor $\gamma = 3.3$). The duration of the individual time series was determined so that it allowed for a number of 1000 simulated waves according to the recommendations in [33].

In addition to the irregular sea states, 23 regular wave trains were tested. Considering the application areas of wave theories as defined in [35] and plotted in Figure 7b, the x-axis shows that the waves primarily will be in intermediate water depths with some of the tested waves in deep water conditions. The y-axis indicates the variation of steepness of the waves. It is seen that all sea states can be considered non-linear, hence stressing the assumption of the linear wave potential theory.

As described, the dominating force contribution is highly dependent on the structure diameter in relation to the wavelength. Considering the boundaries defined in [24] and plotted in Figure 7c, it is seen that, when considering the entire structure as one closed body, most of the sea states are in the load regime where diffraction is dominant. For the longest waves, the model will be in a regime where other load contributions have an effect. The investigated structure has a complex geometry where water can pass through the body, which truly does not follow the theory of [24]. Considering, e.g., just the width of each of the vertical columns (cf. Figure 4), the force might be dominated or affected significantly by more than diffraction.

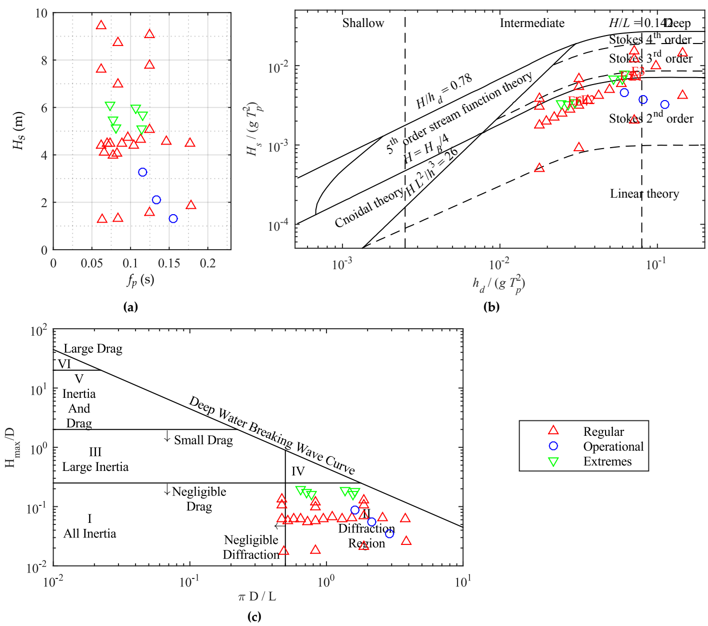


Figure 7. (**a**) Diagram of the sea states simulated in the wave basin; (**b**) the sea states plotted against the depth condition (x-axis), wave steepness (y-axis) and application areas of wave theories as defined in [35]; (**c**) the sea states plotted against wave force regimes as defined in [24], with D being the total width of the structure and not considering the open spaces.

3. Results

This section presents the results obtained from the numerical model and experimental work. These results will be compared to each other and used to investigate the validity of the numerical model.

3.1. Quasi-Static Results

A quasi-static test was carried out for validation of the mooring layout and modeling of line tension. The model was displaced in the surge DoF at a low velocity in order to avoid any dynamic effects. Due to the low velocity, the only restoring force present in the system resulted from the stiffness of the mooring lines. Figure 8a,b plots the measured loads in respectively the starboard and rear mooring line. The experimental results show some scattering of the results, which can be explained by the fact that the model was displaced manually. When the model is displaced, the mooring point will displace vertically as shown in Figure 8c, and this displacement is very sensitive to the physical handling of the model. For a perfect test with no displacement in any other DoF than surge, the scatter would not be expected. Prior to the measurement, one load cell was damaged; therefore, results from the port side line cannot be presented. It would have improved the reliability of the tests since both lines should have shown similar results.

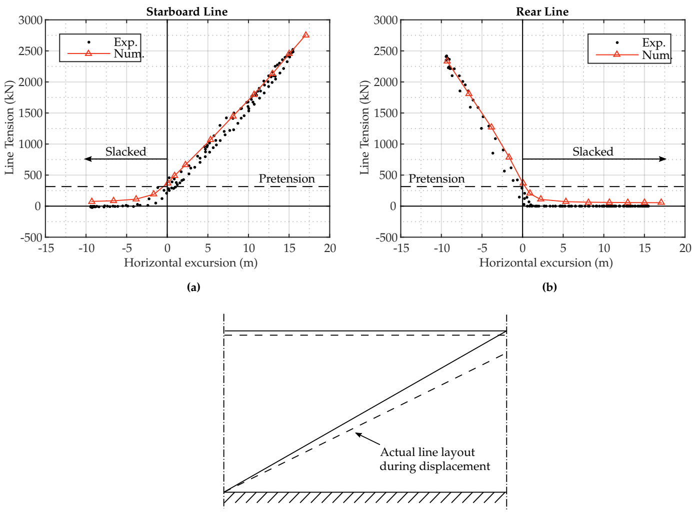


Figure 8. Quasi-static test results for experimental (Exp.) and numerical (Num.) line tension in the starboard (**a**) and rear (**b**) line; (**c**) illustration of the line layout during surge motion.

When observing Figure 8, it is seen that the numerical results are within the scatter of the experimental results, and therefore, this illustrates the similarity of the mooring stiffness in the two models. The difference, which is seen to be approximately constant, can be explained by the tactile inaccuracy in the physical handling of the model as described previously. The most dominant difference is seen when the lines are slacked where the numerical tension is higher than the experiments. In the laboratory, no tension was measured in the completely slacked lines, while some tension was found in the numerical model. This is because the lines are buoyant and give some tension in the connection point. This effect was not seen in the laboratory, possible due to the presence of load cells.

3.2. Decay Test

A decay test was used to compare natural frequencies of and damping in the two models. The test is performed by giving a displacement in one DoF, releasing the model and allowing the motion to decay. Due to the complexity of the model, it is difficult to activate only one DoF at a time, and hence, it is difficult to find natural frequencies for all DoF. The test presented a coupled frequency at 0.078 Hz where the structure was activated in surge, heave and pitch. It was possible to activate only the surge DoF in both numerical and experimental tests, even without applying any restraints on the other DoFs. This means that even though the primary motion was in the surge DoF, some motions were also seen in heave and pitch, cf. Figure 9c. Since the scale of the experiments was 1:64.5, the magnitudes of the heave and pith motions are negligible compared to the surge (less than 4 mm in the laboratory). It was not possible to obtain any reliable results for heave and pitch only. The surge decay test is illustrated in Figure 9a for the configurations with and without the drag element.

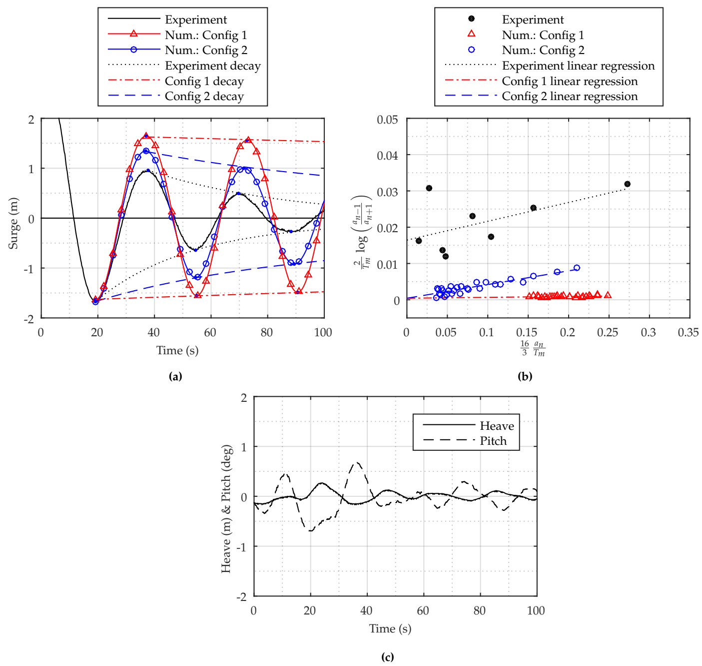


Figure 9. (**a**) Free surge decay test from experiments and Configurations 1 and 2 (with and without the drag element); (**b**) determination of linear and quadratic damping coefficients using linear regression [36]; (**c**) heave and pitch motion during the surge decay test.

In the decay test, the structure moves at its natural frequency f_n, and the motion is dependent on the structure mass M, added mass A, radiation damping B, hydrostatic stiffness K_{hyd} and mooring stiffness K_{moor}.

The peaks and troughs of the motion in Figure 9 were detected and used to calculate the natural frequency as the average of the values. The first oscillation was not considered in order to avoid the influence of releasing the model. Since the structure itself does not provide any hydrostatic stiffness in surge, the natural frequency is mostly dependent on the mass, added mass and mooring stiffness. The mass defined in the numerical model is based on the mass measured in the laboratory and, therefore, does not affect the comparison. When considering the quasi-static test, it was determined that there was good agreement between numerical and experimental stiffness. Therefore, the difference in the two natural frequencies mostly describes any inaccuracies in the calculated added mass in the BEM solver. Considering the results in Table 4, it is however found that the error is approximately 7% for Configuration 1, while it is 4% for Configuration 2, and as such, the results can be considered acceptable.

Table 4. Measured and calculated natural frequency, linear and quadratic damping in surge, together with the relative error.

Parameter	Experiment	Num.: Config 1 Value/Relative Error	Num.: Config 2 Value/Relative Error
Natural surge frequency, f_n	0.0305 Hz	0.0284 Hz/6.9%	0.0293 Hz/3.9%
Linear damping, p_1	0.0164 s^{-1}	0.0004 s^{-1}/97.6%	0.0004 s^{-1}/97.6%
Quadratic damping, p_2	0.0519 m^{-1}	0.0024 m^{-1}/95.4%	0.0378 m^{-1}/27.2%

The decay test can additionally be used to illustrate the damping of the system, which is the combination of a linear and quadratic contribution. Figure 9 clearly shows that the numerical model highly underestimates this fact, with the largest difference occurring for Configuration 1. In [36], Equation (1) is used to determine the linear and quadratic damping coefficients p_1 and p_2.

$$\frac{2}{T_m}\log\left(\frac{a_{n-1}}{a_{n+1}}\right) = p_1 + \frac{16}{3}\frac{a_n}{T_m}p_2 \tag{1}$$

where T_m is the natural period and a_n is the amplitude of the n-th oscillation. When plotting the left-hand side of the equation against the right-hand side, it is possible to use linear regression to calculate p_1 and p_2. This is illustrated in Figure 9b, and the results are listed in Table 4. Due to a limited number of oscillations in the experimental decay test, the linear fit is not as good as for the numerical results.

When considering Table 4, it is clear that the numerical model in Configuration 1 highly underestimates the quadratic drag arising from viscous effects. An error of 95.4% is seen, which is reduced to 27.2% in Configuration 2 by adding the drag element. Since no additional linear damping is added to the model, similar results are obtained from the two configurations with a relatively high error of approximately 98%. Section 3.5 presents an optimized model with additional linear damping in order to illustrate its influence on the obtained results.

3.3. Regular Sea States

A total of 23 regular sea states were tested (cf. Figure 7a) and used to determine the response amplitude operators (RAOs) for the motions, tensions and the mean drift motion.

3.3.1. Motion Response

The motion was measured in surge, heave and pitch. and their RAOs are plotted in Figure 10, together with the numerical results.

Considering the results from Configuration 1 (Figure 10a–c), there is some agreement with the experiments for most of the frequency range in all DoFs, showing the similar shape and amplitude of the RAOs. Both numerical and experimental RAOs show a peak at the coupled motion frequency at 0.078 Hz, but at this frequency, the most severe difference is observed. Due to resonance motion and lack of damping, most stress is put on the linear potential theory, and the numerical model highly overestimates the motion amplitude. For Configuration 2 (Figure 10d–f) with a drag element in all DoFs, better agreement between the experimental and numerical results is obtained, even at the peak of the RAOs. Still, there is an overestimation, but the error is decreased.

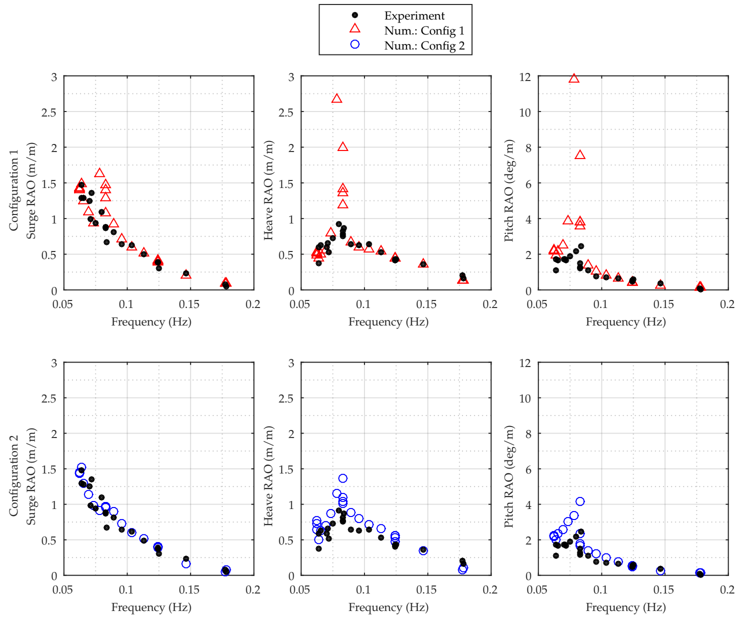


Figure 10. Measured and calculated motion response amplitude operators (RAOs) in surge, heave and pitch.

3.3.2. Mean Drift

Figure 11 presents the results for the calculated and measured mean drift motion. In the numerical and experimental model, the mean drift during wave exposure is defined as the mean displacement from the static position. Considering the entire frequency range, both over- and under-estimations are seen, and due to the large error in the motion amplitudes at the peak frequency, Configurations 1 provides paramount drift motion errors at this frequency. A very large underestimation is seen with even negative drift, while the drift at the remaining frequencies shows a similar trend as for the experiments. Despite the better resemblance with experiments, Configuration 2 also shows some deviation from the experimental results. Considering Figure 11, it is seen how the drift motion is overestimated at the lowest frequencies and underestimated for many of the higher frequencies, but with a trend following the trend of the results. Despite this error, the motion amplitudes were shown in Figure 10 to resemble the experiments. Therefore, the error in the drift has proven not to affect the motion amplitude significantly.

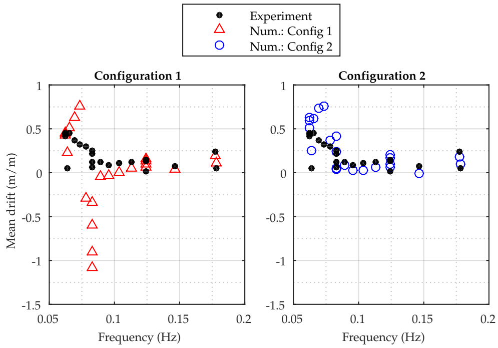


Figure 11. Measured and calculated drift motion RAOs for the two configurations.

3.3.3. Tension Response

Considering the difference in measured and calculated motion response, especially in the drift motion, some error was expected in the calculated tensions, which are plotted in Figures 12 and 13. Overall, the numerical model shows good agreement with the experiments, but with a high overestimation of the tension around the peak frequencies in Figure 13, a consequence of the high overestimation of the motions. As expected, this is most dominant for Configurations 1 (Figure 13a,c). For the low frequency waves, the tension amplitudes are underestimated, but considering Figure 12, which presents a direct time series comparison between experiments and Configuration 2, the peaks of the tension time series are similar for both models, indicating that the numerical model describes the maximum values well. A difference can be observed between the tension troughs in the figure, which explains the smaller amplitudes.

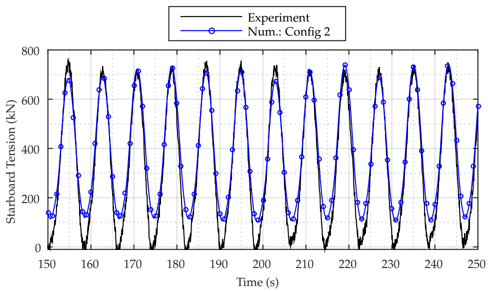


Figure 12. Direct comparison of measured and calculated tension time series for a regular sea state with $H = 5.1$ m and $T = 8.0$ s.

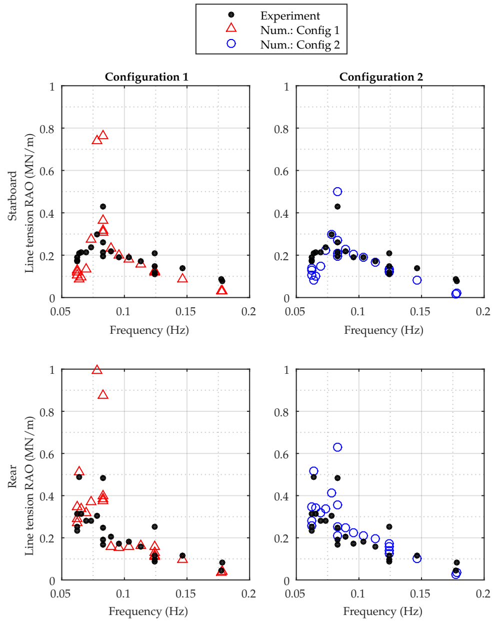


Figure 13. Measured and calculated tension RAOs in starboard and rear lines.

3.3.4. Influence from Wave Height

As seen in Figure 7a a number of the regular sea states covered a similar wave frequency with varying wave heights in order to investigate non-linearities in the response of the structure. Figure 14 illustrates the motion RAOs and mean drift against the wave height for these experiments and Configuration 2. For the surge DoF, the numerical and experimental results show similar tendencies, but the numerical model overestimates the response. For the shortest waves ($f = 0.12$ Hz), there is good agreement independent of the wave height, but a large error is seen at 0.83 Hz for the lowest wave height. This frequency corresponds to the peak seen in the heave and pitch DoF (cf. Figure 10). The numerical response is seen to be almost linear in surge, but the experiments show a non-linear influence from wave height.

The response in the heave DoF illustrates non-linearity, and it is highly influenced by the wave frequencies. The tendencies found in the experiments and the numerical model are similar, and the magnitude of the difference seems independent of the wave height. A similar observation is found for the pitch DoF, with good agreement for the shortest waves where the smallest motions are present. More dominant errors are found at the natural frequency and for the longest waves where a large response will occur.

The figure clearly indicates that the calculated RAOs are highly influenced by the incoming wave height, and not linearly dependent on it. The magnitude of the deviation between numerical and experimental results seems more dependent on wave frequency than wave height.

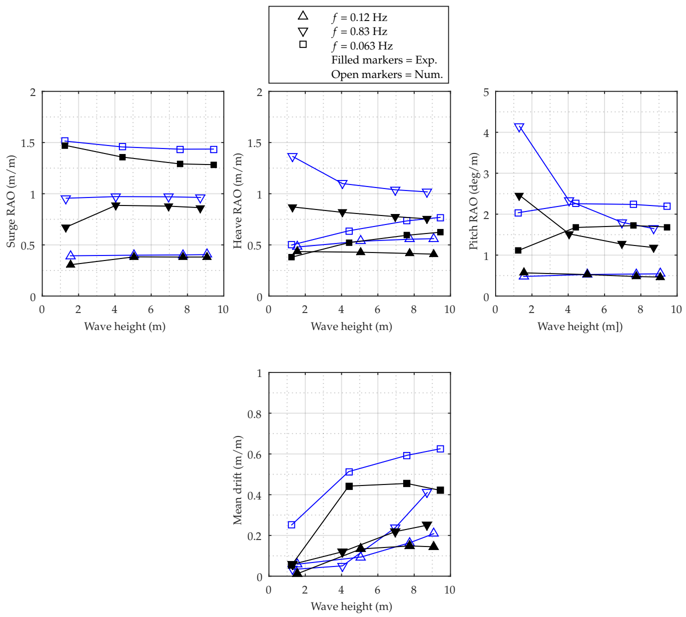


Figure 14. Experimental and numerical RAOs for varying wave heights.

3.4. Irregular Sea States

The extreme values of the irregular sea states were found and presented in Figures 15 and 16. In this study, the maximum simulated and measured value is considered together with the most probable maximum, cf. Equation (2), which is commonly used for design in, e.g., [10].

$$T_{MPM} = \mu + \sigma\sqrt{2\ \ln(N)} \tag{2}$$

where T_{MPM} is the most probable maximum (MPM) tension in N waves, μ is the mean tension and σ is the standard deviation. This study uses the MPM value in 1000 waves.

Since the numerical and experimental wave time series are not identical, but have similar frequency domain parameters, it is not possible to compare the maximum values directly. Considering Figure 15, there is good agreement in both configurations for the two tests with highest peak frequencies (operational conditions). These also have the smallest wave height and are the most linear waves. This agreement is seen in both the rear and starboard line. For the larger sea states, the error for Configurations 1 begins to increase significantly. For the considered sea states, it is not possible to show results for these two configurations for the sea states with frequencies lower than 0.1 Hz (the extreme sea states). Under these conditions, the motions became so large that the solver could not find a solution. The largest error is found for the sea states with peak frequencies around 0.12 Hz. Considering the tested sea states in Figure 7, these sea states also correspond to the steepest sea states. Still, the maximum deviation for Configuration 2 is 26% for the T_{MPM} for the rear line, while it is only 11% for the starboard line.

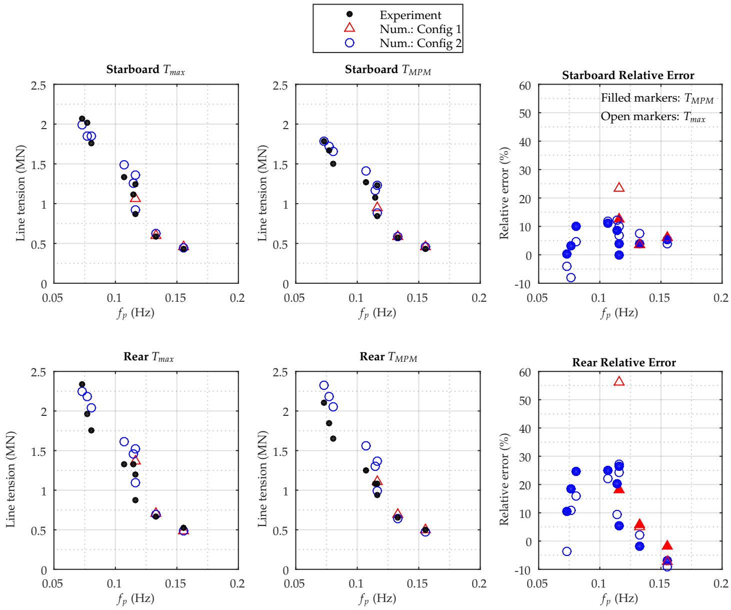


Figure 15. Comparison of extreme tensions for the irregular sea states.

The extreme motion of the device is also of interest, and the results are plotted in Figure 16. Despite the good agreement seen in the previous figure, a larger deviation is found. The numerical model particularly overestimates the pitch motion. Even though some quadratic damping is added in the pitch DoF, it still appears that more damping is needed. For Configuration 1, the relative error exceeds 100%, with a maximum of 150%. The drag element in Configuration 2 decreases the error to a maximum of 78%, which is still a significant overestimation. The smallest error is found for the operational sea conditions in deep water, with a smaller wave steepness.

The surge motion is overestimated with a maximum of 32% in the most extreme situation. This sea state is also the sea state in most shallow water and, according to Figure 7, also the sea state where the diffraction contribution is smallest and drag and inertia most important. The surge maximums are underestimated for the sea states with the highest frequency.

The heave motion causes the smallest relative error, with a maximum absolute value of 27%.

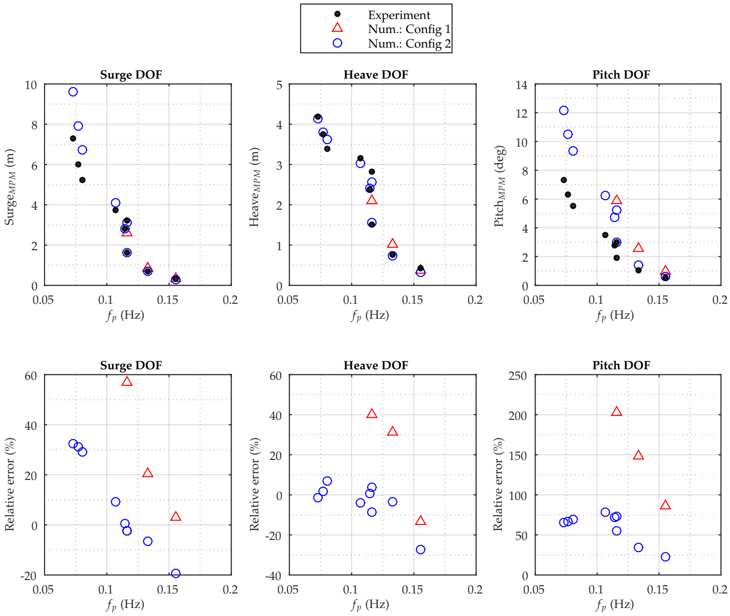


Figure 16. Comparison of extreme motions for the irregular sea states.

3.5. Optimized Model

As stated previously, the objective of this paper is to validate that a numerical model can be used in an initial design procedure, where no experimental data are available for optimization and tuning of the model. The results presented in the previous sections indicate that the structure can be modeled with some overestimation of response compared to the experiments. Considering Figure 9, the deviation appears to be caused by especially insufficient damping. Some quadratic damping has been added by the drag element, but the figure showed significant underestimation of the linear damping. Figure 17a illustrates a decay test where 370 $\frac{\text{kN}}{\text{m/s}}$ linear damping has been added in surge.

The additional damping clearly results in a better agreement between the surge motion in the experiment and model, which is also illustrated in Figure 17b, where the linear regression now indicates a minor difference between both linear and quadratic damping. For the optimized model, the additional linear damping results in a coefficient $p_1 = 0.0171\ \text{s}^{-1}$, which corresponds to a relative error of 4.3% from the experiments.

Since no decay tests are available for heave and pitch, it is more difficult to tune these values. The work in [13] presented a method that can be used, but the present study made a coarse sensitivity analysis to find values of linear damping in heave and pitch, which improves the model results for the regular tests. By adding 1000 $\frac{\text{kN}}{\text{m/s}}$ and 4000 $\frac{\text{kNm}}{\text{rad/s}}$ in respectively heave and pitch, the model showed better agreement with experiments. Figure 18 presents the motion RAOs for the optimized model. Clearly, the figure indicates that the additional linear damping is the main cause of difference between Configuration 2 and the experiments. By adding the damping, also the mean drift is better estimated.

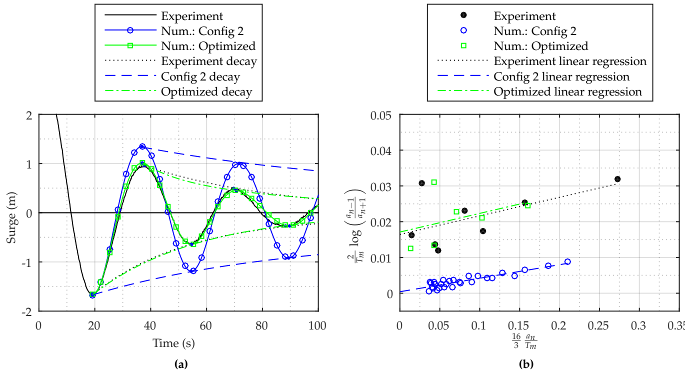


Figure 17. (**a**) Decay test from experiments, the model with drag elements (Configuration 2) and the optimized model with additional linear damping; (**b**) determination of p_1 and p_2 from the experiments, Configuration 2 and the optimized model using linear regression.

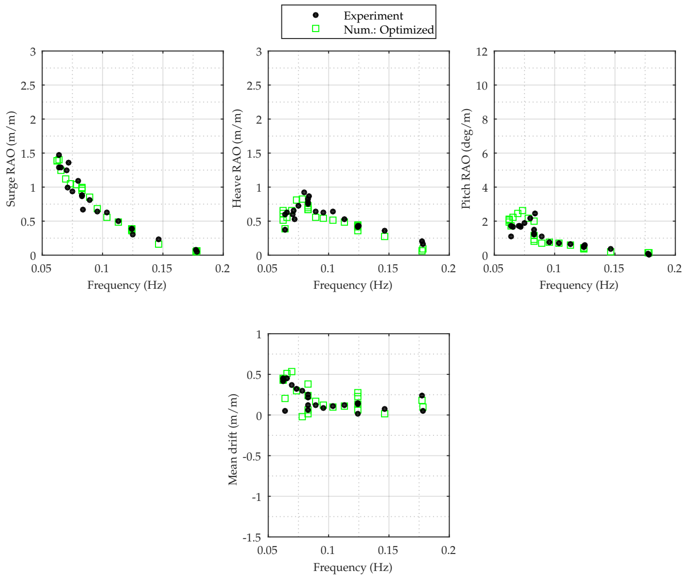


Figure 18. Comparison of the optimized numerical model with experimental data for all regular sea states.

The results clearly indicate the importance of tuning the model. This, however, requires the use of experiments, and as mentioned, this is not always available in initial design. The quadratic damping in this study has been added based on the geometry of the device and not experimental data, and therefore, Configuration 2 resembles a model that can be used initially to analyze the mooring system before a final solution is chosen, experimental data or more sophisticated models are produced and the numerical model is optimized.

4. Discussion

This study has used a large set of experimental data to validate a numerical model to be used in the initial design of mooring systems. It has been attempted to ensure similarity between model and laboratory in order to compare results, but sources of error are present during experiments. As stated previously, a small scale of 1:64.5 puts high demands on measurement precision, model manufacturing, anchor positioning, etc., and even minor uncertainties on measurements have high influence on the full-scale values. Similarly, in some of the tests, e.g., the quasi-static test, the model was displaced manually, which eventually will introduce some inaccuracy. The wave basin at Aalborg University is equipped with a passive absorber, and the maximum reflection during the test campaign was approximately 20% for the longest waves. The reflective waves were not modeled in the numerical model and could provide some divergence between model and experiment. Finally, it must also be mentioned that the anchors used in the test campaign were relatively large (cf. Figure 5) and could potentially have influenced the wave field around the model, at least for the longest waves. During the tests, it was attempted to minimize the influence of all sources of error, and hence, they are assumed not to affect the reliability of the results.

A quasi-static test was performed (cf. Figure 8) for validation of the static behavior of the numerical model, its capability of modeling the mooring tension and stiffness and the mooring layout. Good agreement was seen and proved that a good resemblance between the numerical and experimental layout was present and that the defined stiffness of the lines had been achieved. This was additionally seen from the decay test in Figure 9. Close values of the surge natural frequency were achieved, also indicating that the calculation of added mass was correct. Due to the complex geometry, it was not possible to generate reliable decay tests for heave and pitch without activating more than one DoF at a time. The added mass can also affect the natural frequencies, but since it is also indicated to be modeled correctly in surge, it is expected to show similar results in heave and pitch. Considering the behavior found in the dynamic tests, this seems to be correct.

The decay highlighted a significant underestimation of the damping in the numerical model, both for linear and quadratic damping. By adding a drag element, the error of the quadratic damping was decreased, but naturally, the linear damping still showed a significant error. Considering the difference in Figure 9 and the influence of adding a drag element, it is clear that CFD simulations and experiments are needed in order to determine drag coefficients and produce a better model of the system. This was out of scope of this study, but is crucial to consider before a final mooring design can be achieved. The underestimation of linear damping in the model was illustrated in Figures 17 and 18, where additional damping had been added and better agreement was found for the RAOs and drift motion. A potential source of this error is the presence of the thin heave damper plates in the model, which need to be solved using either a very fine mesh or a thin plate approximation, which is not included in the present version of NEMOH. This capability is included in other software packages like WAMIT [37] and will be implemented in the next releases of NEMOH. The present study did not investigate this problem further, but its influence will be investigated in future research.

Based on a range of regular wave tests, the RAOs were defined in Figure 10. Agreement between the experimental and numerical results was found in most of the frequency range for both model configurations. Some error was seen, particularly for heave and pitch at the natural frequency, and most dominantly when no drag elements were applied. By introducing drag into the pitch and heave DoF, significantly better results were obtained. At the peak frequency, the most outspoken motions will occur and hence put most demand on the introduction of drag, but it is also at that frequency where most stress is put on the assumptions of the linear potential wave theory, which is assuming small amplitude wave and body motions.

The drift motion showed the largest error in the model. The model uses the Newman approximation, which is known to be poor in shallower water depths and when resonant motions are present. The use of full QTFs potentially could improve the model and should be investigated in future research. The calculated drift coefficients are based on the first order results from the linear potential theory. Here, the motions are used to calculate the drift, and these are overestimated due to the lack of the drag and damping. From the optimized model in Figure 18, it is seen that by adding more damping, the motions are better described, and a similar observation can be made for the drift.

From the irregular sea states, the extreme tension and motions from experiments and numerical code were compared, and the code showed a maximum relative error of approximately 26% for the line tension. Considering that this paper focuses on the applicability of the numerical model at an initial design phase where no experimental data are available and no tuning performed, this error is within an expected and reasonable range. Based on the objective of the study, the capability of the code to model the line tensions is, therefore, validated. The motion of the device was highly overestimated in pitch and probably needs more damping than what was introduced by the drag element. This was particularly shown in Figure 17. The overestimation of extreme surge might be a result of the overestimation of drift. Despite the higher relative error of the motion, the relative error of tension was at an acceptable level.

The study has shown that a relatively reliable numerical model can be constructed and used in initial analysis to model extreme design values for mooring lines, even without tuning it to experimental data, but only adding a drag element determined by a simplified method without the use of experiments. The analysis provided an understanding of the errors and limitations, which can be taken into account in future analysis of the structures. It is expected that this approach can be used for other similar WECs as presented in a previous section. The results naturally need to be treated with caution, and the best description of the device is achieved by using experimental data to optimize the model.

Since the model overestimates loads and motions, it could be highly beneficial to tune the model to lower the loads and, thereby, decrease the need for strength of the lines. When considering that design standards like, e.g., [10] or [11] introduce safety factors in the range of 1.4–1.67, much safety is now in the system. From a survivability point of view, this can be desirable, but it will not help in reducing the cost of the moorings. For a final and complete design, it is therefore desirable to optimize the model as much as possible. For initial design, use of the model in this paper will provide usable results that can be applied in further investigation of mooring systems.

5. Conclusions

The present paper presented the outcome of an experimental test campaign and used the results to validate a numerical model, constructed by use of the BEM code NEMOH and the time domain mooring analysis tool OrcaFlex. Focus was put on the expected sea conditions for a specific deployment site and sea states covering the expected frequency and wave height range. This included both operational and extreme conditions. The aim of the paper was to investigate the potential of the codes to calculate the extreme response for line tension in particular and, hence, their applicability in initial mooring design for large WECs. In addition, the work presented the influence of applying drag elements with the use of simplified theory, and not laboratory or CFD results, in order to improve the BEM model. The focus was to achieve an understanding of the obtained error when considering the extreme response, in order to use the model in initial design phases where, e.g., experimental data are not available. The main error was shown to be caused by underestimation of the damping in the model. By adding a drag element without the use of experimental tuning, it was possible to improve the quadratic damping significantly, but the linear damping was still underestimated. The results are expected to be applicable to similar devices with passive mooring and where the PTO is in safety mode during extreme events. The model showed an error that can be accepted in initial phases; however, the study highlighted the inaccuracies from parameters such as linear and quadratic damping, and these must be minimized before a complete and final mooring design can be achieved.

Acknowledgments: The present study was funded by the Energy Technology Development and Demonstration Program (EUDP) and its project "Mooring Solutions for Large Wave Energy Converters" (Grant Number 64014-0139). The authors wish to acknowledge the Floating Power Plant for input on the experimental work.

Author Contributions: J.B.T., F.F. and J.P.K. defined the outline of the study and the experimental work in a shared effort. J.B.T. mainly performed the experimental work with input from F.F. and J.P.K. J.B.T. analyzed the data and performed the numerical analysis. J.B.T. drafted the paper, while F.F. and J.P.K. provided important input and review needed for finalization of the paper.

Conflicts of Interest: The authors declare no conflict of interest.

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Paper F

On Mooring Solutions for Large Wave Energy Converter

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The paper has been published in
EWTEC, ISSN: 2309-1983 (1141), 2017.

On Mooring Solutions for Large Wave Energy Converters

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***Abstract*—The present paper describes the work carried out in the project *'Mooring Solutions for Large Wave Energy Converters'*, which is a Danish research project carried out in a period of three years from September 2014, with the aim of reducing cost of the moorings for four wave energy converters and improving the applied design procedure. The paper presents the initial layouts and costs and illustrates which solutions could potentially reduce cost. Different methods for analysis of the systems were applied, ranging from simple quasi-static analysis to full dynamic analysis and experimental work. The numerical methods were compared to the experimental data, and results showed significant underestimation of tensions in the quasi-static model while reasonable overestimation was found in the dynamic analysis even without major tuning of the model. The dynamic analysis has then been implemented in a meta-model based optimization process with the aim of optimizing the mooring layout for each WEC according to cost of the systems.**

***Index Terms*—Wave Energy Converters, Mooring, Station Keeping, Synthetic Lines, Dynamic Analysis.**

I. Introduction

The energy sources used throughout the world are transforming into renewable energies as a consequence of the rising energy demand and an increasing desire for cleaner and more reliable solutions. There are different types of renewable sources, such as solar, wind and ocean energy. Wind and solar energy sources are well-established and widely used in many countries, while the large energy source generated by waves is still not at a stage where it can contribute to the worldwide energy production. Despite a major focus during the last decades, the wave energy sector is still at an early pre-commercial state; a consequence of the existence of several problem areas that need to be solved before the sector can be incorporated in the commercial market [1], [2]. The main factor being the need to lower the levelized cost of energy (LCOE) for wave energy converters (WECs) so that the sector can supplement the existing renewable sectors and compete with the non-renewable.

In [2] the potential for cost reduction was investigated, and it was found that the following areas could provide contribution:

- Improvement of structure.
- Improvement of operation and maintenance (O&M).
- Improvement of energy yield.
- Improvement of station keeping.
- Improvement of grid connection.
- Improvement of installation.

Naturally, these topics should all, therefore, undergo research and optimization, but in [3], particularly mooring was found to be a significant potential for cost reduction. In e.g. [2], [4], [5] the cost of moorings for WECs was stated to be approximately 10% of the total structural cost and with estimations of up to 30%. This could be explained by the fact that many of the WEC developers have used existing and traditional solutions from other offshore sectors like the oil and gas (O&G) sector. However, there are many differences between this sector and the wave energy sector. WECs are generally much smaller than oil and gas structures and expe-

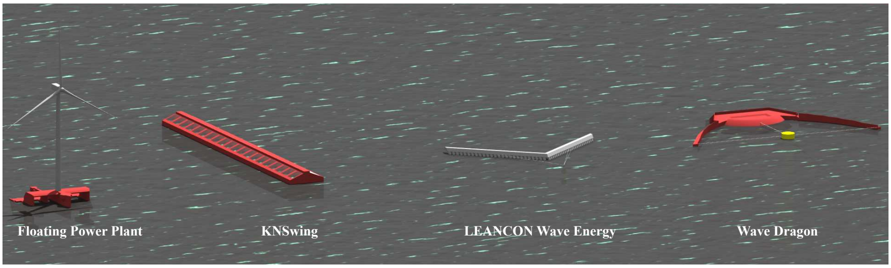

Fig. 1. The four partner WECs in the present research project: Floating Power Plant, KNSwing, LEANCON Wave Energy and Wave Dragon.

rience much different responses. This is particularly the case since oil and gas sectors cannot tolerate large motion while this can be accepted and sometimes even desired for WECs. The existing solutions were, therefore, also stated in [2] as having very little potential for cost reduction, and use of innovation and new solutions was stated as a necessity in order to reduce the cost, as done by e.g. [6], [7].

In addition to the high cost, it is also paramount to notice that the rate of mooring failures for WECs have been high [5]. For instance, the Danish wave energy sector has experienced several strandings of WECs during testing due to failure of the mooring systems [8], [9]. This has not only caused crucial damage to the devices but also meant loss of finances and limited further investigation and improvement.

As a result of these problems related to mooring of WECs, a Danish research project was initiated in September 2014. Despite having a variety of different devices with different sizes and energy absorption principles, several of the Danish WECs have similar characteristics and are expected to be deployed under similar conditions. This project focuses on WECs that can be considered relatively large and is, therefore, titled *"Mooring Solutions for Large Wave Energy Converters"*. The work was a cooperation between seven partners, including:

- Two universities:
  - Aalborg University, Denmark
  - Chalmers University of Technology, Sweden
- One commercial design supplier:
  - Tension Technology International, UK [10]
- Four WEC developers:
  - Floating Power Plant, Denmark [11]
  - KNSwing, Denmark
  - LEANCON Wave Energy, Denmark [12]
  - Wave Dragon, Denmark [13]

The main purpose of the project was to improve the mooring solutions for each of the WECs, considering not only cost but also reliability of the systems. An objective was to ensure that a final solution could be certified according to a design code like DNV-OS-E301 [14], API-RP-2SK [15], ISO 19901-7:2005 [16] or IEC 62600-10 [17]. Additionally, the project aimed to gain knowledge into mooring design procedures, alternative solutions to what is initially considered and to gain experience that can be used in other applications and for other WECs.

The present paper will describe the work that has been carried out during the project and present some of the main findings. The paper is structured with seven sections including this introduction. Section II will provide a short introduction to the partner WECs, followed by Section III which describes the mooring systems that were initially considered for each WEC including their cost and design procedures. Section IV describes the preliminary analysis that was performed in order to identify potential solutions, and Section V describes the analysis of applicable design procedures and tools. Section VI describes the full analysis of the solutions and the applied optimizations procedure, and the paper is concluded with a section of discussion and perspective.

II. Wave Energy Converters

There are a considerable amount of different wave energy concepts worldwide today; some early stage and recent while others are older and have been undergoing development throughout a number of years [18]. Traditionally, the devices are characterized according to energy absorption technology as either oscillating water columns (OWC), wave activated bodies or overtopping devices [19]. The WECs included in this project use several of the mentioned technologies, but common for all is that they are considered as relatively large structures and are going to be deployed under similar environmental conditions (shallow to intermediate water depths, cf. Table I).

TABLE I
Extreme environmental conditions for the four WECs.

WEC	h_d	100 year extreme H_s	T_p	$v_{current}$	v_{wind}
Floating Power Plant	30 m	6.6 m	11.5 s	1.3 m/s	33.0 m/s
KNSwing	40 m	9.9 m	14.1 s	1.0 m/s	39.9 m/s
LEANCON	25 m	8.3 m	12.9 s	1.5 m/s	34.0 m/s
Wave Dragon	25 m	8.3 m	12.9 s	1.5 m/s	34.0 m/s

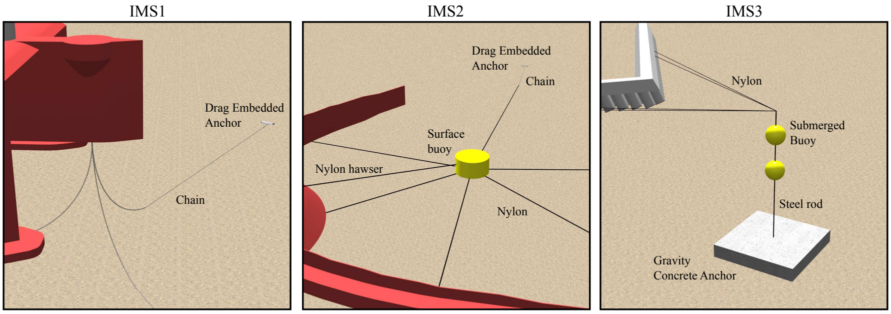

Fig. 2. Illustration of the three mooring systems considered for the partner WECs prior to the project.

In addition to this, they are all equipped with PTO systems that are not influenced by the mooring systems. This means that the moorings are considered as passive [5]. In spite of these similarities, they all use different mooring systems.

Figure 1 illustrates the four partner WECs, and before describing the mooring system in the next section, the following subsections will provide a brief description of each WEC.

A. Floating Power Plant

Floating Power Plant is a device combining energy absorption from both wind and wave. The concept has been undergoing comprehensive research since it was introduced in 1995. In 2008, a 37 m wide model was deployed in Danish waters, providing a large database of data on the system and environment. Since the test campaign, the system was redesigned, and currently the goal is to deploy an 80 m wide model at a UK site. This project investigates a 60 m wide model previously planned for deployment at the Belgian coast. The Floating Power Plant is constructed of a floating foundation, on which a wind turbine is located and a system for harvesting of wave energy. This is done by using the concept of pitching bodies with four floaters attached to the foundation. During storms, the PTO is disabled and the floaters are ballasted so that their natural frequencies are far from the wave frequency. The model is illustrated in Figure 1.

B. KNSwing

The KNSwing WEC is an early stage model, which has been undergoing several laboratory test campaigns in Denmark and Ireland, cf. e.g. [20]. The structure is a ship-like model equipped with a number of OWC chambers. The full-scale model is 240 m long and planned for deployment at the Danish part of the North Sea.

C. LEANCON Wave Energy

LEANCON Wave Energy is a device using the OWC principle for harvesting of wave energy. It is a V-shape structure with two rows of OWC tubes on each arm. During storms, the WEC is equipped with a storm protection mode that closes the OWC tubes and fills the model with air. The draught of the structure is thereby reduced, which expectedly reduces the loads on the structure. The device is planned for deployment in the North Sea, but the first commercial scale device will be located at the Danish Wave Energy Center (DanWEC) [21] with a width of approximately 120 m. The device has previously been tested at Aalborg University and in 2015, a 1:10 scale device underwent three months of offshore testing.

D. Wave Dragon

Wave Dragon is an overtopping device as illustrated in Figure 1. The structure is equipped with air chambers, which can be used to adjust the draught of the system. In stormy weathers, the air chambers are filled with water, and the model lowered below the SWL. The device has been tested comprehensively under several campaigns. Aalborg University has carried out different experiments in order to investigate the energy conversion, structural response and mooring tensions. In 2003-2005, the structure was deployed at the Danish test site, Nissum Bredning, for large scale testing. The next and commercial model will be a 1.5 MW structure with a maximum width of approximately 150 m. The structure will be deployed at DanWEC at first, but is also planned for the North Sea.

III. Initial Mooring Layouts and Design Procedures

In the first study of the project, cf. [22], the initial layout of each mooring system was assessed and the applied design procedures were stated. Many similarities were seen between the WECs and many of the same conclusions could be made from each of them. By first considering the layouts, three different solutions had been applied to the WECs, each illustrated in Figure 2. The layout was constructed 1) as a traditional catenary turret system, composed of mooring chain and drag embedded anchors (denoted initial mooring system 1 (IMS1)), 2) as a single point mooring (SPM) system with a surface buoy, a nylon hawser and mooring lines composed of nylon and chain which are anchored to the seabed by drag embedded anchors (IMS2) and finally 3) as a single anchor leg mooring (SALM) system with two submerged buoys, a deformable tether and nylon lines as hawser (denoted IMS3). From the assessment, the following main conclusions were drawn:

- Much of the experience from the existing O&G sector was applied.
- Some more novel concepts were applied.

Considering the layouts in Figure 2, it is clear that despite all being large structures, having passive mooring requirements and comparable environmental conditions, the applied moorings are much different.

In addition to considering different layouts, it was also found that the procedures used to design the systems had some differences.

Similar for the devices is that they have all been undergoing research by now and have all been tested experimentally. This has provided some indication of expected response and line tensions, but none of the devices had undergone full dynamic analysis at the time. The main differences in the design procedures were as follows:

- Difference in extreme wave return period.
- Difference in included environmental loads.
- Difference in response calculations.
- Difference in validation according to design standards.

Results showed some variation in the definition of design conditions for each WEC. It was clear that most focus had been put into investigation of wave exposure while wind loads had only been considered for two of the devices and current load only for one. In addition, the considered safety levels of these contributions were varying. According to e.g. DNV-OS-E301 [14], 100-year extreme wind and wave loads should be considered in the ultimate limit state (ULS) and 10-year extreme current load. In ISO, [16], IEC [17] and API [15], this is defined as 100-year events from all environmental loads. Some of the devices had only been designed for 50-year wave events, some for 100-year events while these considerations, in general, had not been made for current and wind.

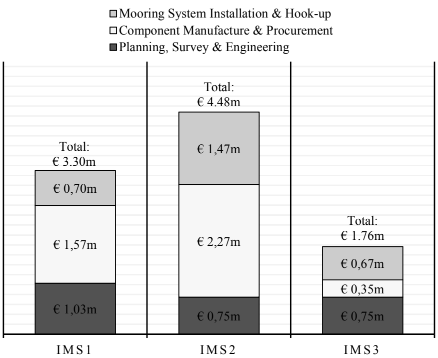

Fig. 3. Chart showing the results from the cost estimations of the initial mooring solutions. Adapted from [23].

For estimation of the loads, many different methods were used. In some cases, laboratory results were scaled up, multiplied with a factor in order to account for changes in geometry, while quasi-static methods had been used by some to evaluate the design tensions. This is a method that can be justified for O&G structures where the mass of structures is large and the corresponding motion response comparatively low. As mentioned, WECs can allow for larger motions and have much smaller masses. Therefore, it is expected that a dynamic analysis will provide different results. In the design standards, it is also required that full dynamic analysis is used for the final design.

Clearly, none of the systems could be considered final solutions at the time and needed deeper investigation and more thorough design. Because of the design methods at the time, the reliability and the expected lifetime of the system were different, and the cost of the systems difficult to compare. Still, [23] calculated the expected price of the systems throughout a lifetime of 20 years, in order to evaluate the potential for each solution.

The results are presented in Figure 3 and include all CAPEX and contains estimations of cost for planning, surveys and engineering, together with components manufacturing and the final installation and hook-up. It is obvious that the IMS3 results in the lowest overall cost of €1.76m compared to €3.30m and €4.48m for the IMS1 and IMS2 respectively.

Considering the cost of components manufacturing and procurements, it is clear that the IMS3 provides the lowest cost, due to the very simple system, with light and few materials. The IMS1 is more expensive as heavy chains are used, but as only three lines were considered at the time, the price was still not as high as for IMS2 which combined six lines of partly chain and synthetic ropes. This resulted in a need for much more material. The need for six anchors in this system also provided high cost. Considering the price for installation and hook-up, it was clear that the price could be considerably reduced by reducing the weight of the system.

It is difficult to compare the cost of these systems, because they are four different WECs and have been designed based on different methods. However, from the cost estimations it was observed that the weight of the systems highly affected the overall price, and it was seen that a SALM system potentially could be a cost-effective solution and, therefore, the following parameters were considered relevant to investigate:

- Application of light and compliant synthetic materials instead of steel chains in order to lower cost.
- Investigation of SALM type mooring systems.

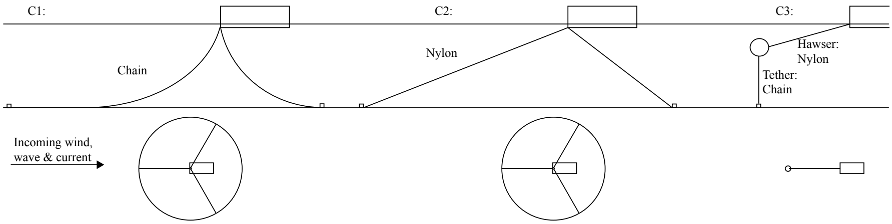

Fig. 4. Illustration of the three mooring solutions investigated in the preliminary analysis [23].

TABLE II
RESULTS FROM THE QUASI-STATIC ANALYSIS OF C1-C3 (CF. FIG. 4). ADAPTED FROM [23]

	C1	**C2**	**C3**
Linear horizontal stiffness	348 kN/m	51 kN/m	164 kN/m
Cable length	Ø80 R5 studless chain: 3 × 1160 m	Ø120 Nylon: 3 × 267 m	Ø92 R4 studless chain: 22.2 m Ø184 Nylon: 30 m
Seabed footprint radius	1150 m	268 m	-
Cable Mass	Chain (128 kg/m): 445 t **Total: 445 t**	Nylon (8.17 kg/m): 6.5 t **Total: 6.5 t**	Chain (169.3 kg/m): 3.8 t Nylon (16.2 kg/m): 0.5 t **Total: 4.3 t**
Cable cost	Chain: €1,844,122 **Total: €1,844,122**	Nylon: €105,358 **Total: €105,358**	Chain:€15,558 Nylon: €7,824 **Total: €23,382**

IV. Preliminary Analysis

A preliminary analysis was initiated and described in [23] in order to make a more direct comparison between a mooring system using synthetic ropes, a system using chain and a SALM system. This allowed for direct comparison of the cost reduction potential of the stated considerations as now,

- The same floating structure was considered,
- The same response calculation method was applied,
- The same environmental conditions were used.

The study used a simple quasi-static analysis, and only considered the cost of the mooring lines. The investigated structure and environmental conditions resembled those that could be expected for the partner WECs, and three mooring systems were defined as illustrated in Figure 4. This resembled a turret catenary system (C1), a taut synthetic system (C2) and a SALM system (C3) with synthetic hawser and a chain tether.

The results of the analysis are listed in Table II and support the conclusion drawn from the cost estimations. C3 provides the least costly solution. The most expensive solution is C1 due to the heavy chain. During the analysis, it was proven difficult to find a realistic solution as the heavy chain in the shallow water depth resulted in a very stiff system with corresponding high loads. This is seen in Table II where the linearized horizontal stiffness is listed. This leads to a need for strong and even heavier chains. In order to introduce some compliance to the system, very long lines were needed, giving a system that is unrealistic to ever build. Additionally, when having very long lines, only the elasticity from a limited length of the bottom lying chain can be included since a considerable part of the chain lying on the bottom cannot be engaged at extreme events where it is needed, because it is buried in the seabed mud. The C2 system provided a realistic and cheaper solution than C1.

The analysis clearly illustrated the potential of introducing compliant and light synthetic lines and to consider a SALM system. Based on this and the desires from the WEC developers, it was decided that the systems defined in Table III would be considered for each of the WECs. Designing a mooring system is an iterative process, and several parameters like e.g. line length, number of lines etc. can be varied. For each of the systems it was defined which parameters would be optimized in later design; these are listed in Table III.

V. Investigation of Design Procedures

During the initial assessments, it was found that different methods had been used to estimate the mooring response and design loads, mainly considering quasi-static results or direct implementation of laboratory experiments. The latter naturally provides the most reliable solution when considering only the wave exposure. For a final design, it is necessary to also include current and wind loads and, as such, necessary to perform experiments that are more sophisticated or consider numerical analysis. In order to evaluate the validity of a quasi-

TABLE III
DEFINITION OF THE MOORING SYSTEMS CONSIDERED FOR THE PARTNER WECS TOGETHER WITH THE PARAMETERS THAT ARE USED FOR OPTIMIZATION OF THE SYSTEMS.

WEC	Mooring System	Optimization Parameters
Floating Power Plant	Taut, synthetic, turret system.	Footprint radius, mooring line diameter, number of lines.
KNSwing	Taut, synthetic, turret system.	Footprint radius, line diameter, number of lines.
LEANCON	SALM system with submerged buoy and synthetic hawser.	Volume of the two buoys, mooring line diameter, hawser line length.
Wave Dragon	SPM system with surfacepiercing buoy and synthetic lines.	Footprint radius, mooring line diameter, number of lines, buoy volume, hawser diameter.

Fig. 5. Picture of the model used in the experimental work.

static analysis, which had been used so far, and validity of full dynamic tools, a two-month test campaign was conducted. A simplified model of the Floating Power Plant in scale 1:64.5 was used and tested in regular, operational and extreme sea states. A detailed description of the experimental work can be found in [24]. Firstly, a comparison with the quasi-static approach was performed. In this method all dynamic effects were neglected, non-linearities were linearized and only horizontal degrees of freedom (DoF) were considered. The results (cf. Figure 6) showed that relatively good agreement was obtained with an absolute error of up to 15% for mild sea states. However, for the extreme events, which are used for evaluation of design loads in ULS, the results were found to be highly underestimated with an error of up to 50%, cf. Fig. 6. For a design case, this is highly undesirable and a more sophisticated model must be used. A full dynamic simulation must be considered where all load contributions, DoFs and the coupled effects between lines and WEC are included.

In order to find the most suitable dynamic analysis tool, a screening of available software packages was performed, considering that it should ensure that the final analysis could be certified by a certification company. The most significant requirements to the software packages were as follows:

- Ability to perform coupled analysis.
- Analysis in the time domain.
- Capability to model non-linear mooring line stiffness.
- Complete description of wind and current loads.
- First and second order wave effects.

Seven commercial tools and one non-commercial tool were investigated in [25]. From that screening, the commercial tools *OrcaFlex* [26] by Orcina Ltd. and *DeepC* [27] by DNV-GL were chosen for further investigation. A case study was used to compare the results from the two software packages, and it was decided that *OrcaFlex* was to be used in dynamic analysis performed in the project.

In order to validate the numerical tool, the results from the experimental campaign were used, as presented in [28]. Often, the experimental work is used to tune the numerical model, which ensures high agreement between the model and physics. This, however, was not done in this study as it was desired to investigate the accuracy of the numerical model and to evaluate whether or not this tool could be used in an initial investigation of the moorings for each of the devices where not all had experimental data available. The model used radiation/diffraction data from the open source BEM code *NEMOH* [29] and included drag elements where a simplified method had been used for determination of drag coefficients. The relative errors between the tension found in the numerical and experimental models are presented in Figure 6. Compared to the results obtained from the quasi-static model, significant improvement was seen, and instead of underestimating the tension, as was the case for the quasi-static analysis, this model made some overestimation. From a safety point of view, this is considered desirable and, moreover, the maximum error was reduced to 11%. Based on this fact, it was assumed that the models could be used for all the WECs in the project, with errors in the same range as seen in the validation.

VI. Dynamic Analysis and Mooring Optimization

As stated in Table III, different parameters could be varied for each WEC in order to optimize the layout. For each change, the behaviour of the system changes and consequently the corresponding line tension and WEC motion. Each variation of the mooring layout, therefore, requires a full dynamic time domain simulation in order to evaluate if the system fulfils the requirements in design standards [14]–[17] and the requirements to motions defined by the developer. This means that at least a three-hour simulation must be simulated for each mooring configuration and despite relatively effective

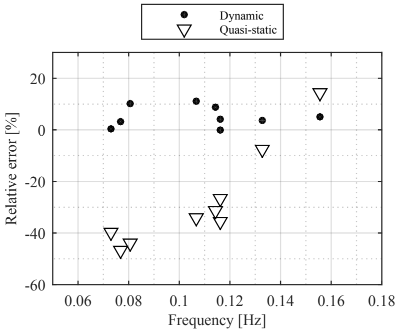

Fig. 6. Comparison of error between line tensions from experiments and quasi-static and numerical model.

computational time for an *OrcaFlex* simulation, it becomes unrealistic to cover the complete test domain.

In order to find the optimal mooring solution for each WEC, a meta-model based optimization procedure was adapted and applied to the current systems. This means that only a limited number of evaluations are needed. A meta-model is then constructed based on the initial evaluations and used to find the optimal configurations, then evaluate the response for these and use the results to improve the model. This is done until a defined termination criterion is met, after which the most optimal solution is identified. In the present study, the cost of the system was used as the objective of the optimization while only solutions fulfilling response requirements were considered. The MATSuMoTo MATLAB toolbox [30] was used to implement the optimization procedure. Running the optimization procedure is currently being done.

Prior to running the optimization procedure, a cost database was constructed based on the same parameters as used in the initial cost evaluation. This is intended to allow for the possibility of comparing cost prior to and after the project. The effect on the LCOE will be evaluated in later work, in which also the cost database will be presented together with an evaluation of the optimization and the parameters that drive the cost of the systems.

VII. Discussion and Conclusion

The present paper has described some of the outcomes of the project *"Mooring Solutions for Large Wave Energy Converters"* and the work that was carried out during the years that the project has lasted. The baseline of the project was defined by an initial assessment of the current state of the mooring design of four Danish WECs, finding that each had applied different solutions, which gave variations in the cost of the systems. In addition, the design procedures were different, and all procedures lacked more work before the mooring design could be considered final.

Different solutions were investigated, finding that cost reduction could potentially be found from applying synthetic ropes instead of chains and by considering a SALM system. Applying these systems were expected not only to reduce cost, but also to improve reliability of the system since much more compliance was introduced, which reduces the loads and helps decrease the risk of failure.

Different methods for design of mooring were investigated, which illustrated that the quasi-static method, which had been used previously, had a potential to underestimate the line tension. A full-dynamic tool was found and compared to experimental data. The work showed that good agreement could be accomplished, even without tuning the model to experimental data. This method is then expected to be used in full dynamic analysis of the mooring solution for each of the investigated WECs, with the aim of reducing the cost. An optimization process has been defined, and the work is presently undergoing work and will be presented in later papers.

The present study has mainly focused on the ULS as this state often has the most dominant effect on the mooring system cost. In order to ensure that the systems can be certified by a certification company, future work should also include fatigue limit state (with more sophisticated modelling of the PTO) and put more focus on accidental limit state. With the results from the upcoming optimization process, it will be possible to define what parameters that mostly affect the cost of each solution and thereby find the layout most suitable for each device and deployment site. Before a final design, it is similarly necessary to reduce any uncertainties by conducting additional experimental work that verifies the results and allows for further optimization of the numerical model. This, however, is out of scope of this project and should be considered in future projects.

Acknowledgment

The present paper has been funded by the Energy Technology Development and Demonstration Programme (EUDP) through the project *"Mooring Solutions for Large Wave Energy Converters"*. The authors wish to acknowledge and thank all who have provided input, discussions and results to the project throughout the last three years.

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Paper G

Cost Optimization of Mooring Solutions for Large Floating Wave Energy Converters

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The paper is under review for publication in
Energies, ISSN 1996-1073, 2017.

Article

Cost Optimization of Mooring Solutions for Large Floating Wave Energy Converters

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Academic Editor: name
Version December 8, 2017 submitted to Energies

Abstract: The increasing desire for using renewable energy sources throughout the world has resulted in a considerable amount of research into and development of concepts for wave energy converters. By now, many different concepts exist, but still the wave energy sector is not at a stage that is considered commercial yet, primarily due to the relative high cost of energy. A considerable amount of the wave energy converters are floating structures, which consequently need mooring systems in order to ensure station keeping. Despite being a well-known concept, mooring in wave energy application has proven to be expensive and with a high rate of failures. Therefore, there is a need for further improvement, investigation into new concepts and sophistication of design procedures. This study uses four Danish wave energy converters, all considered as large floating structures, to investigate a methodology in order to find a cheap and reliable mooring solution for each device. The study uses a surrogate-based optimization routine in order to find a feasible solution in only a limited number of evaluations, and a constructed cost database for determination of mooring cost. Based on the outcome, the mooring parameters influencing the cost are identified and the optimum solution determined.

Keywords: Mooring; Station Keeping; Wave Energy; Optimization; Meta-Model; Surrogate Model; Cost; Wave Energy Converters (WEC)

1. Introduction

The rising demand for sustainable and renewable energy in the world has led to an increasing research into and development of alternative energy resources. By now, energy from e.g. wind and solar is well developed and an active part of the energy mix in industrialised countries worldwide. Despite a comparatively large energy potential, one resource that still is not a part of the energy mix is wave energy. During the last decades, the amount of research in wave energy absorption has been significant, resulting in a considerable amount of concepts for new wave energy converter (WECs), with varying levels of development. Despite the effort, the wave energy sector is not yet at a commercial stage, and further improvement must take place before wave energy can contribute to the energy mix.

[1–3] lists the levelized cost of energy (LCOE) for a range of energy resources, indicating the high price of wave energy compared to oil and gas (O&G) and even other renewable resources. As a result, there is an urgent demand to decrease the cost in order for wave energy to evolve from the current pre-commercial stage. According to [4,5], several parameters can be improved and take a considerable part in the cost reduction of wave energy. Despite different evaluations of the importance, station keeping moorings are listed as a driver towards lower cost, as it is estimated by [6,7] to compose

20-30% of the total structural cost of a WEC. In [8], the mooring is estimated to take up 8% of the CAPEX cost.

In addition to the cost, by now mooring has also taken part in the failure of several WECs due to insufficient durability of the mooring system [7,9,10]. Consequently, a Danish research project entitled "Mooring Solutions for Large Wave Energy Converters" (MSLWEC) was initiated in 2014, which aimed at reducing cost of the system, improving the applied design procedure and increasing the durability of the systems. Figure 1 illustrates a diagram of the project work and presents how the previous tasks have provided the basis and experience for this paper.

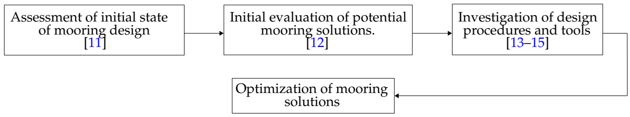

Figure 1. Flow diagram of the previous and related work in the MSLWEC project.

The project took its point of departure in the Danish Wave energy sector and four WECs, all considered to be large floating structures with passive moorings, meaning that the mooring does not take active part in the energy conversion. The four WECs are the Floating Power Plant [16], KNSwing [17,18], LEANCON Wave Energy [19] and Wave Dragon [20], cf. Figure 2. In the early research, the initial layout and design procedures were investigated, cf. [11], and they highlighted a significant need for a more thorough and detailed design and investigation of the applied mooring systems. This conclusion was based on the fact that all environmental conditions were not fully included and, in most cases, a quasi-static approach was used. Finally, the project also concluded that there was a common tendency of applying traditions from the O&G sector in e.g. using heavy mooring chains for the system. The following task [12] evaluated the initially applied systems, identified the use of mooring chains as an inefficient solution and instead highlighted compliant, synthetic ropes as potentially cheap and useful solution and, furthermore, identified a single anchor leg mooring (SALM) system as a strong potential for WEC mooring. Other studied like e.g. [21,22] similarly identified synthetic ropes as an appealing solution.

The following study [13] produced a large database of experimental data, used it to validate the initially considered quasi-static design approach, and it identified a clear underestimation of mooring loads in extreme seas. Consequently, there was a need for a full dynamic analysis as well as for further optimization of the initial mooring solutions since there was a clear possibility that these would not be capable of surviving in design storm conditions with extreme wind, wave and current.

Many different software packages are available for mooring analysis and have the capability to analyse the motion response and tensions under environmental load exposure according to limit states defined in design standards as [23–26]. A number of tools were investigated in the project, cf. [14], and have also been listed in other publications like [27]. A selected software package was validated against the experimental data in order to gain knowledge on the applicability of the tools on initial design, and the software proved its ability to model line tensions with acceptable overestimation of the tension without major tuning of the numerical model.

As presented in [13,27,28], the mooring characteristics are highly dependent on the site specification, mooring layout, materials etc. and the response is, therefore, highly affected by the choices. The mooring design procedure is iterative and it can be extremely time consuming to cover the full design space in order to find a solution that fulfils all defined requirements and, at the same time, provides a low cost. Based on the gained experience from the previous work on selecting tools, building hydrodynamic models and designing mooring, an optimization procedure must be utilized in order to find a cost-optimized solution, which introduces more reliable and fully designed solutions compared to the initial layout.

In several studies like e.g. [29–31], the energy absorption has been the objective of optimization with additional investigation of mooring line loads. The studies vary in the level of details, number of investigations of mooring configurations and applied methodology, but they generally focus on operational conditions and the aim of improving the energy absorption as much as possible. This type of mooring is consequently reactive or active, and the actual cost of the moorings is not the objective of the studies. Naturally, optimizing mooring loads might reduce the needed strength of e.g. lines and thereby reduce cost, but no actual cost investigation was done. Other optimization studies like e.g. [32] treat WEC farm layout in order to achieve the most feasible layouts for energy harvesting.

A passive mooring does not take active part in the power take-off (PTO), and the cost is mostly determined by the extreme sea states during which survivability must be ensured. The present study focusses on the large floating WECs with passive moorings and uses an optimization procedure to reduce the mooring cost while securing that a reliable solution is found. The study continues the already presented work in the project "Mooring Solutions for Large Wave Energy Converters" and uses the four Danish WECs in Figure 2 as case studies.

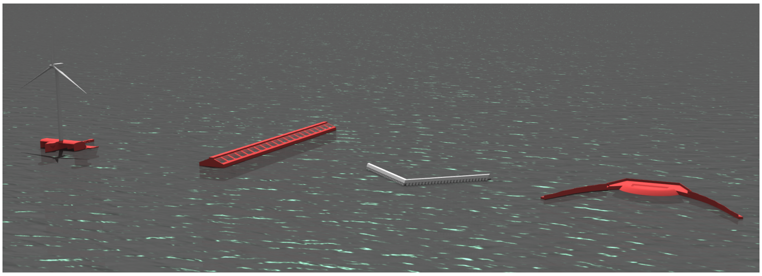

Figure 2. The four WECs considered in the mooring optimization assessment. From left to right: Floating Power Plant, KNSwing, LEANCON Wave Energy and Wave Dragon.

The paper is structured with four sections including this introduction. Section 2 describes the applied method and the design variables for the four cases, while Section 3 presents the results from each case. In Section 4, the work is summarized and discussed.

2. Method

This section presents the methodology used for the optimization study and describes the four cases by means of environmental conditions, design limits and choice of optimization parameters.

2.1. Mooring Cases

The presented WECs are planned for deployment at different locations and have differences in their design requirement. Because of this, four cases are defined, one for each WEC and its mooring system which is designed and optimized for the relevant deployment site. In previous publications like [12], the potential of different mooring solutions were assessed, and based on this, three different mooring solutions are considered for the four WECs. These are illustrated in Fig. 3 and covers (a) a single anchor leg mooring (SALM) system with submerged buoys, a deformable tether and a nylon hawser; (b) a taut turret system with nylon lines; and (c) a single point mooring (SPM) system with nylon lines and hawser.

All lines are composed of a chain part at the seabed connection, a synthetic line and a chain part at the fairlead. This is necessary for re-tensioning, to account for creep and to account for installation tolerances. A 2 m long piece of chain is located at the seabed, while a chain length corresponding to 6% of the total line length is located at the WEC connection. Table 1 lists the system applied to each WEC.

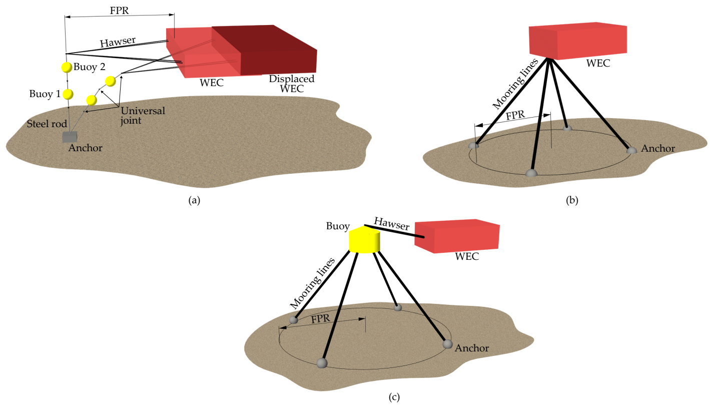

Figure 3. General illustration of the mooring layouts considered in this study. (a) Is a SALM system with illustration of the displaced layout. (b) Is a taut turret system. (c) Is a synthetic and taut SPM system. The figure defines the footprint radius FPR, for each layout.

Table 1. Definition of design cases for each of the WECs in Fig. 2. The table list all considered environmental conditions together with the defined restrains on surge and pitch. ✗ denotes no limit.

	Case 1	**Case 2**	**Case 3**	**Case 4**
WEC	Floating Power Plant	KNSwing	LEANCON	Wave Dragon
Mooring solutions	Taut turret Fig. 3b	Taut turret Fig. 3b	SALM Fig. 3a	Taut SPM Fig. 3c
Water depth, h	30 m	40 m	25 m	25 m
Significant wave heigh, H_s	6.6 m	9.9 m	8.3 m	8.3 m
Peak wave period, T_p	9.3 s	11.4 s	10.5 s	10.5 s
Relative depth, h/L_p	0.14	0.12	0.1	0.1
Current velocity, v_c	1.3 m/s	1.0 m/s	1.5 m/s	1.5 m/s
Wind velocity, v_w	33.0 m/s	39.9 m/s	34.0 m/s	34.0 m/s
Surge design limit	$\pm$ 29 m	$\pm$ 44 m	$\pm$ 30 m	$\pm$ 27 m
Pitch design limit	$\pm$ 15°	✗	✗	✗

2.1.1. Environmental Conditions

Each WEC is planned for deployment at a specific site either at the DanWEC test facility in Denmark, the Danish part of the North Sea, or at the Belgian Coast. Prior to the optimization, the environmental conditions for each site were assessed and the 100-year extreme conditions were specified and can be read from Table 1. It is assumed that all waves are long-crested (2D), irregular and distributed in a JONSWAP spectrum with a peak enhancement factor, $\gamma = 3.3$.

The current is assumed steady over time, while the wind is modelled as a steady component and a time-varying gust component. The latter is described by an NPD wind spectrum according to [33]. Both wind and current are assumed to be varying in the vertical direction and are modelled with a power law profile.

The wind speed defined in Table 1 corresponds to the 1-hour mean value in a height of 10 m, while the current velocity is at the still water level (SWL). All environmental loads are assumed to be aligned; hence, a full 2D problem is analysed.

2.1.2. Design Criteria

Different design standards are available and can be used in the design of mooring systems, e.g. [23–26]. These standards define the necessary requirements and ensure survivability in operational and extreme conditions by considering design in ultimate, fatigue and accidental limit state (ULS, FLS and ALS). For a passive mooring system, the mooring should not affect the PTO and [34] lists that ULS has the most dominant influence on the cost. The present study, therefore, only considers the ULS for which reason it is the 100-year extremes that are listed in Table 1. In several publications like e.g. [35,36], the safety levels defined by the design standards have been discussed. A relaxation of safety levels for WECs compared to O&G structures have been suggested, due to the lower consequence if a failure occurs. In [35], the safety levels for API-RP-2SK and DNV-OS-E301 are compared, and it is found that the API standard provides higher safety than the DNV. ISO and IEC provide similar safety factors as API. In other works like [37–40], the topic of reliability assessment of WECs is treated with detailed work on calibration of safety factors, estimation of extreme values and reliability assessment approaches.

In order to follow the suggestion from [35] of using relaxed safety, the DNV-OS-E301 standard and the most relaxed consequence class (CC1) are used in the present analysis.

The standard focuses primarily on ensuring sufficient strength of the mooring lines and anchors to withstand the induced tensions. According to DNV-OS-E301, the design tension is defined by equation (1):

$$T_{C,mean}\gamma_{mean} + T_{C,dyn}\gamma_{dyn} \leq S_C \tag{1}$$

Where S_C is the characteristic strength corresponding to 95% of the minimum breaking strength T_{MBS} and $\gamma_{mean} = 1.10$ and $\gamma_{dyn} = 1.5$ are the safety factors for respectively the mean and dynamic part of the line tension. $T_{C,mean}$ is the mean tension, while $T_{C,dyn}$ is the dynamic part of the tension and is defined by equation (2):

$$T_{C,dyn} = T_{MPM} - T_{C,mean} \tag{2}$$

Where T_{MPM} is the most probable maximum with a 63% probability of exceedance when the extreme peaks tend to follow a Gumbel distribution. By applying the safety factors for the given consequence class, a target annual probability of failure of 10^{-4} is obtained.

According to DNV [41], the anchor should be designed for the same design tension as the lines, while the characteristic anchor resistance provided by the manufacturer is reduced by a safety factor $\gamma_m = 1.3$.

DNV-OS-E301 is only considering survivability in ULS and, hence, the excursion is not specified in the standard as a design criterion. However, the WECs are equipped with umbilicals, which puts a limit on the allowable excursion, because tensions in these must be prohibited [27]. The design of the umbilical is, therefore, often a part of the mooring design and a compromise between cost of umbilical and mooring. In the present study, the excursion limit in Case 3 was defined by the developer, while it was approximated for the remaining cases by assuming a lazy-S layout for the umbilical, cf. Figure 4. By accounting for the minimum bending radius of a suitable cable, clearance between seabed and sea surface and for water level variations (high water level (HWL) and low water level (LWL)), it is possible to calculate a cable length and an allowable excursion. This limit is defined in Table 1 and illustrated in Figure 4. Naturally, this is only a pragmatic approach to obtain an estimation of the excursion limit, but in the final design, a more detailed investigation and design of the interaction between umbilical of WEC must be included in the mooring design.

The WEC in Case 1 is equipped with a wind turbine, which puts additional restraints on the motions. In operational conditions, the power production of the turbine must be considered, while it is merely vital to secure stability in the ULS. For the given turbine, a pitch limit of $\pm 15°$ was defined by the developer, cf. Table 1.

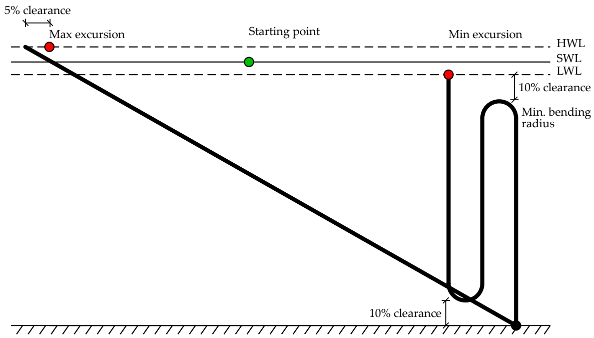

Figure 4. The considered umbilical configuration and definition of the allowable excursion.

2.1.3. Optimization Parameters

Considering the layout of the mooring solutions in Fig. 3, several parameters can be varied for each solution and will change the characteristics of the system. Table 2 lists the optimization parameters for each case together with the maximum and minimum values considered for each parameter. As seen, the mooring and type of optimization parameters for Cases 1 and 2 are identical, while it varies for Cases 3 and 4.

For all cases, the Bridon Superline Nylon [42] is considered with the range of diameters listed in the table. The lines are modelled with a non-linear stiffness curve and structural parameters according to [42].

The definition of the footprint radius, FPR, is illustrated in Fig. 3, and varies significantly for each case. In Cases 1 and 4, the maximum FPR is chosen in order to ensure that there is no interaction between the WEC and the lines, cf. Figure 5. For Case 2, this problem is not present, and the upper value is chosen based on the length of the device. For Case 3, it is based on the umbilical.

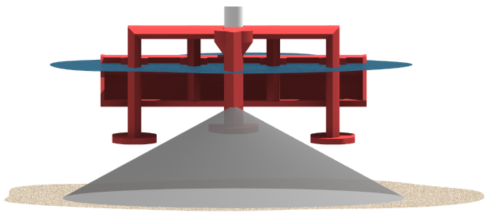

Figure 5. Illustration of the WEC in Case 1 and the necessary limit on FPR in order to avoid interaction between line and structure.

The unstretched mooring line length is not considered as a direct optimization parameter, even though it is varied for each configuration. For the WECs in Cases 1, 2 and 4, a vertical pretension is specified and fixed for all simulation and, therefore, the line length is varied dependent on the FPR and line stiffness in order to achieve this pretension. In Case 4, the vertical pretension is calculated to ensure vertical equilibrium between line tension and buoyancy of the surface buoy in calm sea. In Case 3, the line length is varied directly because of the FPR. In this case, the steel rods are not a part of the optimization, but are dimensioned based on the obtained design tension in the rods. The maximum dimensions of the buoys are decided based on the length of the tether. In Case 4, the value is based on investigation of the available buoys from commercial manufacturers.

In Case 4, the hawser length is not varied since the zero-position of the buoy is fixed, and there is no pretension in the line. In early stages of the work, it was intended to use a nylon hawser, but a suitable solution was not found, and a rigid bar is used instead.

The anchor type is determined prior to the optimization as drag embedded anchors with vertical strength for Cases 1, 2 and 4, while a gravity-based anchor is used for Case 3. The anchors are designed based on the achieved tensions in the lines at the anchor point and hence, the necessary anchor size is determined after each simulation is completed and is not an optimization parameter.

Table 2. Definition of the considered optimization parameters for each case and the applied value ranges. ✗ denotes that the parameter is not considered.

Parameter	Case 1		Case 2		Case 3		Case 4	
	Min.	**Max.**	**Min.**	**Max.**	**Min.**	**Max.**	**Min.**	**Max.**
Mooring line diameter (mm)	40	192	40	192	✗	✗	40	192
No. of mooring lines (-)	4	10	4	10	✗	✗	4	10
Hawser line diameter (mm)	✗	✗	✗	✗	40	192	✗	✗
Footprint radius (m)	30	40	80	250	40	75	25	100
Buoy 1 diameter (m)	✗	✗	✗	✗	1.5	6.0	3.5	15.0
Buoy 2 diameter (m)	✗	✗	✗	✗	1.5	6.0	✗	✗
No. of optimization parameters	**3**		**3**		**4**		**4**	

As seen from Table 2, the optimization considers between 4-10 lines in the layouts. In some applications, cf. e.g. [11], configurations with three lines are considered. In this study, however, this number is considered insufficient in order to increase durability and redundancy, which is considered a vital parameter for a mooring system [27]. This is mainly a problem in the ALS, but must be taken into account early in the process. Figure 6 presents the influence from the number of mooring lines on the static behaviour of a mooring. Considering the first graph, it is indicated that the horizontal mooring stiffness, as expected, increases with the number of lines. By having only three lines, it is possible to achieve around 40% of the stiffness with 10 lines. Considering the second graph, the advantage of having more than three lines is clear. The figure presents the relationship between intact stiffness and the stiffness when one line is broken. With only three lines, almost all stiffness is lost and even though two lines are remaining, the durability of the system can be expected to be much less. With 4-10 lines, 60-80% of the stiffness remains after one line fails. By also considering the third graph, it is seen that the static position of the device at failure of a line is significantly worse when only having three lines. The displacement is approximately 55% larger than with 10 lines with one broken, while it decreases to 12% with 4 lines. The displacement is illustrated in Figure 6 with examples for 3 and 6 lines, where the displacement in the latter is almost negligible and hardly visible.

A traditional SALM system does not provide the same amount of redundancy as only one line connects the buoy to the seabed. For the given system, the hawser is composed of four lines, meaning that there is some redundancy if one nylon line breaks. However, in general, the system is much more vulnerable to failure. As a result, DNV-OS-E301 requires an additional safety factor of 1.2 in tension.

In a final design, it is necessary to ensure survivability in both ALS and ULS. As previously stated, this paper focusses on the ULS, but the above considerations and the fact that a solution with three lines has been discarded, account for a simple and coarse evaluation of the ALS.

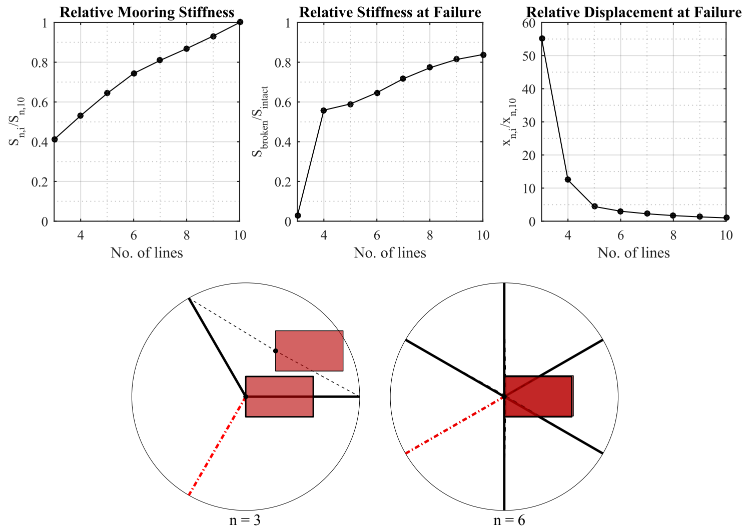

Figure 6. Influence of mooring line number on intact horizontal stiffness, stiffness after failure of one line and the static position of WEC after failure. The figure also illustrates two examples of the WEC position after a line failure. The red line indicates the broken line.

2.2. Mooring Analysis Procedure

There are different methods available for evaluation of the wave-WEC interaction, generally divided into experimental and numerical methods. Experiments provide highly reliable solutions, but are time-consuming, often expensive and it is difficult to include all environmental loads without introducing significant sources of errors, especially in small-scale tests. Sophisticated non-linear numerical models can be established through e.g. computational fluid dynamics (CFD) or smoothed particle hydrodynamics (SPH), but the computational demands for such calculations are immense, which is why the methods are not suitable for initial mooring design where many iterations are needed and the simulation time is long. Often, the boundary element method (BEM) is used, which assumes inviscid, irrotational fluids and uses linear potential flow theory with its assumption of small steepness waves and small amplitude structure motion. In the ULS, this theory is stressed significantly, due to the extreme wave condition and compliant mooring which allows large motions, but this work builds up on the experience gained in previous work [15] where the applicability was validated for initial design and analysis.

In this study, the open-source BEM code NEMOH [43] is used, which provides the frequency dependent added mass, radiation damping, wave excitation and Kochin coefficients. As explained in [44], the latter can be used for calculation of the mean wave drift coefficients using the far-field formulation.

The output of NEMOH is coupled to the commercial time domain mooring solver OrcaFlex [45], which utilizes Cummins equation to determine the coupled response of the WEC and mooring lines, by including all first and second order wave load contributions together with the wind and current loads. The output of the software package is a time series of line tension and WEC motions. The construction of the numerical model follows the procedure presented in [15].

Using linear theory on Case 4 is not without problems, as the device is an overtopping WEC where a safety system increases the draught in extreme seas to increase overtopping and decrease the loads. The BEM cannot directly account for overtopping, and by increasing the structure volume below the SWL the loads are actually increased. In later research, this problem will be addressed and an improved model will be used. At present, the model is used with this inaccuracy, which especially is causing a significant problem with second order drift. Consequently, this first optimization of Case 4 has only been for 1^{st} order wave loading, while both 1^{st} and 2^{nd} order effects are considered for the remaining cases.

2.3. Cost Database

For calculation of cost of each mooring system configuration, a database was constructed where the cost was calculated based on specification of the mooring components. The database included a high level of details and considered cost of both CAPEX and OPEX for a lifetime of 20 years. The database was based on the authors' experience and knowledge from other marine projects, manufacturers and providers.

The following subsection provides some of the assumptions behind the numbers, but it is not possible to list costs directly in this paper.

2.3.1. CAPEX

Planning, design, survey and engineering conforms a significant part of the mooring cost and was included as a fixed price based on the type of mooring system. The cost included assumption on the entire design process from determination of environmental conditions, site and seabed surveys (including vessel and labour), analysis and detailed engineering, together with estimation of cost for review and certification and planning of the installation. All cost was found based on assumption of the duration of each task and the expected day-rates. Final inspection and project management was also included.

The *manufacturing and procurement cost* is primarily based on prices provided by manufacturers of the given components. For the nylon line, the cost is dependent on the MBS and includes the protective jackets, a chain part at the connection point between line and WEC and the cost of connections. The anchor cost is based on the type of anchor and necessary weight to ensure survivability and includes the chain at the anchor point.

The cost of buoys and stainless steel rods is based on a fixed cost per unit weight and with estimation of cost for universal joint, top-swivels and connection hardware. The cost of fairleads with tensioners was included and dependent on the number of lines.

Estimating the cost of a turret system is highly complicated and takes significant part in the overall cost. For the given cases where turrets are included, the cost is a fixed price based on experience from comparable applications.

All component cost includes certification and delivery to a North European port.

The *installation and hook-up* cost is mostly dependent on cost of the needed vessels and the amount of labour needed. For each type of mooring system, the types of vessels, cranes etc. were decided, and the current day-rate for these were collected. Generally, it was assumed that one line could be installed per day and a total of two days were needed for tensioning of all lines. For each vessel day, four person-days were needed. The installation cost included cost for preparation in the port and estimation of waiting time for a suitable weather window.

The *decommissioning* cost was based on a dismantling cost corresponding to 110 % of the installation and hook-up, while material disposal was estimated to 1 % of the CAPEX. A fixed value for seabed clean-up was found and dependent on the type of mooring system, as some systems have more seabed interaction than others.

The CAPEX is based on existing experience and current cost of vessels and labour. Naturally, this introduces some uncertainties in the calculations, particularly because presently, the cost rates are at the lowest level in a number of years due to the decline in the oil industry. Similarly, installation cost etc. has been made with assumptions on waiting time, which is highly dependent on the site and season. For final cost evaluations, more specific input can be made to the cost database to decrease uncertainties, but presently it provides generic and realistic estimations on the cost.

2.3.2. OPEX

The OPEX cost is based on the lifetime of 20 years and includes insurance, inspection and maintenance. The insurance is estimated to 1%/year of the total CAPEX, while 4 vessel days and 16 person-days are assumed for inspection and maintenance. This corresponds to checking and adjustment of tension after one month and after 1, 5 and 10 years. Replacements throughout the lifetime are assumed to correspond to 1.5% of the CAPEX based on experience.

Similar to the CAPEX, the OPEX cost introduces some uncertainties mainly resulting from the fact that some level of novelty is seen in the concepts. This means that no historical evidence is available to support the reliability for a 20-year deployment. Additionally, the risk of external damage to the system is difficult to take into account in the cost evaluation. Finally, the insurance cost takes up a considerable part of the OPEX but in most industries, there is an increasing trend in this cost. Any type of failures due to mooring will expectedly lead to a significant increase in the insurance cost and at present, several years of deployment is needed to build a reliability record in the industry and more knowledge on OPEX cost.

2.4. Optimization Procedure

According to the problem defined in Section 1, the objective of the current work is to minimize the cost of the applied mooring system while securing that the found system satisfies the ULS. This results in the optimization problem in equation (3):

$$\min_{\boldsymbol{x}\in\mathcal{D}} f\left(\boldsymbol{x}\right), \tag{3}$$

where $\boldsymbol{x}$ is the variable vector, $f\left(\boldsymbol{x}\right)$ is the objective function and $\mathcal{D}$ is the design space, cf. Table 2. Since the cost and response are directly related to the combination of variables in the design space, a full simulation is required for each mooring configuration. The objective function, therefore, evaluates the cost of the mooring based on a dynamic simulation of the complete system under the extreme conditions. As the complete design space has a significant size, it is not feasible to evaluate the complete space to find the minimum solution and instead a methodology should be applied aiding in the search for minimums with only a limited number of function evaluations. For the objective function used in this study, no derivative information is available for identifying the minimum as no analytical description of the response surface can be constructed. Therefore, a derivative-free surrogate (also denoted meta-model) based optimization algorithm is used, which has the clear advantage of only requiring a relatively limited number of function evaluations compared to e.g. Evolutionary Algorithms [32]. In the surrogate-based optimization, the response surface is described by a surrogate model, constructed from the results of a limited number of function evaluation and used to identify the minimums. Several variations of derivative-free algorithms have been developed and presented in e.g. [46,47], and this paper utilizes the surrogate-based algorithm in the MATLAB Surrogate Model Toolbox (MATSuMoTo) [48], and the model will be further explained in later sections. Figure 7 illustrates the steps in the algorithm as defined in [29,48].

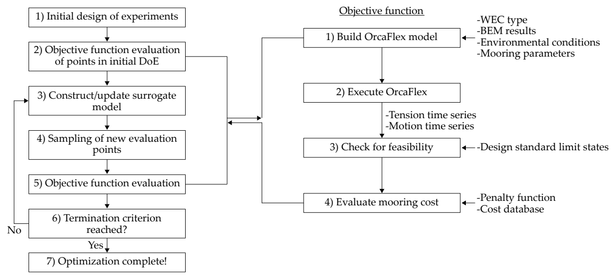

Figure 7. Flow-chart of the applied optimization procedure.

As illustrated in Figure 7, the procedure is initiated by evaluating the objective function in a number of tests (Steps 1-2) used to construct a surrogate model (Step 3). A new number of points are selected (Step 4), and the objective function is evaluated in these points (Step 5) and used to update the model (back in Step 2). This procedure is continued until a chosen termination criterion is met. The following sections will provide explanation of each step in the procedure.

2.4.1. Initial Design of Experiments

In order to make the surrogate model, it is essential to have initial knowledge of the design surface by conducting a number of simulations. In order to ensure that sufficient data is obtained and an efficient surrogate model can be constructed, the sampling points must be chosen to provide as much information on the response surface as possible. A design of experiments (DoE) strategy is initiated, which covers several different concepts [49,50]:

- Latin hypercube
- Corner
- Latin square
- Full Factorial
- Fractional Factorial

It is not possible to determine prior to the evaluation which strategy is the best choice, but according to [49] the Latin Hypercube is often suitable. The present study utilizes this method. A sufficient number of evaluations in the DoE must be chosen, and in the given study the numbers were based on the following equation (4):

$$n_{\mathrm{DoE}} = 2(d+1) \tag{4}$$

where n_{DoE} is the number of samples in the DoE and d is the number of optimization parameters [50].

2.4.2. Surrogate Model

In general terms, the surrogate model can be expressed by equation (5):

$$f(\boldsymbol{x}) = s(\boldsymbol{x}) + \epsilon(\boldsymbol{x}) \tag{5}$$

Where $f(\boldsymbol{x})$ is the output of the objective function in point $\boldsymbol{x}$, $s(\boldsymbol{x})$ is the output from the surrogate model and $\epsilon(\boldsymbol{x})$ is the difference between them [48]. Many different types of surrogate models are available and presented in [46,51] and are generally being either interpolating (Radial basis functions (RBF) and Kriging) or non-interpolating (Polynomial regression models and multivariate adaptive regression splines (MARS)) [51].

Different surrogate models suit different problems, and it is not possible to determine beforehand which model to choose without testing each of them. Considering that the purpose of the optimization is to find an optimum in short time, testing each surrogate model to find the best is not feasible. In order to limit the influence of a bad surrogate model choice, model ensembles (or mixture models) can be used, which utilize weighted combinations of two or more models and emphasize the models that perform well (low error and high correlation coefficients) and restrict the influence from the poor models (large error and low correlation coefficients). The mixture models are represented as equation (6) and (7) [51]:

$$s_{mix}(\boldsymbol{x}) = \sum_{r=1}^{N} w_r s_r(\boldsymbol{x}), \tag{6}$$

$$\sum_{r=1}^{N} w_r = 1, \tag{7}$$

where $s_{mix}(\boldsymbol{x})$ is the output of the mixture model at point $\boldsymbol{x}$, N is the number of surrogate models in the mix, s_r is the output of the r^{th} model and w_r is the weight of it. MATSuMoTo uses the Dempster-Shafer theory to combine the models and takes advantage of correlation coefficients, maximum absolute error, median absolute deviation and root mean square error to calculate the weight w_r for each model, based on the performed objective function evaluations in the procedure [52].

Naturally, the number of surrogate models in the mix results in a larger number of models that need to be updated in each loop in the procedure and put higher demands to the computational effort. On the other hand, the chance of selecting a poor model is minimized, and a much better description of the response surface is obtained [51].

Several studies like [46,51] have compared and investigated which models and model ensembles that perform best, and [51] found that the use of RBF, either alone or in combination with other models, generally provided a reliable solution. Consequently, this study uses a model ensemble of a cubic RBF and a MARS model.

2.4.3. Sampling Technique

Different methods can be considered for choosing the sampling points in each loop. In general, either a randomized sampling technique or the constructed response surface can be used. In this study, the randomized method is used.

Within the randomized method, two different strategies can be used. A local search can be considered where the current optimum points are perturbed. This method is most suitable for problems where only a single minimum is present in the response surface, as the solver will tend to search towards one minimum and will not necessarily find more. The other strategy is a global search where the solver still perturbs the best points so far, but also selects a number of uniformly distributed points in the whole design space. As the optimization progresses, the perturbation is decreased in order to find the best solution. When MATSuMoTo is no longer improving the output over a consecutive number of trials, the algorithm will restart with a new DoE and construct a new model based on the new evaluations in order to aim the search in other areas of the design space [50].

When selecting the best candidate points, two criteria are used: The distance and response surface criteria. The distance criterion is based on the distance to already evaluated points, while the response surface criterion is based on the value predicted by the surrogate model. A score is assigned each point as a weighted sum of these two. In order to select point close to the expected minimum, large weight is put on the response surface criterion, while large weight must be put on the distance criterion to select points in unexplored areas.

The optimization can either be considered as an integer or mixed-integer problem dependent on the variables. The different problems affect the sampling strategy as presented in [48,53,54] and for the mooring cost optimization, the definition of parameters are listed in Table 3. Most parameters are considered integer, such as line number, line diameters and footprint radius. Considering the installation process and allowable tolerances during installation, it is not relevant to consider the footprint radius as a continuous parameter. The buoys can be produced in custom sizes and are, therefore, continuous parameters in the optimization procedure.

Table 3. Definition of integer or continuous parameters.

Parameter	Integer	Continuous
Mooring line diameter (mm)	✓	✗
No. of mooring lines (-)	✓	✗
Hawser line diameter (mm)	✓	✗
Footprint radius (FPR) (m)	✓	✗
Buoy 1 volume (m^3)	✗	✓
Buoy 2 volume (m^3)	✗	✓

2.4.4. Penalty Function

Naturally, the cheapest mooring solution will be the one consisting of the smallest amount of materials, because cost for components, installation etc. will be small, but this solution might not fulfil the design requirements defined in Section 2.1.2. In order to ensure that the solver accounts for this and search for the cheapest solution that also fulfils the requirements, a cost penalty is applied to the inadequate solutions.

Different types of penalty functions can be applied when the criteria are exceeded e.g. letting the cost be a fixed and high value or adding a fixed value to the actual cost. The first solution is not applicable as the result will provide a plateau on the response surface, and it will be more difficult for the solver to detect which solutions perform the best. The latter function will not provide a plateau, but will not be dependent on the performance of the system; hence, if the cheapest solution results in the largest exceedance of the requirements, it will still appear cheaper than a more expensive solution which performs better and is closer to satisfying the requirements. Consequently, the following penalty function is applied through the following equations (8) and (9):

$$p_i = \frac{X_i - X_{i,C}}{X_{i,C}} \tag{8}$$

$$\text{Penalty} = \sum_{i=1}^{N} p_i \cdot scale + Fp \tag{9}$$

where p_i is the penalty associated with the violation of one design criterion, N is the number of design criteria, X_i is the simulated motion or tension and $X_{i,C}$ is the associated design criterion. The *scale*-factor is defined as a fixed value in € and ensures that the mooring solutions that perform worst have the largest prizes, so that MATSuMoTo will diverge from them. The Fp-factor is defined as a fixed penalty in €, which is used to ensure that none of the insufficient solutions are cheaper than the most expensive and adequate solution. The *scale* and Fp-factors must be determined beforehand by assessing the extreme cost difference between the possible solutions.

2.4.5. Termination Criterion

The optimization is finished when the solutions are converged, hence the minimum mooring cost is found. As stated in [48,51], MATSuMoTo is asymptotically complete, indicating that if an indefinite number of calculations are performed, the global minimum will be found with a probability of one. However, the termination criterion is listed as a maximum number of evaluations, after which the minimum cost is identified.

OrcaFlex is generally computational effective and can run more simulations simultaneously. It is possible to run a potential large number of simulation and 300 evaluations were selected.

Considering Figure 8, the progress of the optimization for each of the cases is plotted. It is clearly observed how the optimization procedures manage to identify cheaper solutions until reaching a value where the solution converges. In the analysis of each case, the routine had one restart as mentioned previously, which means that the optimization is not based on one model constructed from 300 evaluations, but two models with less evaluations. Often an approach is used where the routine is not restarted in order to check the convergence with a higher number of evaluations, but this has not been done in this research and it is not expected to affect the result. From the figure, it is concluded that even less evaluations would have been sufficient for identifying an optimum. For Case 4, it is seen how the solutions in the DoE already provided a solution close to the optimum, but still manages to make further improvement.

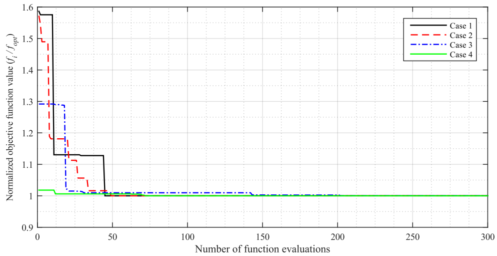

Figure 8. Progress of the found objective function values for the four optimization cases. The function values f_i have been normalized according to the optimum value f_{opt}.

3. Results

This section presents the outcome of the optimization routine when using the method as described in the previous section. Each case is presented separately and followed by a common discussion in the next section.

3.1. Case 1

The mooring solution in Case 1 consists of a synthetic turret system, and based on the optimization, the optimal layout has parameters as listed in Table 4. The mooring cost has been normalized according to the total CAPEX and OPEX of the WEC.

Table 4. Results of the optimization of the mooring system for Case 1.

Parameter	Optimum Value
No. of mooring lines (-)	6
Mooring line diameter (mm)	192
Footprint radius (m)	40
Horizontal stiffness (kN/m)	954
Normalized mooring system cost $\left(\frac{c_{mooring}}{c_{total}}\right)$	0.13

Figure 9(a) illustrates the evaluated points, with the optimum as a red marker. Figure 9(b) presents a contour plot for the cost of the mooring when normalized according to the total WEC cost. The figure presents systems with and without cost penalty, but the satisfying solutions are found in the normalized cost range 0.13-0.15. In this plot, it is possible to detect the influence from each parameter.

In the top right plot, the footprint radius is plotted against the number of lines. The line diameter is kept constant in this plot and corresponds to the optimum. It is clearly seen that the footprint radius only provides little influence on the cost, while approximately 15% cost reduction can be achieved by varying the number of lines. Naturally, a large number of lines increase the cost due to installation, the amount of anchors and line materials etc., but it also highly influences the mooring stiffness, corresponding motions and line tensions. Decreasing the line number from 6 to 4, decreases the mooring stiffness with 32% and results in larger excursions. The fewer lines also need to each take up a larger part of the load and the tension with four lines is 10% larger than with six lines, resulting in insufficient strength. Increasing the line number to 10 lines increases the stiffness with 63%, but the larger number of lines means that each line only takes 75% of the tension experienced in the system with six lines. This means that the anchor size can be decreased. The installation cost is primarily dependent on the number of lines and is, therefore, largest for the system with 10 lines (80% larger than the cost for four lines), but since the motion limit is exceeded with 4 lines, the cost penalty has been applied to the system, and the difference appears smaller in Figure 9. Clearly, the optimum value is found as a balance between finding the lowest number of lines, where low installation cost is present, and finding a high enough number of lines to ensure small tensions in the lines with a corresponding need for smaller anchors and certainty on the line strength.

Considering the number of lines against the line diameter, it is clear that the most dominating parameter is the diameter. Having a small line results in a very compliant system which in many cases causes a violation of the surge limit, while the line also provides less and insufficient strength. It is necessary to use some of the largest considered diameters in order to find a suitable solution, and the cheapest is found by using the strongest lines and reduce the line number.

In the final plot, this tendency is also observed as the footprint radius only provides minor importance while the line diameter is paramount to optimize. Considering all the graphs by Fig. 9, it is noticeably that a large part of the evaluation provides insufficient solutions and it would merely have been necessary to consider a few of the largest line diameters.

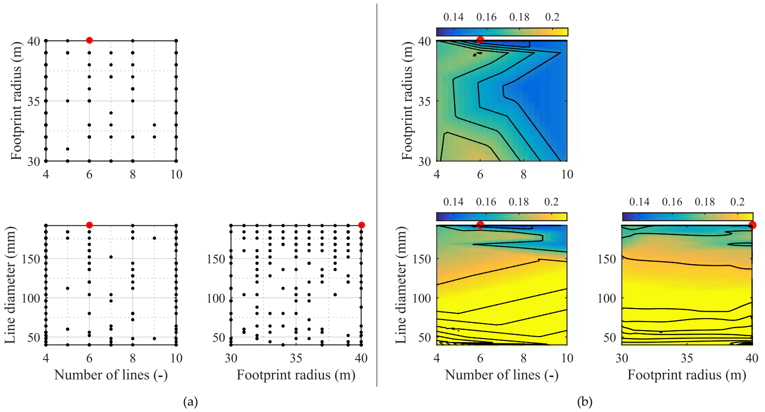

Figure 9. (a) Sample point from the optimization of Case 1. (b) Cost contour plot of the optimization of Case 1. The mooring cost has been normalized according to the total WEC cost.

3.2. *Case 2*

As presented in Table 2, Case 2 considers an overall mooring design similar to the system in Case 1. However, this structure is located in a larger water depth and has less restriction on surge and none in pitch, cf. Table 1. By also considering Table 2, it is observed that this case allows for a larger footprint radius, which provides a larger range to find an optimum solution. Table 5 presents the results from the optimization.

Table 5. Results of the optimization of the mooring system for Case 2.

Parameter	Optimum Value
No. of mooring lines (-)	6
Mooring line diameter (mm)	192
Footprint radius (m)	100
Horizontal stiffness (kN/m)	548
Normalized mooring system cost $\left(\frac{c_{mooring}}{c_{total}}\right)$	0.17

Figure 10(a) presents the sample points. It is clearly indicated that the solver identified the need for large line diameters and aimed the search at these diameters. Similarly, it is indicated that smaller footprint radii were sufficient.

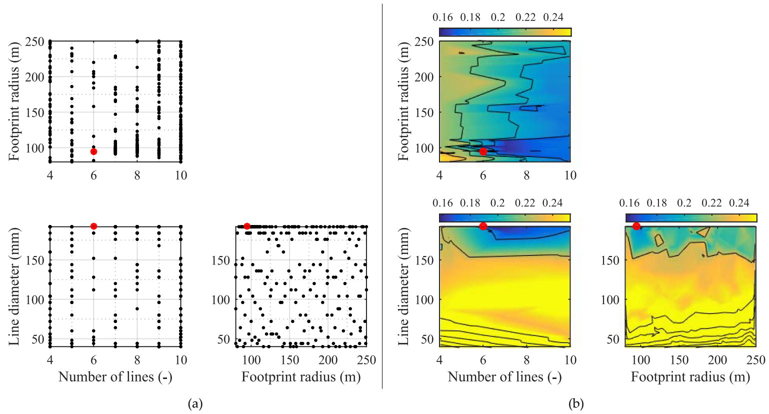

Figure 10. (a) Sample point from the optimization of Case 2. (b) Cost contour plot of the optimization of Case 2. The mooring cost has been normalized according to the total WEC cost.

Figure 10(b) presents the cost contour for the problem where the working solutions are found in the normalized cost range 0.15-0.19. Similar to the previous case, one parameter (the optimum) is kept constant in each diagram, and the influence from the remaining can be identified.

When considering the footprint radius against the number of lines, a similar tendency as in Case 1 is observed. Variation in the number of lines provides a cost difference of approximately 20%. This difference is primarily caused by the influence on anchor and line loads. A large number of anchors might cause a high cost, but having too few lines causes large loads and insufficient strength. The cost is less independent of the footprint radius.

In the plot of line diameter against number of lines, the influence from the former is seen to be crucial for the cost. Decreasing the line diameter leads to a paramount cost increase, because of the penalty function, as the line strength tends to become insufficient or the compliance so large that the excursion limit is exceeded. The number of lines plays a less important role, but it is seen how too few lines cause a higher cost.

In the final plot, the footprint radius and line diameter are presented with the number of lines kept constant. Similar conclusions can be made from this, which indicates that the main parameter is the line diameter. The footprint radius has a minor influence but by decreasing it, the line material is decreased and results in some minor cost reduction.

3.3. Case 3

Case 3 consists of a larger number of optimization parameters compared to the first two cases. Table 6 presents the result of the optimization.

Table 6. Results of the optimization of the mooring system for Case 3.

Parameter	Optimum Value
Hawser line diameter (mm)	176
Footprint radius (m)	50
Buoy 1 diameter (m)	3.7
Buoy 2 diameter (m)	6
Horizontal stiffness (kN/m)	46
Normalized mooring system cost $\left(\frac{c_{mooring}}{c_{total}}\right)$	0.48

When considering Figure 11(a), it is obvious that the code identified the minimum and concentrated the evaluation around this point. In Figure 11(b), where the working range of the normalized cost is 0.48-0.52, it is apparent that a significant parameter for the cost is the line diameter. Similar to Cases 1 and 2, only the largest line diameters provide sufficient strength, but in this system a larger range of diameters are adequate. Considering the very low stiffness presented in Table 6, it will be expected to obtain smaller loads. In the top graph, the influence from the two buoys can be identified. Buoy 1, which is the bottom buoy, provides the lowest influence on the cost, while the top buoy provides most influence in the system and hence determines the cost. The best solution is found by having a large buoy at top, while the other can be relative small. The footprint radius (and thereby the hawser line length) is also providing an important influence on the system, as it highly determines the overall stiffness. Having a large radius provides large compliance and mainly causes problems with exceedance of the surge restraint. A small radius still ensures satisfaction of all requirement, but results in larger loads and, hence, need for a larger anchor. Compared to Cases 1 and 2, the mooring cost of this system is taking up a larger part of the total cost. This is partly because the device itself is cheaper, but also because the gravity type anchor in this system is expensive and other types should be investigated.

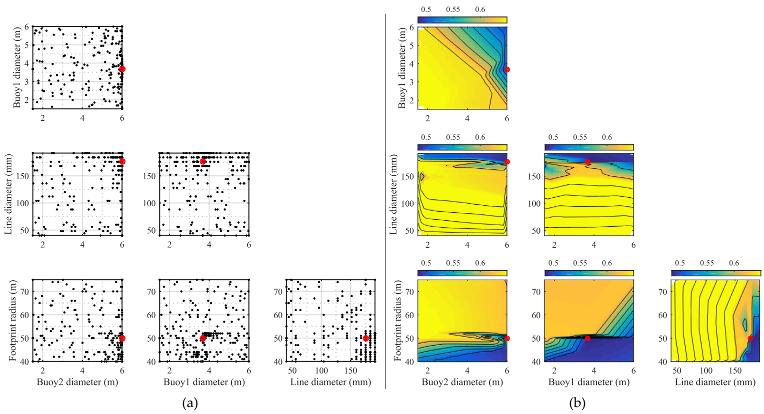

Figure 11. (a) Sample point from the optimization of Case 2. (b) Cost contour plot of the optimization of Case 3.

3.4. Case 4

As described in Section 2.2, the hydrodynamic model of the WEC in Case 4 provides a significant inaccuracy, which also affects the optimization procedure. Table 7 presents the result for this case.

Table 7. Results of the optimization of the mooring system for Case 4.

Parameter	Optimum Value
No. of mooring lines (-)	10
Mooring line diameter (mm)	192
Footprint radius (m)	91
Buoy diameter (m)	3.5
Horizontal stiffness (kN/m)	373
Normalized mooring system cost $\left(\frac{c_{mooring}}{c_{total}}\right)$	0.22

Due to the implications of using linear theory on an overtopping device, very large loads are seen and cause difficulty in finding an adequate solution. Consequently, the strongest line is chosen and with the highest number of lines. Figure 12 clearly identifies the problem of finding a solution since a significant part of the design space is giving a high cost due to the penalty function. The normalized cost of the working systems are in the range 0.22-0.24. The number of lines provides an influence as it helps distribute the loads into more lines and thereby secures sufficient strength in each and also lighter anchors. Similar, the footprint radius can be used for modifying the cost through its effect on the stiffness and load on anchors. Clearly, a large radius is desired as the longer lines introduce more compliance.

The buoy size is also playing an import part in the system response and cost. By having a large buoy with high buoyancy, the stiffness of the system is primarily an effect of the line stiffness, while also a large pretension in obtained. A smaller buoy results in more vertical compliance and reduces the line tensions. The optimizer identifies a small buoy as the most feasible solution and instead requires strong lines and larger footprint radius. Table 12 shows that this system results in a much stiffer system than any of the others.

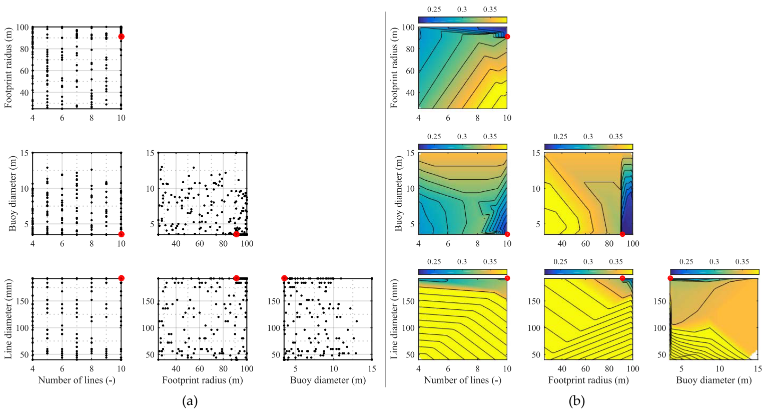

Figure 12. (a) Sample point from the optimization of Case 4. (b) Cost contour plot of the optimization of Case 4.

4. Discussion and Conclusions

This paper used a surrogate-based optimization model to find the most suitable mooring configuration for four large floating WECs, considering design in the ULS and aiming to find the least costly solutions. Based on the presented environmental conditions and design constraints for each device, numerical models were constructed in the BEM code NEMOH and time domain model OrcaFlex. In connection, a database of costs was constructed and used to calculate the total lifetime cost for each solution.

Based on the optimization routine, four working solutions were found. Even though it is not possible to detect from the normalized values in Fig. 9-12, it was found that the cost for Cases 1, 2 and 3 approached comparable values despite some differences in the mooring layouts as Case 3 is equipped with a buoy and hawser, while Case 1 and 2 are turret systems. The cost for these two are becoming high due to the turret system, while Case 4 becomes expensive due to larger loads and more anchors. Case 3 provides a clear low value when compared to the other cases and can both be explained by the fact that the mooring layout is much different, but also by the fact that the WEC is extremely light compared to the other devices and with a much smaller draught, which induces smaller loads on the WEC. As the devices are different in layout, it is not possible to directly compare the cost between the cases. In [8], the cost of mooring for a single point absorber buoy is listed to take up 8% of the total CAPEX, which for the current four cases are in the range 8-25%. The larger structures, hence, results in relatively more expensive moorings due to increased loads.

For all cases, the line diameter provided the largest impact on the cost of the mooring, as a relatively large line diameter was needed in the layouts in order to ensure sufficient strength to avoid line failure and to provide enough stiffness to avoid undesirably large motions. In the SALM system, some cost was saved by adjusting primarily the size of the top buoy as it highly influences the stiffness, cf. Fig. 13. When having a small buoy, the stiffness is low for small motions, but the risk increases of fully stretching the system where the stiffness curve becomes steep and large tensions can occur (also seen in Fig. 13). Similar, the footprint radius was important to restrain in order to avoid large motions. In addition to the line diameter, the SPM system was influenced by most parameters. In this case, very high loads were obtained due to the problems of using linear theory on the device, but it was shown how the number of lines and footprint radius could affect the stiffness and thereby loads and cost, cf.

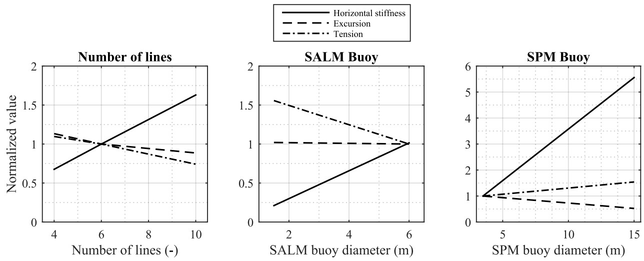

Figure 13. The influence on number of lines and buoy sizes on the mooring stiffness, tension and excursion.

Fig. 13. The surface buoy provided a great impact on the cost as it is a paramount influencer on the stiffness. Having a large buoy provides a much stiffer system, and thereby introduces large line loads, also because much more pretension is in the lines due to higher buoyancy. The study showed that decreasing the size as much as possible provided the cheapest solution. Similar to the SALM system, the use of a small buoy in this solution increases the risk of fully stretching the system, with risk of high loads. In this case, however, it was still found more feasible to use a small buoy, cf. Fig. 13. For this particular case, there might be more cost savings, possibly by first improving the model to account for the energy dissipation by overtopping. Afterwards, it is highly relevant to consider a synthetic hawser as well. This will introduce additional compliance in the system, but needs to be balanced with the stiffness from the buoy. Having a large buoy implies that much tension will be put in the hawser and, in this case, a small buoy is expected.

When considering Figure 9-12, it is clear that some of the optimum solutions are found in minimums on the response surface where the gradient is large. Even minor changes to the system layout in these areas can result in significant changes of the cost, meaning that the solution is very sensitive to input parameters and their uncertainties. In some applications, it would be reasonable to search for areas where also the gradient is low in order to find solutions where the cost estimate is more reliable. In addition, the study also showed that some parameters had a minor effect on the mooring cost. From a safety point of view, it would be beneficial e.g. to use more lines if the effect on cost is only minor in order to apply more redundancy and safety. These considerations are not included in the optimization routine, but a very strong benefit of the surrogate-based model is that information of the entire response surface is achieved and allows for additional and manual evaluation of the surface and potential use of other solution than the global minimum.

The paper showed that it was possible to use a surrogate-based optimization routine to determine an optimum mooring solution with only a limited number of evaluations. The total computational time for each case was in the range of 25-30 hours, which is reasonably low for a design process. For each case, a solution fulfilling all specified design criteria and ensuring survivability was found, and the parameters affecting the mooring cost for each layout were identified. In future studies, it would be natural to improve the hydrodynamic models further and investigate its further potential cost reduction. This study only considered a surrogate-based optimization procedure with a limited number of models, while many other exist, both surrogate models and other types of optimization routines. The advantage of the present method is considerable where only a limited number of function evaluations are needed and can be used for this type of problem.

Acknowledgments: The work in this study has been funded by the Energy Technology Development and Demonstration Program (EUDP) through the project "Mooring Solutions for Large Wave Energy Converters" (Grant number 64014-0139). The authors wish to acknowledge all project partners, particularly Floating Power

Plant, KNSWing, LEANCON Wave Energy and Wave Dragon for input to the project, data on the WECs and review of this paper.

Author Contributions: J.B.T., F.F. and J.P.K. defined the overall outline of the study. J.B.T. produced the numerical models and implemented the optimization routine. K.B. collected cost data and produced the cost database, with input from J.B.T., F.F. and J.P.K. J.B.T. ran the routine, analysed the data and made the outline and first draft of the paper. F.F., J.P.K. and K.B. provided review and input to the finalization of the paper.

Conflicts of Interest: The authors declare no conflicts of interest.

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Paper H

Sensitivity and Cost Analysis of Mooring Solutions for Large Renewable Energy Structures

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The paper is submitted to
OMAE, The 37th International Conference on Ocean, Offshore & Arctic Engineering, 2018.

SENSITIVITY AND COST ANALYSIS OF MOORING SOLUTIONS FOR LARGE RENEWABLE ENERGY STRUCTURES

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ABSTRACT

Mooring of offshore structures is a well-known concept in the naval and offshore Oil & Gas sectors, but has been proven to cause complications and be infeasible in the newer wave and wind energy sectors when applying the traditional solutions and experience. Early-stage wave energy concepts are often planned for deployment in shallow to intermediate water depths, where the traditional catenary systems are insufficient, and have experienced several failures due to large line loads. As a result, considerable amount of line and anchor material is often required and causes undesirably high mooring cost. A need for applying more novel systems and materials is present in order to increase durability and decrease the cost. This study focusses on three different mooring systems, which have been considered in earlier studies and makes direct comparison between the lifetime cost using existing cost databases and design procedures stated in design standards from certification companies. All systems are primarily composed of synthetic lines, and are designed using validated numerical models together with a surrogate based optimization routine. Furthermore, the study investigates the sensitivity on line tensions from varying environmental loads in order to quantify the sensibility and the importance of pre-assessment of the conditions at the expected deployment site.

INTRODUCTION

In the efforts of limiting the use of fossil fuels on a global scale, more renewable technologies needs to be implemented in the energy production. Moreover, a key factor is having multiple renewable energy sources in the energy mix [1]. This is a consequence of both the temporal availability of most renewables, and because no single source is yet at a level where it can single-handedly provide full energy grid saturation.

While some of the land-based energy systems such as solar and onshore wind turbines are well established and have extensive track-records for deployment, this is not to the same extent true for offshore technologies. Shallow water bottom-fixed wind turbines are in many ways a mature industry with 1,567 MW installed capacity in Europe in 2016 alone [2]. To further expand the renewable contribution from the offshore sector, less mature technologies like floating wind turbines and wave energy converters (WECs) have a significant potential.

Floating offshore structures are a well-known concept from the offshore oil and gas (O&G) industry. While many floating renewable systems try to adopt the mooring technologies developed in the aforementioned industry [3], the lower gross margin tend to be prohibitive to this approach. Instead, a leaner and more cost-effective mooring design is required as a part of ensuring a lower levelized cost of energy (LCOE) from these systems [4,5].

For wave energy this quest have until now seen some unfortunate events where floating pilot projects have stranded due to mooring system failure. As a consequence the Danish Energy Technology Development and Demonstration Program (EUDP) funded the project Mooring Solutions for Large Wave Energy Converters (MSLWEC) in 2014 [6]. The MSLWEC project was aimed at comparing the mooring system concept from several different wave energy developers, and supplying a more detailed and uniform procedure for mooring system design.

In the MSLWEC project, the mooring systems have been compared from both a response and cost perspective, but always associated to the WEC characteristics supplied by the developer. These characteristics vary widely in both weight, structural shape, size, hydrodynamic performance and means of energy absorption. Hence, no isolated comparison of the mooring layout

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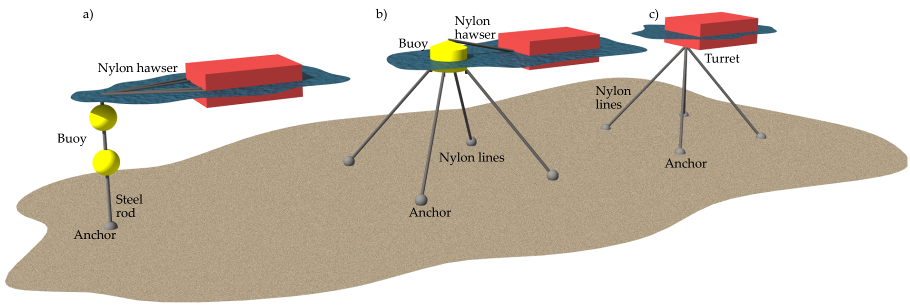

FIGURE 1. a) Single anchor leg mooring (SALM), b) single point mooring (SPM), c) turret mooring

has been performed. The present paper aims at carrying out this parametrized comparison. Three of the mooring concepts from the wave energy developers, all serving as station keeping for a barge structure with no relation to any given WEC concept, will be compared. The three chosen mooring layouts can be seen in Fig. 1. The chosen systems consist of the following: a single anchor leg mooring (SALM), a single point mooring (SPM) and a turret mooring. In all systems, the lines are composed of synthetic, nylon material.

The system comparison of the different mooring systems will be done after each system has been topologically optimized from a cost perspective, and hence a system cost comparison will be presented. To quantify the station-keeping performance of each layout, a frequency domain system response analysis is presented by means of response amplitude operators (RAOs). When assessing the different mooring configurations resilience to changes in the estimated environmental conditions, a parametric variation study is carried out, and the most influential contributers to additional systems loads are presented.

METHOD

The cost comparison and sensitivity analysis is taking its basis in a general structure without relation to any specific WEC, cf. Fig. 1. A barge geometry is considered with overall dimensions that are, nevertheless, determined in order to obtain some resemblance with large floating WECs, ensuring a displaced volume in the range of volumes of the WECs in the MSLWEC project. The structural dimensions are listed in Tab. 1 and the same structure and size is applied to all the mooring systems.

Similarly, the environmental conditions are chosen by considering the site assessment for the WECs, cf. Tab. 1. From this assessment, it was found that the environmental parameters could vary with up to $\pm 15\%$. Consequently, this range will be used to investigate the sensitivity on the mooring system behaviour.

The overall procedure used in the paper is listed below.

1. Determine wave-structure interaction.
2. Optimize each mooring layout, with respect to lowest cost, while also securing survivability.
3. Compare cost for the solutions found.
4. Test the sensitivity of each of the optimum mooring layouts.

Analysis of the Moored Structure

In many applications, numerical modelling procedures are applied for assessment of structure response due to a high efficiency, easy automation and ability to change parameters. Numerical models of the wave-structure interaction range from high-order methods like Computational Fluid Dynamics (CFD) or Smooth Particle Hydrodynamics (SPH) to simple and more time-efficient models like Boundary Element Model (BEM) or Morison's Equation. The present study applies a hybrid model which combines the diffraction and radiation loads from the BEM with the drag-contribution from the Morison's Equation, following the validated procedure in [7, 8]. The open-source code NEMOH [9] is utilized for calculation of the frequency-dependent hydrodynamic coefficients, while the commercial software package OrcaFlex [10] is used to simulate the coupled response of the structure and mooring. The software solves the Cummin's Equation [11], and includes all environmental impact from wind and current together with 1st and 2nd order wave loads [10, 12]. The output of the OrcaFlex model is a timeseries of motions and mooring line tensions.

In order to ensure survivability, design standards like DNV-OS-E301 [13], API-RP-2SK [14] and ISO 19901-7:2013 [15] re-

TABLE 1. DEFINITION OF STRUCTURAL AND ENVIRONMENTAL PARAMETERS FOR THE BASIS CASE.

Parameter	Value
Width	45 m
Length	65 m
Draught	5 m
Height above SWL	5 m
Mass	$15.0 \cdot 10^6$ kg
Significant Wave Height, H_s	8.3 m
Peak wave period, T_p	12.9 s
Spectrum	JONSWAP
Gamma	3.3
Wind velocity, v_w	35 m/s
Current velocity, v_c	1.3 m/s

quires validation of the mooring in the ultimate limit state (ULS) by consideration of 100-year extreme environmental loads together with application of safety factors. The present analysis considers the requirement in DNV-OS-E301, following the considerations in [16]. The requirement to mooring line strength is given in Eq. (1).

$$T_{C,mean}\gamma_{mean} + T_{C,dyn}\gamma_{dyn} \leq S_C \tag{1}$$

$$T_{C,dyn} = T_{MPM} - T_{C,mean} \tag{2}$$

where $T_{C,mean}$, $T_{C,dyn}$ and T_{MPM} are respectively the dynamic, mean and most probable maximum tension. γ_{mean} and γ_{dyn} are the partial safety factors for mean and dynamic tension, while S_C is the design breaking strength corresponding to 95% of the maximum breaking strength.

There are no strict requirements for excursion defined in design standards and it must be chosen based on site restrictions and according to umbilical.

Optimization of Mooring Layouts

The layouts in Fig. 1 each have several parameters that can be varied and that highly affect the performance and response of the system. Table 2 lists the parameters which are considered as variables for each of the mooring layouts. The same type of mooring line, Bridon Superline Nylon [17], is considered for all cases using the structural parameters, including mass and non-linear stiffness, as stated by the line manufacturer [17]. The present optimization problem is given in Eq. (3)

$$\min_{\boldsymbol{x} \in \mathscr{D}} f(\boldsymbol{x}), \tag{3}$$

Where $f(\boldsymbol{x})$ is the objective function calculating the cost of the mooring, $\mathscr{D}$ is the design space (optimization parameters) and $\boldsymbol{x}$ is the variable vector.

The objective function in the present case provides the cost of the mooring system and hence, is a coupled function of NEMOH, OrcaFlex and a cost database. For each point $\boldsymbol{x}$ a time series is simulated, the tensions calculated and used to design parameters such as anchors. The overall layout of the system is imported into a cost database, which calculates the overall CAPEX and OPEX cost for a full lifetime of 20 years. The CAPEX cost includes all cost from engineering, components, installation and decommissioning, while OPEX covers maintenance, replacements, insurance and inspection. A more detailed description of the database can be found in [16].

In order to avoid calculating every single configuration in the design space, an optimization procedure is applied. Several approaches exist [18] such as surrogate-based routines, which has the clear advantage that only a limited number of evaluations are needed and still, information on the entire response surface is obtained. This methods is considered in this paper, following the procedure in [16].

The optimization routine is based on the MATSuMoto toolbox [19], which follows a routine defined as:

TABLE 2. DEFINITION OF THE OPTIMZATION PARAMETERS FOR EACH MOORING CONFIGURATION IN FIG. 1.

Mooring system	Optimization parameters
SALM	Buoy dimensions
	Hawser line length
	Hawser line diameter
SPM	Line length
	Line diameter
	Buoy dimensions
	Number of lines
Turret	Line length
	Line dimensions
	Number of lines

1. Initial design of experiment (DoE).
2. Objective function evaluation in points of DoE.
3. Construction/updating surrogate model.
4. Sampling of new evaluation points.
5. Objective function evaluation in selected points.

In step 1-2 a number of layouts are chosen and the objective function evaluated in these points. The results are used to construct a model which approximates the objective function (Step 3). This is a clear advantage, as the routine now search for solutions in the area where optimum values are expected (Step 4). This procedure is continued until the found minimum value converges and the global minimum can be extracted.

When simulating the response in a given sea state, the line strength might be exceeded. In such cases a cost penalty is added to the objective function value in order to prevent the optimizer from searching for solutions in that area.

Sensitivity Assessment

The outcome of the optimization routine is the cheapest possible mooring layout within the specified ranges of the parameters. This solution does not account for sensitivity to uncertainty on input parameters. The most optimum solution will in great extent be the solution which requires the least material as it decreases the procurement, installation, decommissioning etc. The solution will, therefore, often be for a case where the full strength of the lines and anchors are used. This puts high requirement on the accuracy of the specified environmental conditions as even minor changes can potentially increase the tensions enough for failure to occur. In order to assess this topic, the sensitivity analysis evaluates the influence on T_{MPM} from changes in the environmental conditions. This implies that the optimum solution for each of the mooring systems is simulated with variation in the environmental conditions.

Since an optimized solution is found, it is possible to make a direct comparison between the cost of each, highlighting the most important parameters and identify advantages and drawbacks from each solution.

All cost and tensions are presented as normalized values in the paper. Because the geometry is a general shape, it is merely the relation between the values that are of importance.

RESULTS AND DISCUSSION

The results of the optimization routine for each mooring system is presented in Table 3, and presents the optimum number of lines, line diameters and lengths together with the buoy dimensions.

Each of the systems is ensured survivability according to DNV-OS-E301 and the environmental conditions presented in Tab. 1.

Each system induces a different response by providing varying stiffness in the degrees of freedom (DoFs). Fig. 2 presents the motions response amplitude operators (RAOs) in the surge, heave and pitch DoF and illustrates insignificant differences in heave and pitch. This is due to the relatively large hydrostatic stiffness of the structure compared to the mooring stiffness. The most visible difference is seen for the surge DoF. As presented in Fig. 3, the mooring stiffness in surge is in the same range for both the SALM and SPM system, while it is significantly larger for the turret system. This results in a higher peak frequency in the RAOs as seen in Fig. 2. Fig. 3 additionally illustrates how the larger stiffness in the turret results in a tension that is approximately four times larger than in the SALM system. Consequently, the turret system requires the largest line diameters and highest number of lines, cf. Tab. 3.

Considering Fig. 4, the cost of the SPM system is significantly lower than the cost of the other configurations. There are only minor differences in the engineering cost which covers parameters like site assessment, surveys, analysis, engineering, review and planning. The cost is based on the known day-rates and estimations on needed time. The engineering cost for the SPM and SALM systems is estimated to be similar, while the turret system cost becomes higher due to the complexity of designing the turret bearing.

The component cost forms the largest difference between the

TABLE 3. OPTIMIZED MOORING LAYOUT FOUND FROM THE OPTIMIZATION ROUTINE.

SALM	
Hawser line length	4×40.8 m
Hawser line diameter	Ø160 mm
Buoy 1 volume	113 m^3
Buoy 2 volume	59 m^3
SPM	
Number of lines	4
Unstretched line length	4×140 m
Line diameter	Ø128 mm
Buoy volume	16.8 m^3
Turret	
Number of lines	6
Unstretched line length	6×98 m
Line diameter	Ø192 mm

4

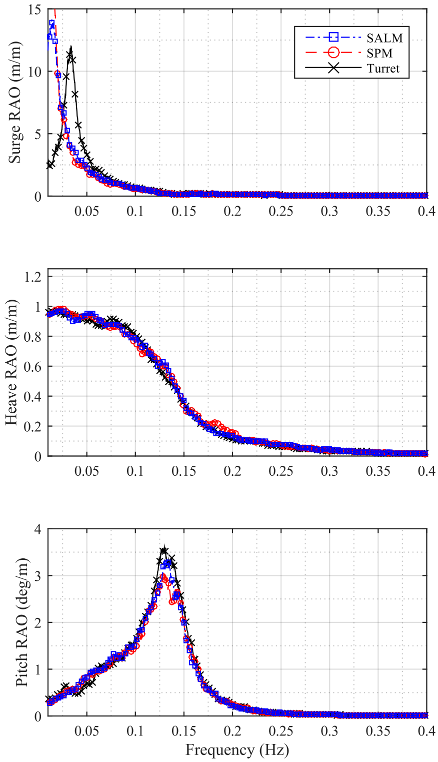

FIGURE 2. Motions RAOs for the three mooring systems.

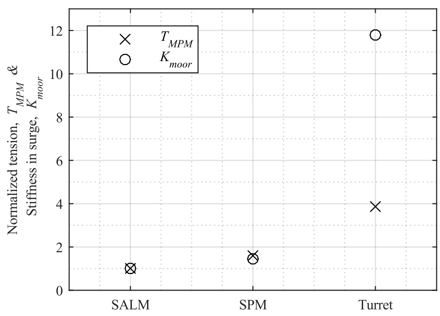

FIGURE 3. Comparison of tension, T_{MPM}, and horizontal stiffness, K_{moor} for the three mooring systems.

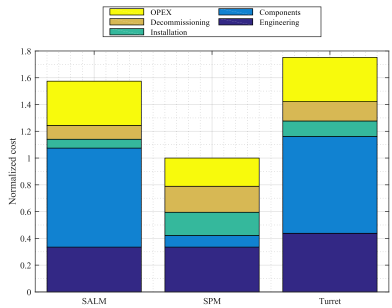

FIGURE 4. Normalized cost of the three mooring systems. The cost is normalized according to the cost of the cheapest system.

SPM system and the SALM and turret system as depicted in Fig. 4. Fig. 5 compares the cost between the different components in each system. The actual mooring line costs lead to only a small difference between the systems, while a paramount difference is seen in the anchor cost. The SALM system has only a single anchor point which needs to provide a significant vertical gravity based strength. This results in a substantial cost compared to the other systems. Considering Tab. 3 and Fig. 3, it is observed that the turret system has a larger number of lines and experiences larger line tensions. Consequently, the anchor and line cost is increased due to a larger amount of material, which, furthermore, also increases the installation cost.

The OPEX cost includes insurance as a percentage of the CAPEX, together with planned inspection and maintenance. These covers four inspections and re-tensioning of the lines over the project lifetime. The insurance and replacements are estimated based on the CAPEX cost. As a result, the inspection becomes identical for the three systems, while the SPM system becomes cheaper for maintenance and insurance. Despite using a relatively detailed and current cost database, the comparison is influences by some uncertainty. Topics such as handling of umbilical, which is treated differently in each system, is not considered yet and might influence the cost significantly. Similar, the gravity based anchor in the SALM system appears unrealistic in

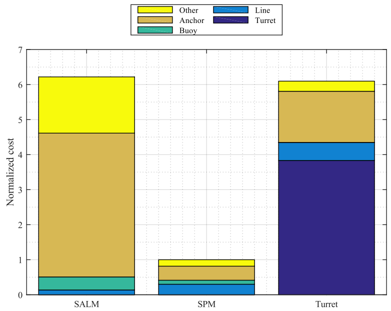

FIGURE 5. Normalized cost of the components used in the three mooring systems. The cost is normalized according to the cost of the cheapest system.

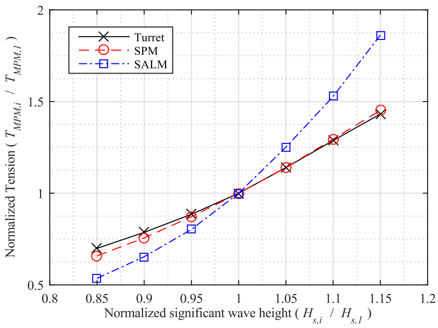

FIGURE 6. Plot of the mooring systems sensitivity to varying significant wave height H_s. The values are normalized according to the basis case in Tab. 1.

these conditions and other solutions like e.g. pile or suction anchors should be considered. Future research, which treats these topics more detailed, should make a more indicative comparison possible.

Sensitivity Analysis

The results of the sensitivity analysis are presented in Fig. 6-8.

Fig. 6 presents the variation in mooring line tension for variation in the significant wave height. As it is expected, an increased wave height results in increased tension and vice versa. The turret and SPM system tend to follow the same tendency and experience tension of $\pm$ 45% from changes in the wave height of 15%. The SALM system appears more sensitive to the variation and provides changes in the tension of -53% to $+86\%$. This is caused by the fact that larger wave heights result in a fully stretched system and a corresponding steep stiffness curve.

The sensitivity to changes in wave period as presented in Fig. 7 is not as intuitive as for the wave height. The tendency for the SALM and SPM is similar, with a maximum value at the basis case. The tension in the turret system increases with increasing wave period. The figure clearly illustrates the necessity to investigate a band of wave frequencies when designing the system, due to the unintuitive influence on tension, which in this case results in variations of up to 20%.

The water depth variations in Fig. 8 present a significant difference between the turret configuration and the other systems. The mooring lines in the turret system are connected directly to the structure, making it significantly more sensitive to water level variations. By increasing the depth by 15%, an increase of 21% in the tension is obtained. The SALM and SPM are almost unaffected by water level increase. On the other hand, a decrease in water level results in a decrease in tension of 10% for the turret system, significantly influences by the decrease of pretension, while the SPM and SALM system experiences an increase of only 5%. Since many early stage WECs are planned for deployment near-shore, this parameter becomes paramount to consider, as relatively large changes in water level can occur due to tides. Turret systems need careful consideration if deployed at sites with large water level variations.

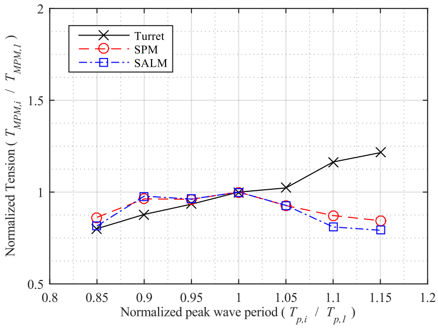

FIGURE 7. Plot of the mooring systems sensitivity to varying peak wave period T_p. The values are normalized according to the basis case in Tab. 1.

6

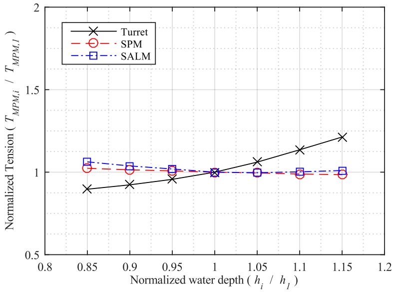

FIGURE 8. Plot of the mooring systems sensitivity to varying water depth h. The values are normalized according to the basis case in Tab. 1.

A series of tests were also conducted with varying current and wind speeds. The wind, as could be expected, did not cause any significant changes in the tensions ($\pm 6\%$), while the current had a larger influence with tension changes of $\pm 10\%$. The SALM system was most sensitive to these changes.

CONCLUSION

This study has used a barge structure as case study for comparing three different mooring configurations: SALM, SPM and Turret. A numerical model has been constructed, using the BEM code NEMOH and the time-domain solver OrcaFlex with inclusion of drag elements. Each system layout has been cost optimized by means of a surrogate-based optimization routine making sure to design the mooring components for the most probable maximum tension.

All the systems have similar performances in heave and pitch since these degrees of freedom are dominated by the hydrostatic parameters of the connected barge structure. Of the three systems the SALM and SPM have comparable surge stiffnesses. On the contrary the turret system has a significantly higher lateral stiffness of over a factor of 10. This is highlighted in the RAOs for each mooring configuration. The turret system also experiences the highest mooring tensions by almost a factor 4. This can be explained by both the thinner lines and the very fixed nature of such a layout.

A high mooring tension is not a suitable parameter to use when assessing system performance, since all have been designed to survive in the same conditions. Instead, the cost of each system is used. When summing up all the CAPEX and OPEX costs, it is clear that the SPM system has significant upside since the SALM and turret system are 58% and 76% more expensive respectively. The critical difference between the total costs of the systems is found in the cost of components, where the SALM and turret are more than 6 times as expensive as the SPM. For the SALM system, the biggest expenditure is by far the cost of anchors. This is intuitively also a focus point of such a system, since all station-keeping is supplied by a single anchor point, and this point also being exposed to significant vertical forces. For the turret system the biggest cost driver is the turret itself. Despite detailed cost data, several parameters are not yet included and might influence the overall cost from each system. Gaining experience and knowledge in this topic, forms a critical point for development of mooring solutions.

Since all three systems have been optimized to utilize their components as much as possible, it is of interest to highlight how sensitive these solutions are to changes in design conditions. Design conditions have been varied by $\pm 15\%$ to show the correlation between the environmental conditions and mooring tensions. For the wave height, a high dependency was found, as expected. While the SPM and turret systems both experience around 45% increased tensions at 15% increased wave height, the SALM configuration was almost twice that, at approximately 90% increased line tension. For changes in the wave period, the impact was less severe. For longer period waves the SALM and SPM configurations experience lower line tensions while the turret system saw an increase of around 20%. This opposite behaviour by the systems can be explained by to the difference in surge natural frequencies. For changes of water depth, negligible influence was seen for the SALM and SPM systems. For the turret system, increased tensions of almost 20% was found. Again, the taut and fairly rigid layout of such a system makes it susceptible to changes to the environmental conditions. Changes to wind and current velocities did not significantly influence the three systems, compared to the other parameter presented.

This paper has shown a comparison between three mooring configurations currently being applied by developers. While each system is able to support the chosen fictive barge structure, their costs differ widely. It is seen that the SPM configuration stands out as a significantly cheaper alternative. Of course, these optimizations should be carried out for the actual device proposed by the developer to include the correct system properties and also capture the coupling between structure and mooring. It is also seen that the systems have different sensitivities to design conditions. This should be pro-actively addressed under the site explorations.

ACKNOWLEDGMENT

The present study has been funded by the Energy Technology Development and Demonstration Program (EUDP) through the projects "Mooring Solutions for Large Wave Energy Converters" (Grant number 64014-0139) and "TetraSpar" (Grant number 64017-05171). The authors wish to acknowledge Kevin Black and James MacKay from Tension Technology International for providing valuable cost data.

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Mooring Solutions for Large Wave Energy Converters
by Jonas Bjerg Thomsen

SUMMARY

The desire for utilizing the sea waves as an energy source has a long history, but the wave energy sector did not experience concentrated focus until the seventies. With today's growing world population and living standard, and the resulting increase in energy consumption, the need for production of renewable energy, such as wave energy, is more important than ever. Despite the relatively long development history and a significant untapped wave energy potential, none of the patented concepts have yet reached a commercial level that allows for feasible deployment and energy production. In this matter, mooring of wave energy converters forms an important topic, as it has not yet been optimized to the wave energy sector, and has given rise to an undesirably high cost and several strandings of devices due to mooring failures.

This thesis investigates mooring solutions for large wave energy converters by taking its basis in existing solutions for a number of devices. The work develops from the initial starting point and uses it to discuss and explore different system layouts and materials in order to find promising solutions. Different design approaches are assessed and compared through validation against physical tests to improve the mooring safety and reliability. Finally, an optimization routine is applied in the design process in the search for cost optimized and durable solutions.

ISSN (online): 2446-1636
ISBN (online): 978-87-7210-112-5

AALBORG **UNIVERSITY PRESS**